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Foreword

VOLUME 55 of the Transactions of The American Society of Mechanical Engineers contains the individual sections published during 1933 under the sponsorship of the Society's professional divisions, as well as the annual Record and Index. The technical papers and reports that make up this volume represent the Society's annual contribution to the permanent record of mechanical-engineering achievement. Most of these papers and reports were presented at meetings of the Society and of its professional divisions and local sections, and were issued originally in pamphlet form. These pamphlets appeared in sections, each section being sponsored by one of the Society's professional divisions. Other papers published by the Society but not included in the Transactions may be located by means of the Index on page 87 of the Record and Index which forms a part of this volume.

All sections of the Transactions are bound together at the end of the year for the convenience of libraries and of engineers who wish all of the papers in permanent form. Copies of the bound Transactions will be found in depositories located in selected engineering, university, and public libraries throughout the world. A complete list of these depositories will be found on pages 83 to 86 of the Record and Index. Copies of the bound Transactions have also been set aside for sale.

The Record and Index of the A.S.M.E. for 1933 which forms a part of these Transactions, is the permanent record of the Society's activities for the year. It includes lists of committees, reports of meetings and committee activities, memorial notices of deceased members, as well as indexes of special A.S.M.E. publications and of *Mechanical Engineering*.

In this volume the papers of each section are grouped together and are numbered serially. The sections are arranged alphabetically. Each paper is designated by a symbol composed of key letters followed by the volume number and paper number. The letters refer to the section of the Transactions to which the paper is assigned. Thus AER refers to the Aeronautic section, IS to the Iron and Steel section, etc. The arrangement of the index and designations for locating material are explained at the beginning of the index, page 96 of the Record and Index, at the end of this volume.

THE PUBLICATIONS COMMITTEE

Progress in Aeronautical Engineering

MILITARY AERONAUTICS

By B. D. Foulois[1]

THE development of military aircraft during the last year has resulted in better performance and increased reliability. The flying characteristics of aircraft have been increased considerably by aerodynamic, structural, and propeller research. New data have been gained, especially with reference to streamline wheels and retractable landing gears. Cockpit covers are being employed to lessen the air resistance. The present tendency is toward closed cockpits for all types of aircraft.

The high stresses resulting from increased speeds now being reached have required more detailed analyses of airplane structures. Research into new construction methods and the attendant applications of better materials have made it possible to compile new and improved design standards. These standards and criteria have been incorporated in a new seventh edition of the "Handbook of Instructions for Airplane Designers."

Propeller development has been directed toward improvement of aluminum-alloy types and production of controllable propellers. Various alloys of magnesium are being developed with the object of utilizing the large savings in weight made possible by this metal. Welded aluminum alloys are also being tested. Two types of controllable propellers have survived preliminary tests and are being given service test. New propeller-testing apparatus has recently been installed at Wright Field and is being utilized in the study of the problem of reducing propeller noises, both in the air and on the test stands. This problem is one of such difficulty that a great amount of research will be necessary before any practical results may be expected.

Structural Designs

Structural research is being particularly directed toward wing design. With the same power, the high performance of military airplanes is directly dependent upon a proper combination of aerodynamic and structural qualities. With this in mind, stress analysis and design were completed for an all-spot-welded corrosion-resistant steel wing of stressed-covering, single-web, cantilever type. The structural members are utilized for the distribution of engine coolant vapor and for collecting the condensate in the wing for return to the engine. This arrangement eliminates the drag and much of the weight of the conventional airplane radiator.

The facilities for stress research have been improved by the development of new equipment for measuring strains in aircraft structures during static test. In carrying out static tests, the stresses in the airplane structures have been actually measured instead of merely finding the ultimate stresses of various types of complete structures. It has been a general practice to use the McCollum-Peters electric strain gages for the study of each structure while under static load, which makes possible a more accurate check of each stress analysis.

A new mechanical type of vibratory tensiometer and frequency meter has been designed and tested to provide a means of measuring the rigging loads and the loads in flying and landing wires when a structure is statically loaded. This apparatus has been successfully used to determine tension loads in round, square, and elliptical wires and tierods and to determine the natural frequency of various types of propeller blades. A new instrument for more accurately and quickly measuring chords of wings having thick airfoil sections has been developed and is being used with good results.

Aircraft

The standard pursuit-type airplanes are Boeing P-12's and Curtiss P-6's.

The air-cooled type typified by the P-12 has proved quite satisfactory in service. They are powered with SR-1340 engines. The liquid-cooled type is typified by P-6E and has been generally satisfactory, although trouble has been experienced with powerplant installations. These airplanes are powered with V-1570 engines, Prestone cooled.

The B/J Y1P-16 is a biplane aircraft powered with a V-1570 liquid-cooled engine. (See Fig. 1.)



Fig. 1 Berliner Joyce Y1P-16 Biplane

New types under test include the Boeing P-26 and Curtiss XP-23. The P-26 is an all-metal semi-low wing, wire-braced airplane. An unusual feature of this model is the fixed stabilizer. Longitudinal stability is procured by small trailing edge flaps on the elevators, operated from the cockpit in the same manner as the conventional adjustable stabilizer. This airplane promises to attain a speed of over 200 mph at 6000 ft.

The XP-23 is a metal monocoque biplane powered by a G1V-1570 engine and side-type supercharger.

The Douglas BT-2C is a standard basic training airplane.

A new attack airplane, the XA-8 (Curtiss), has been given full performance tests. As a result thirteen of this type have been ordered by the Air Corps. They are powered with V-1570E Conqueror engines. The XA-8 is of all-metal construction with low cantilever wing, externally braced. Slots and flaps are used. The monocoque fuselage has a complete convertible pilot's cockpit inclosure and a partially convertible gunner's cockpit inclosure. Two fixed, free-firing guns are located in wheel fairings on each side above the landing wheels. A high speed of 190 mph can reasonably be expected of this plane.

A considerable advance is represented by the Boeing B-9 bombardment airplane. This airplane, powered with an R-1860 engine, showed remarkable performance. It is also being tested with the G1V-1570 engine to secure comparative results with each power plant. The Boeing plane is a low-wing monoplane with monocoque-fuselage, all-metal construction, with retractable landing gear.

[1] Major-General, U. S. Army; Chief, Air Corps, War Department, Washington, D. C.

Presented at the session of the Aeronautic Division, Thursday morning, December 8, 1932, during the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Personnel of the Executive Committee of the Aeronautic Division for 1932: Elmer A. Sperry, Jr., Chairman; Alexander Klemin, Secretary; Carl B. Fritsche, Porter H. Adams, William B. Mayo, and Fred G. Coburn. Associates: H. H. Blee, J. L. Walsh, P. G. Johnson, T. A. Morgan, Archibald Black, V. E. Clark, J. H. Doolittle, W. F. Durand, E. E. Aldrin, William Knight, Bernt Balchen, Jerome Lederer, and T. P. Wright.

The Glenn Martin light bomber airplane is also a low-wing monoplane of all-metal construction. It has developed an unusually high rate of speed in preliminary tests.

The Douglas light bomber is being given service tests. Seven of these planes have been procured, powered with GV-1570E engines.

Standard observation types comprise the Thomas Morse O-19E, the Douglas series O-25C and O-38B, and Curtiss O-39.

Now undergoing service test as a corps observation airplane is the Douglas O-31. This airplane has smooth-skin monocoque fuselage and the G1V-1570-C power plant. The first seven have a gull wing externally braced with tierods. The last five airplanes will have cantilever gears and larger rear cockpits. A high speed of 195 mph can reasonably be expected from these types.

Another service-type observation is the Curtiss YO-40A of the sesquiplane type, powered with SR-1820 Wright Cyclone and equipped with retractable landing gear. One plane of the Army observation type is undergoing test. The Douglas Y1O-35 has a gull-type externally braced wing with monocoque fuselage, retractable landing gear, and two G1V-1570E engines. A normal high speed of 178 mph can be reasonably expected.

The General Aviation YO-27 has a fuselage of tubular steel construction, fabric covered. Two G1V-1570C engines are installed ahead of the leading edge of the wings. The undercarriage retracts into the engine nacelles. The pilot's cockpit has a convertible inclosure. A high speed of 170 mph can be reasonably expected.

An interesting development is the TC-13 airship, non-rigid type of 360,000 cu ft capacity, as compared with 210,000 cu ft for the present standard TC airship.

This airship is now under construction and will employ two-geared air-cooled engines, resulting in 700 hp available for driving specially designed three-bladed propellers. As far as is known, this is the first application to non-rigid airships of geared air-cooled engines, and with the propeller designed for a low rpm consistent with the air speed, a very quiet and efficient installation will result. Another feature of the airship is the five-fin control surface arrangement, with which it is expected to improve ground clearances for maneuvering, low speed control, and surface stability at the higher speeds, and to simplify the rudder-control mechanism. It is expected to obtain a high speed of 65 mph for an endurance of 15 hr, a cruising speed of 52.6 mph for 35 hr, or a hovering speed of 25 mph for an endurance of 100 hr. The airship will be equipped with an observation car that may be lowered a distance of 1500 ft below the airship for special forms of observation. Air blowers, bomb racks for 700 lb of bombs, a long-range radio set, and sleeping and galley accommodations constitute the other major items of equipment. The observation car and windlass arrangement, air blowers, sea anchor, ballast pick-up, and auxiliary power-plant unit for the radio set are being developed by the Lighter-Than-Air Unit.

Another development along this line is the TC-14 airship car, which will be interchangeable with the car on the TC-13 airship, but will omit the bombing and observation-car equipment in favor of a three-engine power-plant installation. This feature is also considered as an innovation in non-rigid airship design. Two of the engines will be geared air-cooled radial engines, whereas the third will be an inverted air-cooled engine of approximately 100 hp mounted in the rear of the car which will drive a controllable- and reversible-pitch propeller. By this arrangement it is expected, through fuel economy, to increase the endurance of the airship at slow hovering speeds, as well as to permit more flexibility of the power-plant installation for take-off and landing conditions.

New automatic pressure-control equipment represents the first attempt to devise a means of automatically regulating and controlling the internal pressure in airship envelopes while in flight to compensate for fluctuations of the internal pressure caused by bumpy weather or changes in air speed. Heretofore, this has been manually controlled by operation of eight controls, which can be eliminated by this device. The inspection equipment for airship envelopes is also a new development and consists of a gas helmet and related air-supply system by means of which it is possible to enter and remain in the gas compartment of an inflated airship for an extended period of time and conduct inspection and minor repairs to the interior of the envelope. This equipment will obviate the necessity for deflation of envelopes to accomplish repairs to ballonets, internal air lines, and suspensions, and should thus increase the life and simplify the maintenance of airship envelopes.

The motorized observation-balloon project has been continued by altering present airship equipment. Tests will be conducted during the latter part of this year to determine the feasibility of motorizing the observation balloon, in order that the balloon may be flown from one position to another rather than maneuvered over the ground while attached to a cable, which is the present standard practice.

Power Plant

Power-plant developments were generally concerned with improvements in performance of service engines and accessories. A large number of tests were conducted, including carburetors, cooling systems, fuels, lubricants, spark plugs, radio ignition shielding, magneto and battery ignition, pressure indicators, silencers, and superchargers.

A new laboratory was completed affording more efficient facilities for dynamometer tests and fuel research. This laboratory is equipped with refrigeration and air-exhaust apparatus to provide a simulation of winter and altitude conditions.

This installation allows the testing of engines under approximate altitudes as high as 30,000 ft with a temperature of —55 F and an air pressure equivalent to the barometric reading at that altitude. This apparatus also permits testing of engines and accessories under simulated severe-weather conditions.

In general, improvements in air-cooled engines have been obtained by a means of higher gear-ratio superchargers, higher compression ratios, and improved cooling characteristics. Many tests were carried out to locate the cause of difficulties reported by the service and to institute proper corrections. The development in liquid-cooled engines is being directed toward obtaining engines with ratings approaching 1000 hp. Some difficulty has been experienced with liquid-cooled engines in service, but steps have been taken to eliminate the source of trouble.

Lubricating Oil. A problem of lubricating oil is the occurrence of decomposition, resulting in production of solid gummy deposits in the cylinder. Correction requires research into lubricating-oil properties. Studies undertaken along this line have resulted in drastic revisions in specifications, the new oils showing marked improvements in conditions of the engines after extended periods of research.

For the present, the Air Corps specifications for oil require a flat viscosity curve nearly equal to the best Pennsylvania characteristics and fairly low pour points, which necessitates rather severe but not the severest dewaxing. Decomposition characteristics are not specified, but are apparently fairly uniform as a result of control of viscosity-temperature curve characteristics. The Air Corps is entering upon a fairly large-scale test program of oil produced by aluminum-chloride synthesis. Preliminary tests and investigation have shown very encouraging results.

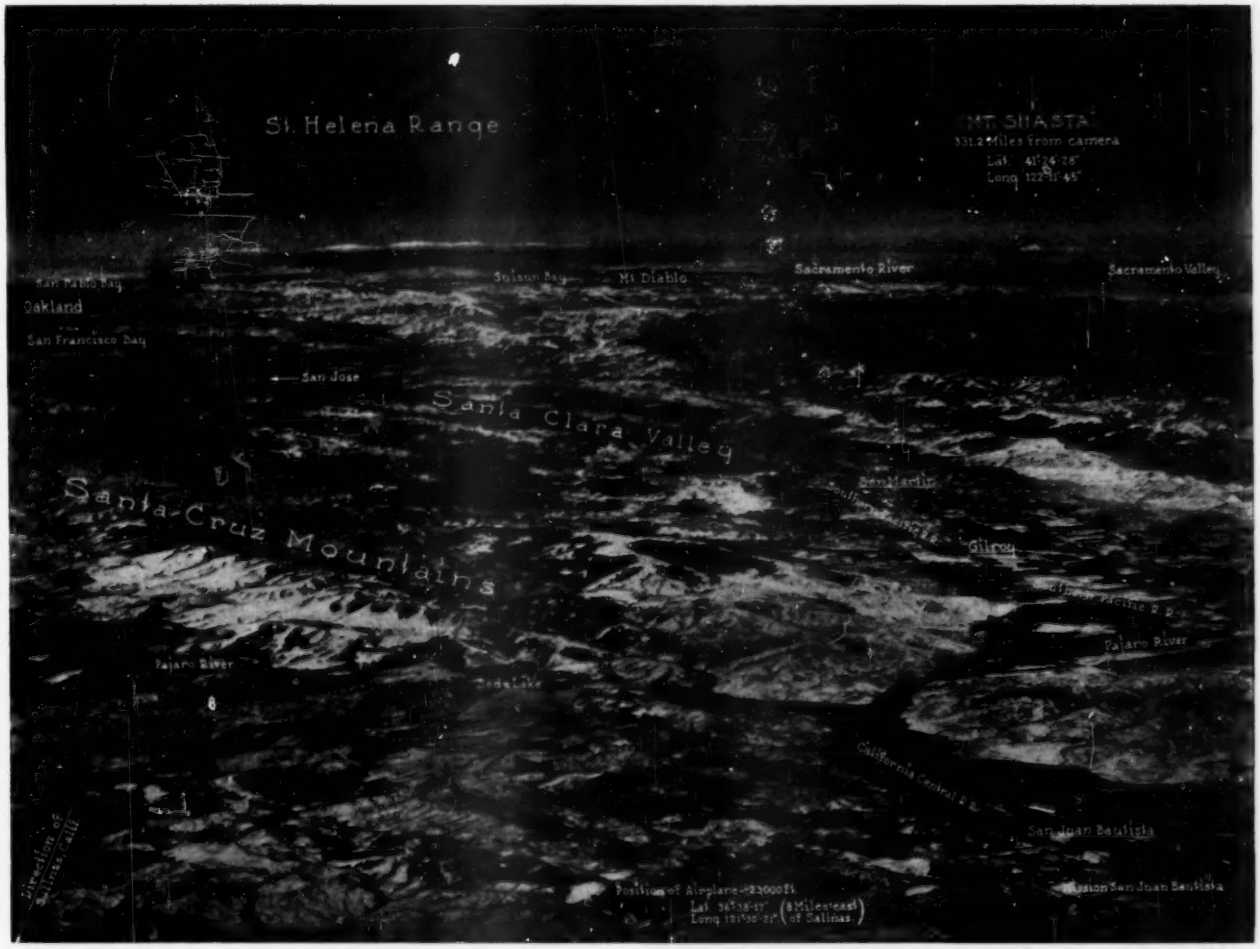


FIG. 2 LONGEST RANGE PHOTO EVER OBTAINED

(Taken in January, 1932, by Captain A. W. Stevens, Chief of Aerial Photographic Unit, Wright Field, flying at an altitude of 23,000 ft, and clearly showing Mt. Shasta at a distance of 331.2 miles, the one exposure covering 7200 sq miles, which also is the greatest amount of the earth's surface ever encompassed in one photograph.)

Engine Pressure Indicators. Development of engine pressure indicators has resulted in two types being successfully developed. The electric-telemeter type was designed to show pressure variations either by observation on a screen or by photographic record. This indicator has the advantage of showing every cycle of engine operation, in terms of either a pressure-volume or a pressure-time relation. The other type developed was of the sampling valve type, which takes a record on a standard steam-engine indicator. By means of these indicators the valve timing of engines has been improved, especially in the case where back pressure is imposed on the exhaust system.

EQUIPMENT

Aerial Navigation. Definite progress has been made in the field of aerial navigation through extensive research and experiments in application of radio phenomena to fog flying and blind landing. New gyroscopic turn and flight indicators have been developed in cooperation with the industry.

A radio compass has been developed which functions as an accessory to the standard aircraft receiver and gives visual indication of the correct direction for flight toward any selected radio-transmitting station. Successful flight landings have been made by the aid of this compass, in some cases supplemented by the high-frequency or ultra-short-wave field-boundary marker.

The fog landing system developed by the Air Corps is characterized by simplicity and low weight. An inexpensive ground transmitter of the 50-watt airplane type is used as contrasted to the costly and complicated radio beacon. Results have been so satisfactory in tests at Wright Field that this system will be put in service in a few selected Air Corps stations very soon. Further tests are in progress to standardize operating procedure and insure the maximum of reliability. A thorough investigation of radio shielding has been carried out. It is realized that radio sets are seriously hampered by interferences emanating from other electrical appliances on the airplane. Over 90 per cent of this normal interference has been eliminated by a new system of electrical installations on aircraft. Basically it employs two-wire ungrounded current in solid conduit with no fuses. The solid conduits have proved more efficient than the braid shielding system. Important phases of this work include integrally shielded starting motors, spiral-slot generator armatures, double-shielded generator controls, a more satisfactory ground for booster magneto and coil, and integrally shielded storage batteries. It is of interest to note that with this system a single conduit can be satisfactorily used from the pilot's cockpit to the engine compartment.

Three mapping projects were completed, for the U. S. Geological Survey, the Coast and Geodetic Survey, and the Corps of Engineers. These projects used the T-3A five-lens cameras

and obtained photographs from which maps were subsequently prepared.

Extremely encouraging results have been procured with the new Kryptocyanine film, which is used for the purpose of obtaining long-distance and high-altitude photographs. This is used in conjunction with the 89-A filter, which excludes practically all visible light; thus the photographs are taken with infra-red light. This type filter and film will be supplied to the service during the coming year. During experimental work with this film some very interesting pictures were taken. In January, 1932, Captain A. W. Stevens, Chief of the Aerial Photographic Unit, Wright Field, flying at 23,000 ft, obtained a picture of Mt. Shasta from a distance of 331.2 miles. (See Fig. 2.) This is the longest-range photograph ever obtained, and in one exposure covers the greatest amount of the earth's surface (7200 sq miles) ever encompassed. The negative was made from a point more than 100 miles farther south than the last point from which Mt. Shasta was visible with unaided vision. Captain Stevens also obtained the highest altitude photograph of history in an exposure made at 39,150 ft of territory lying in the vicinity of Rushville, Ind.

Besides the developments mentioned, a large number of other improvements have been made, especially in the direction of increasing the strength and reliability of aircraft.

Special mention of the Air Corps Inspection System should be made. Instituted several years ago, it has been constantly improved, and it is directly responsible for lessening the number of matériel failures. Besides prescribing detailed and uniform methods of aircraft inspection, all unsatisfactory performance reports are analyzed and the defects found are corrected. As a result the standard of reliability is high in spite of the increased speeds at which military aircraft are operating.

NAVAL AVIATION

By William A. Moffett[1]

The fiscal year 1932 has been an important one in the history of naval aviation. The close of the previous year saw the completion of the Navy's five-year building program, and left the Navy with a complement of modern aircraft with which to carry out its missions and maneuvers.

The early part of the year saw the commissioning of the U.S.S. *Akron*, the first fleet airship and aerial airplane carrier, equipped with specially designed high-performance fighting airplanes, and with provisions for their starting, alighting, and housing while the airship is in flight. The U.S.S. *Los Angeles*, after eight years' operation in valuable experimental and training work, has been decommissioned and laid up in the Lakehurst airship dock, in the interest of economy. Increased production of helium and the development of improved tank cars have further reduced the price of helium, so that it is no longer the important item of cost it once was. In addition, experimental work was started in connection with fuel gas for airships, using a K-class non-rigid airship of 320,000 cu ft volume for the purpose. Preliminary work on the Naval Air Station, Sunnyvale, Calif., the Navy's first West Coast airship base, has been actively under way during most of the fiscal year.

Dive and pull-out test results have necessitated a complete overhauling of former assumptions used in structural analysis. Research in this connection with newly developed instruments which record the stress and strains in the airplane structure in action have resulted in the establishment of new criteria which represent much more nearly the actual conditions in the aircraft under consideration than the old ones. A method of conducting basic structural tests on hull sections, which will result in a rational design basis, has been perfected. In addition, series of beam tests and destruction tests on complete airplanes have increased the fund of data available to the airplane designer.

Marked progress has been made in the development and utilization of corrosion-resisting steel in the construction of naval aircraft. It appears that many service difficulties are being eliminated by the use of this material.

The light-weight high-strength aluminum alloys continue to play an important part in naval aviation, and it is particularly gratifying to note the remarkable development in shop practices and maintenance methods made in connection with the use of these materials. Magnesium is also being thoroughly investigated, as it appears to have decided possibilities for use in the fabrication of non-structural parts.

Probably no one thing has aided more to improve the reliability and life of the aluminum alloys in aircraft than the consistent improvements which have been made in protective coatings. As the result of comprehensive study and research conducted at the Naval Aircraft Factory in Philadelphia, new specifications have been prepared which insure the procurement of high-quality durable finishes.

In the aircraft power-plant field, the power output of existing types of engines has been increased about 10 per cent for the same piston displacement. Supercharging is available and is being incorporated where desired. The increased scope and severity of engine tests, together with improved test conditions which simulate more closely flight conditions, have all helped to improve reliability and durability, resulting in a substantial reduction in maintenance costs. The present endurance test includes a 150-hr run. Most of the 1932 special appropriation for high-speed development was spent on power-plant development. During this year the two-row radial engine has been successfully tested in laboratory and flight and put on service test.

A study of aircraft trouble reports has focused attention on "plumbing," since it appears that it, especially in the fuel system, is the chief source of trouble.

Fuel and oil specifications have been revised upward to take advantage of the improvements in the refining art in the United States. In addition, investigations of fuel injection and safety fuels have been undertaken.

Hollow steel propellers have been developed to serviceable standards and are proving satisfactory in service. The variable-pitch propeller, having passed satisfactory tests, is now about to be put into use in service squadrons.

Considerable progress has been made in the development of aircraft instruments. These developments include a compass for scouting planes, an azimuth compass for patrol planes, an extremely sensitive altimeter, an improved octant, an accelerometer for dive bombers and fighting planes, and combined engine-gage units.

Experimental quantities of improved ground-speed indicators and drift sights have been purchased and have been given extensive service tests. Dead-reckoning navigation has been facilitated by improved chart-board equipment and its accuracy increased by the new types of compasses now in use.

The first production dive bombers, a new and distinct type of air arm, were put into service during the year. Several contracts were awarded for two-seat fighters and for two-seat scouting airplanes that embody numerous novel and interesting features. Some of these scouts are of the amphibion type.

The development of suitable radio equipment for aircraft has been pushed, especially in the direction of providing adequate radio for fighting planes and dive bombers. The entire electrical system of naval aircraft has been given more careful

[1] Rear Admiral, U. S. Navy; Chief, Bureau of Aeronautics, Navy Department, Washington, D. C.

study during the past year than ever before. Increased radio sensitivity has demanded more complete shielding of all electrical equipment and wiring. Complete conduit systems have been provided. Engine-driven generators of adequate capacity to handle the power required by both the radio and the electrical system have been developed.

Last winter there was undertaken the first of a series of tests to determine the feasibility of and the difficulties to be encountered in the operation of an aircraft carrier under severe winter conditions. The aircraft carrier U.S.S. *Langley*, with a skeleton complement of airplanes, spent two weeks off the New England coast and carried on routine flight operations.

While the results were very gratifying, these tests have indicated certain lines of development which are now being pursued with a view to improving operations under conditions of severe weather.

Experiments in an attempt to decrease the weight of winter flying suits have been particularly fruitful. Several experimental suits were made and tested under severe operating conditions during the past winter. As a result of these tests, certain minor changes were made in types found most suitable. It has been possible to decrease the weight of these suits 50 per cent, as a result of these studies. Suits have also been made considerably less bulky. A large number of these improved flying suits are being manufactured for service use.

Extensive researches are under way to provide the service with a satisfactory quick-attachable parachute. There appears to be a real need for parachutes of this type, and every effort is being expended in the development of such a parachute. Several types are now undergoing service test. In addition, a parachute with a quick-release harness is being thoroughly studied to determine its suitability for naval usage. Investigations are also under way to determine the suitability of cotton as a substitute for silk in the manufacture of parachutes.

The development of various equipment, including fire extinguishers, life rafts, and oxygen breathing apparatus for high-altitude work, has been proceeding at a most satisfactory rate.

AIR TRANSPORT

By Archibald Black[3]

Apart from the changes in airplane design covered in another section of this report, developments of a business nature represented the most important factors in air transport during the past year. Passenger traffic has continued to increase very substantially despite the unprecedented severity of the general business depression. (See Fig. 3.) The rate of increase, however, was not quite so rapid in the past twelve months as in previous periods. For the first seven months of 1932 the rate of increase in number of passengers was 19.7 per cent over the traffic carried in the same period of 1931, while passenger miles increased 20.8 per cent in the same period. Air mail fell off markedly (see Fig. 4), due almost entirely to the effect of the business depression, but the increase from 5 cents to 8 cents in the minimum rate also had its influence. The poundage of air mail carried in the first seven months of 1932 was 9.6 per cent less than that carried in the same period of the previous year. Air express packages showed a remarkable increase, the poundage carried in the first seven months of 1932 being 68.5 per cent in excess of poundage carried in the same period of the previous year. No change of any consequence took place in miles of route in operation of regularly scheduled services in 1932, but the airplane mileage flown in the first seven months of 1932 increased about 25 per cent over that flown in the same period of the previous year.

[3] Consulting Aeronautical Engineer, New York, N. Y.

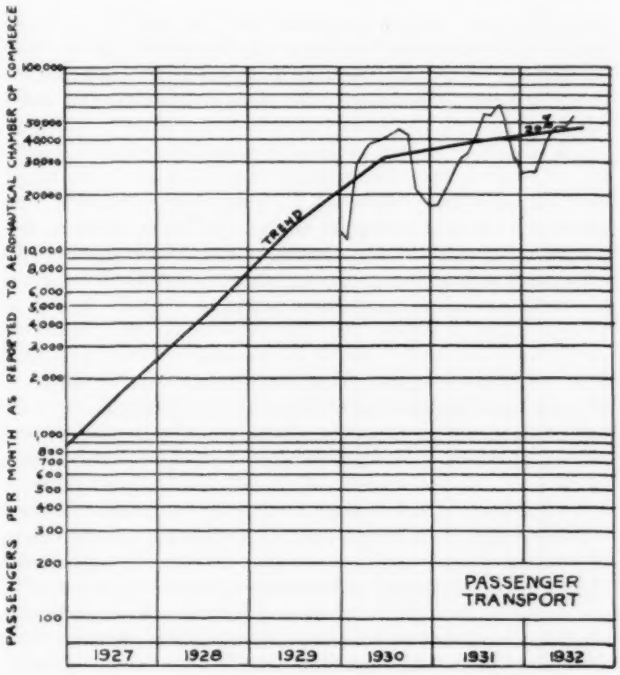


Fig. 3 Chart of Air-Passenger Transport

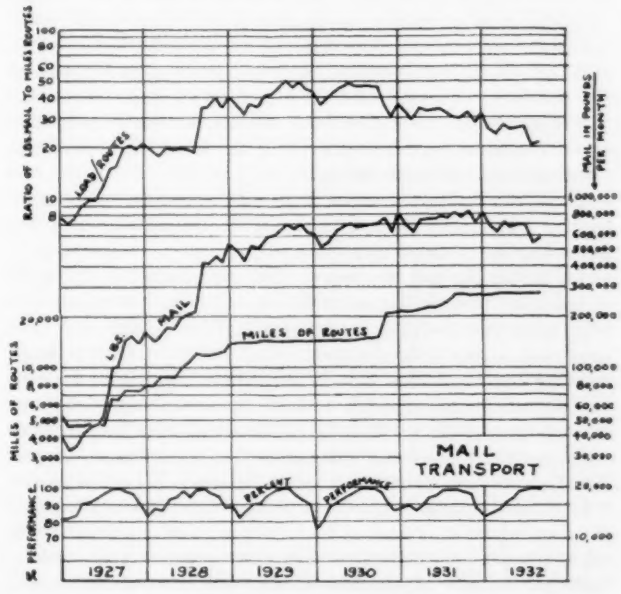


Fig. 4 Chart of Air-Mail Transport

AIRPORTS AND AIR NAVIGATION

By F. C. Hingsburg[4]

Very substantial progress has been made in the development of airports and landing fields since the passage of the Air Commerce Act of 1926. Records of the Aeronautics Branch of the Department of Commerce in Washington reveal that there are available at present in the United States 547 municipal airports, 619 commercial airports, 367 Department of Commerce intermediate landing fields, 64 Army and Navy flying fields, 41 mis-

[4] Aeronautics Branch, Department of Commerce, Washington, D. C.

cellaneous state and private fields, and 459 marked and listed auxiliary or emergency landing fields, a total of 2097. However, many of the airports listed are practically undeveloped in so far as accommodations other than a cleared landing space are concerned.

The principal trend of progress in connection with the operation of airports has been improving operations so as to provide for handling a large increase in the number of arrivals and departures without reduction in safety. At certain times of day a number of arrivals and departures must take place practically simultaneously, and a system must be worked out to handle the traffic at such times with the maximum dispatch as well as the maximum of safety. In this connection traffic control at periods of great traffic congestion is receiving the earnest attention of a large committee of airport and air transport experts. Control by radio communications and visual signals are both being carefully studied with a view to reaching as close as possible an approach to an ideal standard method, with due regard to the fact that variations from the general standard are likely to be required to meet special conditions in various localities.

Considerable attention is being given to passenger convenience. The prospective passenger now arrives at the airport passenger station, where signs and uniformed attendants guide him without confusion through the ticket and baggage preliminaries and invite him to await in a comfortable waiting room the arrival of his ship at a designated embarking gate or canopied gangway. Hangars, workshops, and warm-up lines are generally far removed from this point, and miscellaneous flying is segregated in another section of the airport. Very shortly before the departure time the ship taxies up to the embarkation point, shuts off its engine or engines, and uniformed attendants assist the passenger on board and show him to his seat. The same general procedure takes place upon disembarkation, and throughout it compares very favorably with the handling of traffic at a well-ordered railway terminal.

The system of intermediate fields established and maintained by the Aeronautics Branch of the U. S. Department of Commerce on the Federal airways has undergone considerable revision during the past two years. The increase in size and speed of transport aircraft has necessitated larger and smoother fields with improved facilities. Lighting equipment has also been improved, a recent development being the 36-in. double-beam-rotating beacon. This unit is constructed with two doublet-lens assemblies at opposite ends of the drum-shaped housing and equipped with a 110-volt, 1000-watt airways incandescent lamp, mounted in a lamp changer, with a spare lamp. It projects two beams of light separated by 180 deg. When the beacon is fitted with clear lenses, each beam is of 1,250,000 candlepower. In cases where it is desired to furnish a code characteristic in color, the beacon is operated with clear lenses at three revolutions per minute in synchronization with colored and coded course lights or code beacons.

The Federal airway weather reporting and communications system as described in the Transactions for 1931 has been very comprehensively improved in a number of ways. The number of primary radio broadcasting weather and communications stations has increased to 69. The teletypewriter communications system has been expanded to 13,000 miles with 245 stations. Experimentation with teletypewriter transmission developed the feasibility of transmitting facsimile weather maps to all points reached by this system. To provide the additional service at no increased total expenditure for the system, it was found necessary to purchase the machine component used on the long lines, instead of leasing it. This purchase included 135 page-type machines, making it possible to provide facsimile weather maps at 71 airports and important intermediate fields. Automatic transmission is accomplished by the use of perforated tape functioning in a manner similar to that of the perforated-paper records used in player pianos.

To the system of radio range beacons described in the A.S.M.E. Transactions for 1931 sufficient stations have been added to bring the total to 94. In addition, a few relocations have been necessary. The radio marker beacons which are installed at important points along airways between radio range beacons have been equipped for short-range voice transmission. A further substantial improvement in radio marker beacon installation comprises the installation of loop antennas with higher powered transmitters giving, in addition to the characteristic marker-signal and voice-communication service, radio range signals effective over a radius of from 25 to 50 miles.

There have been promulgated a number of important amendments to the Air Commerce Regulations and revisions in the interpretation of the Air Commerce Regulations governing scheduled operation of interstate passenger air-transport service. These were announced in the Air Commerce Bulletin of May 16, 1932. Of great importance in this connection was the establishment of the "Scheduled Air Transport" pilot rating and the requirement that transport pilots flying airplanes engaged in scheduled operation of interstate air-transport service under a certificate of authority must hold such a rating.

With the announcement of the requirements for the new scheduled air-transport license and the new scheduled air-line operation requirements, interest in the use of the radio range beacon in flying was immediately stimulated. Pilots of all air lines are practicing radio flying diligently in hooded cockpits in order to achieve the new Scheduled Air Transport rating.

AIRSHIPS

By C. P. Burgess[5]

No important new airships have been completed in the year under review, but quiet progress and development have continued. The U.S.S. *Akron* operated on the West Coast for over a month, with no base except an "expeditionary" mooring mast, a short mast of the simplest type with a minimum amount of equipment. On the west-bound flight across the continent, a severe storm and violent squalls were encountered in the mountain passes. The airship successfully won through against the buffeting of the wind squalls and the danger of being cast upon the mountain tops.

The elaborate ground-handling gear at Lakehurst, described in the Progress Report for 1931,[6] has been further developed, and has proved correct in principle, although its early operation was marred by an accident which put the *Akron* out of commission for over two months.

The U.S.S. *Macon*, sister ship to the *Akron*, will be completed early in 1933, and will be based on the new airship station at Sunnyvale, Calif., near San Francisco. This station will be the most perfect and up-to-date airship operating base in the world.

The *Graf Zeppelin* has carried out a regular and almost unnoticed program of round-trip flights between Friedrichshafen and Pernambuco. The schedule for the year calls for ten round trips, and at present, October 10, the ship is keeping up to schedule. The Zeppelin Company is making good progress on the construction of the new airship *LZ-129*, which will be slightly larger than the U.S.S. *Akron*. The ship will be inflated with helium, and propelled by four Diesel engines of 1000 hp each. Several different makes of engines are being considered, but no final decision or choice has yet been made. Loss of weight

[5] Bureau of Aeronautics, Navy Department, Washington, D. C.
[6] See *Mechanical Engineering*, vol. 53, no. 12.

through consumption of fuel in flight will be compensated by valving hydrogen from ballonets suspended inside the helium-gas cells.

AIRPLANE DESIGN AND CONSTRUCTION

By Richard M. Mock[7]

Due to general business conditions and overproduction in previous years, comparatively few new civil airplanes made their appearance in 1932. In recent years there have always been an appreciable number of new airplanes, often of quite progressive design, brought out by individuals and new independent companies. Evidently this decrease in the number of new airplanes of this class was due to difficulties in raising funds. The established manufacturers also produced very few new models for civil use, and many who had no previous military experience tried to develop service planes. It is believed that, due to this, the manufacturers, realizing the limitations of the present civil market and recalling their past over-production, as well as appreciating the experimental costs of a new type, have almost been overreluctant about appropriating funds for new development works and production. Most of the manufacturers have disposed of their previous large inventories.

It has been apparent for some time that in 1933 most of the airplanes built in 1929 or before would be due for replacement. In spite of the fact that this constitutes the greater part of the equipment now in service, very few manufacturers of transport or private planes have built experimental models to be fully developed and ready for limited production in 1933. In the non-transport market used-plane prices are so low as to discourage almost any manufacturer. However, it is felt that there soon will be a definite market for economical and modern non-transport airplanes. In the transport field it is felt that practically all of the planes built in 1929 or before will be replaced. Of course the expected replacement will not be 100 per cent, but it is reasonable to expect, especially in view of the recent increases in air traffic, that some expansion will take place which will also require new equipment. Many manufacturers were slow in realizing this and are now rushing to finish new models to be able to take care of the spring market.

The problems of commercial operators (regular scheduled flying) have been reflected in transport-plane design. More attention than ever was given to the detail design of parts that are apt to give trouble in service and for ease of accessibility and ease of replacement of various units. Airline operators have even gone so far as to draw up detail specifications for man-hours allowed for replacing certain units of the airplane. Though this has tended to reduce operation expenses, it has increased engineering costs, which apparently must be reflected in increased unit cost of the airplane. A great deal of attention on the part of commercial airplane designers has also been given to the reduction of time necessary at an airport for discharging and loading passengers and baggage, refueling, and replacing certain equipment such as batteries. As a result, baggage and accessory compartments are usually accessible from outside the airplane.

As a structural airplane material, duralumin seemed to be increasing in use, and there seemed to be a definite tendency to use it for fuselage construction as well as for movable control surfaces. Material for wing construction seemed to be divided as heretofore except for the tendency away from fabric covering because of the desirability of having a rigid cover on a cantilever wing. There seemed to be a definite preference for a flat dural skin over using it in the corrugated form. Though magnesium alloys were used quite extensively in Europe, both in sheet and in castings, they made little progress in the United States. In spite of the excellent pioneering of the Budd Company with the shot-welding process on stainless steel, it is still only slightly used except on fittings and vital parts which must be especially corrosion resistant. The Boeing "Totem," a monoplane flying boat, has all stainless-steel fittings.

Monocoque fuselage construction in duralumin seems to be increasing in popularity, and as the year comes to a close two new monocoque commercial designs have made their appearance. They are both 600-hp, low-wing, metal-construction transports; namely, the Northrup "Delta" and the plane produced by the American Airplane and Engine Corporation and taken over by the General Aviation Corporation.

In the design of landing gears there seem to be two distinct schools—one that believes that for aerodynamic efficiency the landing gear must be retracted and the other that believes that the retractable gear is an unnecessary extra complication, extra weight, and cost, and that a clean external gear is as efficient and offers less possibility of an accident. The two mentioned planes are examples of this. The former has a streamlined fixed external gear, while the latter retracts the wheel partly into the wing, leaving the lower portion to be covered by movable fairing, producing a bulge below the wing.

More attention than ever was paid to fuselage shape and wing-fuselage interference. Cantilever monoplanes are ever increasing in proportion to biplanes and externally braced monoplanes, and there seems to be a definite tendency to an increase in taper ratio. On single-engine designs the trend seems to be definitely toward the low wing, chiefly because of the ease of retracting the landing gear and the slightly reduced structural weight.

There seemed to be an ever-increasing interest in single-spar cantilever-wing construction, both of the fabric-covered type developed by Stieger in England and the stressed-skin type exploited in Germany. In Germany, Heinkel brought out a low-wing monoplane with a plywood-covered single-spar wing having slots and flaps giving a speed range of 1 to 4.

In general the use of ring cowls on radial engines became quite common both here and abroad. In Europe the Townend ring has been more popular, while here it has been adapted to most old designs, while most new planes have been designed to use a full over-all or N.A.C.A. hood cowl.

There have been a great number of efforts to reduce noise and vibration in airplane cabins. Rubber engine mounts are very common, and practically all cabins are lined with some sound-absorbing and insulating material such as kapok or dry-zero, while one organization uses a mixture of down and feathers. Double windows with an air space in between have also just come into use.

Most cabin planes have windows that cannot be opened by the passengers and have the heating and ventilation system controlled by the pilot or co-pilot. Individual air ducts that can be operated by each passenger, of the type developed by the Fokker Aircraft Corporation, are universally used. Most planes are heated by bringing air in contact with the exhaust manifold, though a Prestone heating system is being used on some new planes under development.

Exhaust collector rings are usual equipment on radial engines, and a few organizations have been experimenting with mufflers. Eastern Air Transport has developed an exhaust muffler that seems to work quite well. One of the new transport planes under development will have geared-down propeller drives to help reduce the propeller noise as well as increase the climb.

Controllable-pitch propellers became more common during the year, especially the mechanically controlled propeller hub of the Smith Engineering Corporation. These should account

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for increased efficiency of multi-engined airplanes with one engine disabled as well as improved climb and take-off on all types.

Aircraft operators have begun to realize that the ultra-fast airplane has the same advantages over the slow airplane as the slow airplane has over fast ground transportation and that a fast airplane can be operated economically. As a result the trend in new designs is for greater speeds, both through surplus in power and improved aerodynamic cleanliness. While a cruising speed of 110 mph was considered excellent a few years ago, at least 140 to 160 mph cruising is now necessary on a modern design, while there are some airplanes already in operation maintaining a schedule cruising of 175 mph.

In the transport field there is a tendency away from the large airplane which a few years ago seemed definitely to be on the horizon. A ten-place plane, operating frequently, seems to be the desire of many operators, though it is felt if they are in operation on an hourly schedule and air traffic increases as it has in the recent past, this size of plane will be too small before it has served its useful life, since it is unlikely to have schedules more frequent than one plane an hour.

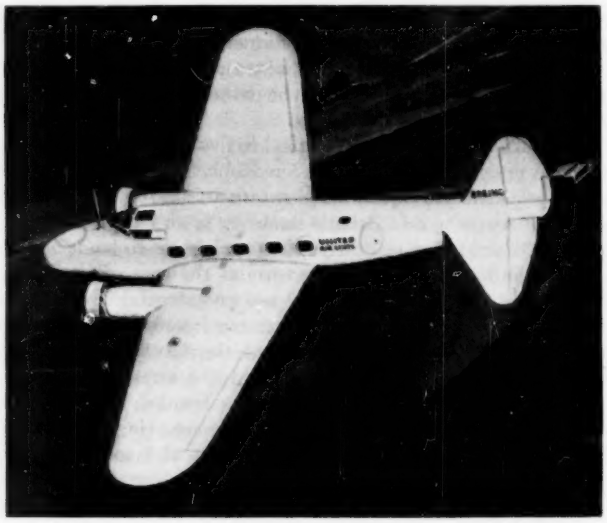


FIG. 5 APPEARANCE, AS PLANNED, OF BOEING 10-PASSENGER HIGH-SPEED TRANSPORT BEING BUILT FOR UNITED AIRLINES

There also seems to be a tendency toward the single-engine transport, since many operators find this type more economical and yet entirely satisfactory for daylight operations over flat country. However, it is believed that if single engines developing more than 700 hp were available, larger single-engine airplanes would be built.

As the year comes to a close a number of experimental single-engine, low-wing, monocoque, ten-place Cyclone- or Hornet-powered transports were being developed. Besides the General Aviation plane and the Northrop "Delta," Lockheed and Vultee in California are developing planes of this type. The Lockheed is expected to cruise at 180 mph and have a high speed of 220 mph with ten people and 500 lb of baggage. In addition there are a few planes of this size and type which are yet in the "secret" stage. Fleet Aircraft, Inc., is continuing production on an improved-model "Fleetster," with Hornet engine. It has a plywood-covered cantilever high wing and a dural monocoque fuselage.

Since landing speeds are not increasing in proportion to top speeds, speed ranges must be improving. As a result the problem of flying controls that are adequate at both maximum and minimum speeds is growing more acute.

Various devices for increasing wing lift have been tested by the N.A.C.A. and are slowly being used in actual construction. The Northrop "Delta" uses split flaps and has floating ailerons above the wing at the trailing edge.

The Boeing Company has developed a servo-rudder which is understood to reduce the load necessary to keep the Boeing tri-motor flying on a straight course with one side engine stopped from 90 lb to 12 lb.

Beech Aircraft has brought out a negative stagger cabin biplane with a split rudder that opens up and acts as an airbrake when both rudder pedals are pushed forward.

The only new multi-engine plane developed and put into production in the United States during 1932 was the Stinson model "U," a ten-passenger tri-motor similar in construction to the original Stinson tri-motor except for a roomier cabin and the addition of a short cantilever lower wing which supports the outboard engines and carries the struts for bracing the upper wing. The stub wing also carries the landing gear. Like the Bellanca Airbus, the stub wing has heat-treated welded-steel-tube spars and contains a compartment for mail or baggage. This load is near the center of gravity and is accessible without disturbing the passengers.

Another passenger transport completed during the year was the 40-passenger Ford mentioned in last year's report. However, as far as can be learned, though this plane was completed in the spring, it was never flown. The design and construction was exceptionally interesting, especially the clean outboard-engine installations, with the engines completely within the wing and driving a propeller mounted in front of the leading edge by means of a shaft.

Boeing, Curtiss-Wright, and Douglas are developing twin-engine retractable-landing-gear transports which, it is understood should be completed about the end of 1932 or early 1933. The Boeing design (see Fig. 5) will be a 10-passenger, low-wing monoplane which is calculated to cruise at 150 mph at sea level with 75 per cent power, and since the engines are supercharged to give sea-level performance at 5000 ft, the cruising speed should be about 155 mph at this altitude. The estimated figures indicate that with full load this plane will be able to maintain 1000 ft altitude with one engine dead and the other developing 75 per cent power. With one engine at full throttle the plane is expected to have a climb from 1000 ft of about 100 fpm.

The Curtiss-Wright transport, or "Condor" as it will be called, is understood to be a sesquiplane with two Cyclone engines located in the leading edge of the lower wing and will carry 15 people and a crew of two, besides 650 lb of mail or baggage. It is expected to cruise at 140 mph.

The Douglas transport will be a low-wing design, powered with two 700 hp Wright Cyclone engines. With 12 passengers and 1000 lb of mail, it is expected to cruise at 165 mph and have a range of 1000 miles.

There were somewhat more new planes produced in the non-transport class during the year. However, most of them were continued refinement of detail. Stinson produced the model "R," which is believed to be the first cabin sport plane with retractable landing gear. (See Fig. 6.) It is like the Stinson "Junior" of previous years, except for the addition of a stub wing for the retractable landing gear and the somewhat roomier fuselage. Besides increasing the width and depth of the fuselage, it was shortened.

Few outstanding features of design in the non-transport class were brought out. One of the commonest refinements was the addition to many of the open-cockpit planes of a demountable transparent cockpit cover for winter flying.

In the flying-boat-amphibion field there were two interesting attempts to reduce the excessive cost of a hull of conventional

construction for a sport plane. Amphibions, Inc., produced a three-place cabin model with a hull having a welded-steel-tube frame with a dural bottom and having the sides and deck covered with waterproof duck. Curtiss-Wright produced an amphibion version of the "Junior" with a special hull of plywood covered with waterproof fabric attached by a gluing process. The hull construction was developed in small-boat manufacture.

Both the plane produced by Amphibions, Inc., and the "F-15" Coast Guard patrol boat built by General Aviation showed the tendency toward having a short hull with a single boom supporting the tail. The "F-15" has an interesting retractable beaching gear that is a permanent unit of the airplane. The wing is of wood, with a stressed birch-plywood covering. Thus the wing adds to the buoyancy of the plane in an emergency. In Germany a novel tailess amphibion design was exhibited with two Diesel engines in the twin hulls driving a single propeller above the wing through a driveshaft.

Among the special planes built for the National Air Races were a number of interesting and efficient designs. Outstanding among these was the "Gee Bee," which broke the world landplane speed record. The short fuselage and extremely small tail surfaces and pilot's cockpit are worthy of notice. Unlike most of the wire-braced low-wing designs, the "Gee Bee" had its wing well above the bottom of the fuselage, tending to eliminate the usual pocket above the rear portion of the wing where it meets the fuselage. Most of the other racers were low-wing, wire-braced designs of about the same proportions as the British racing Moth of 1927. An interesting design feature on one, the Howard racer was the faired tandem-wheeled landing gear, which resembles that on the German G-38, but on a much smaller scale.



FIG. 6 STINSON MODEL "R" FOUR-PLACE CABIN PLANE, LYCOMING ENGINE, WITH RETRACTABLE LANDING GEAR

An exceptionally interesting plane in the cross-country race was built by Claire Vance of California. It consists of a plywood-covered tapered cantilever wing with an engine nacelle in the center. The nacelle is carried back to the trailing edge of the wing, with the pilot's cockpit directly behind the engine. The fully cowled engine is in front of the leading edge of the wing. The tail is supported on two booms. Landing gear is of the external, conventional three-strut type and includes two tail wheels.

In the realm of rotating-wing aircraft there was much activity during the year. The Wilford gyroplane with rigid cantilever blades, described in last year's Progress Report, has been successfully flying for some time. A type of rotor control was developed eliminating the need for ailerons and elevators for lateral and longitudinal control. Since the blades are cantilever, it has been proved feasible to fly on the lower wing alone and let the rotating blades idle without any load, and thus reduce the overall resistance for high-speed flight.

It is understood that in England a type of rotor control has been worked out for the autogiro which consists of a control stick which is really a continuation of the rotor mast down into the cockpit. By rocking the stick from side to side or fore and aft, the masthead, to which the blades are hinged, also moves, producing lateral or longitudinal control.

Both in England and in the United States cabin autogiros were produced. The American machine, built by Pitcairn, is for four people and besides being a cabin type differs from the conventional autogiro in that the lower wing with upturned tips has been replaced by a highly tapered cantilever wing with large dihedral. Two fins and two rudders are used.

AIRCRAFT-ENGINE DEVELOPMENTS

By C. F. Taylor[8]

In reviewing aircraft engine progress for 1932, it seems pertinent to mention some of the more important research work in the general field of internal-combustion engines, on account of its possible future application in the field of aircraft power plants.

Perhaps the outstanding piece of applied research has been the study of the combustion process in the compression-ignition engine, with particular reference to the effect of fuel composition on the "delay period" and on the combustion rate. (See ref. 1 to 5.) The work of Boerlage and Broeze in Holland and the photographic studies of combustion by the National Advisory Committee for Aeronautics are considered particularly significant. This work should have important results in practical application, particularly with regard to control of fuel composition to make possible increased engine output for high-speed compression-ignition engines, with perhaps eventual application to aeronautic work.

The excellent work with regard to the measurement of detonation which has been carried on for some years by the Cooperative Fuel Research Committee has been continued during 1932. Significant results have been obtained relative to the coordination of laboratory tests with actual service conditions. (See ref. 6.)

Research on cycles involving fuel injection with electric ignition has been actively continued during 1932 (ref. 7), and this, together with the development of hydrogenated "safety" fuels, may eventually result in a significant improvement in fire protection for aircraft.

Other significant research that might be mentioned includes work on journal-bearing friction (ref. 8), engine balance (ref. 9), and ring cowling (ref. 10), all of which may have important effects on the design of aircraft power plants.

Under the heading of application, mention should be made of the fact that one of the large air-transport lines has recently put into experimental service an engine using fuel injection into the cylinders, combined with spark ignition. Results to date have been so satisfactory that more extensive application of this type of engine is quite likely.

Engines of the 14-cylinder double-row-radial type, of American manufacture, were put into experimental operation during the year. At least two manufacturers are experimenting with this type of engine.

With regard to conventional engine types, the year was marked by significant increases in the ratio of power output to weight and to piston displacement, especially in the case of air-cooled radials. This was largely accomplished by the use of higher compression ratio, anti-detonating fuels, supercharging, and higher rotative speeds. The progress in this direction was well illustrated by performance records made at the 1932 Cleveland Air Races (ref. 11). As a result of such developments, the power

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and speed rating of most of the more important American air-cooled radial engines has been markedly increased during the year.

Another contributing factor toward the higher power output of the radial engine has been progress in metallurgical technique, particularly with regard to aluminum forgings, notably for pistons, and magnesium castings. There has been a marked increase in the use of such materials for aircraft engines during the year.

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AERODYNAMICS

By Elton W. Miller[9]

The most obvious developments in aeronautics during the last year have been the marked increase in the cruising speeds of military and commercial airplanes and the noticeable increase in the safety of flight on air lines. In these developments center the aerodynamic advances of the year. Higher speed is made possible, among other things, by a better understanding of fluid flow about bodies and of the factors affecting the mutual interference and drag of airplane parts. Higher speeds in turn call for a better understanding of structural loading and more rational structural design methods. Considerations of safety in flying require better controllability at low speeds, lower landing speeds, and a solution of the spinning problem. In each of these fields there has been definite progress during the year. Airships and rotating-wing aircraft also have come in for their share of attention.

Interference and Drag. The investigation by Wood of the N.A.C.A. of the relative efficiencies of various wing-nacelle-propeller arrangements has been continued throughout the year, with the result that the designer need no longer be in doubt as to the best position for the nacelle in any particular design, or if practical considerations dictate an arrangement other than the best, he may know how much he is paying for the compromise. This investigation has included variously cowled nacelles for tractor propellers in combination with monoplane wings and biplanes. Tandem propellers and pusher propellers have also been tested in combination with monoplane wings.

The mutual interference of simple bodies has received careful attention by the British; the work of Jacobs, of the N.A.C.A., on the effects of small protuberances on streamline bodies and on airfoils, should prove of assistance in the general "cleaning up" of aircraft. The mutual effects of wings and bodies have been exhaustively studied by Parkin, of the University of Toronto, within the limits of scale of his equipment, and an important contribution in the same direction has been that of Klein, of the California Institute of Technology, on the effects of fillets on wing-fuselage interference.

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Cowlings of the N.A.C.A. or of the ring type have been quite generally applied by designers to the air-cooled engine, but more complete information is required to enable the engineer to design the most efficient cowling for each particular case. Whether landing gears should be retracted or merely simplified and made clean seems to be still an open question.

The study of drag and interference is closely related to that of fluid flow about bodies. Deserving of special mention are the studies of flow about bodies of revolution by Millikan and Dryden and about airfoils by Theodorsen. The work of the British and of the N.A.C.A. dealing with flow studies by direct photographic methods should also be mentioned.

Load Factors. The aerodynamic loads actually experienced by military and naval airplanes in maneuvers and by transport airplanes when passing through gusts or vertical currents have been a matter of increasing interest in this country and abroad. The work of Rhode, of the National Advisory Committee for Aeronautics, on the pressure distribution over wings and tail surfaces offers valuable material on which to base the rational determination of load factors. In the same connection a new instrument has been developed by the N.A.C.A. during the year to record acceleration and air speed over a period of time. A number of these instruments are now in use on military and commercial airplanes, and the data being accumulated will doubtless be of great value.

Control at Low Speed. Many accidents in the past point to the need for more effective control at low flying speeds. This need has been appreciated for a long time. The Handley Page tip slot has been widely applied in Great Britain as the answer to this question, but even the British appear more recently to feel that there are possibilities of adequate control by other means. In this country an aversion to complication and to the addition of movable parts may possibly explain the failure to adopt the Handley Page slot.

The N.A.C.A. has attacked the problem by investigating thoroughly and systematically the properties of all existing and some new control devices, in combination with rectangular and tapered wings, and wings with special forms of tip. Some of these combinations seem very promising when properly applied. The results of a number of wind-tunnel tests have been published during the year by Weick and his assistants, and the more promising devices are being tested in flight.

Landing Speeds. As cruising speeds have increased, it has been necessary to investigate means for reducing landing speeds in the interest of general safety. Many devices have been offered for increasing the speed range, such as the Hall high-lift wing, the Zapp flap, and the Fowler variable-area wing. The N.A.C.A. has conducted research throughout the year to the same end, and one of the more promising devices resulting from this investigation is a small fixed auxiliary airfoil ahead of the main wing. Flight tests as well as wind-tunnel tests have been conducted, and the investigation is still in progress. Some of these devices, particularly the latter, have the further advantage of increasing the gliding angle, thus reducing the space necessary for landing after clearing an obstacle. In the same connection, wings of especially low aspect ratio and tailless airplanes appear to offer some advantages.

Spin. Probably no problem of aeronautics has been attended with so much mystery or has received such a large share of attention in all parts of the world as the problem of spinning. Research has been conducted and is still in progress in various laboratories, and there have been contributions by numerous commercial engineers, as by Edmund T. Allen and J. M. Gwinn, Jr. In England a 12-ft vertical tunnel has been constructed and is now in operation, in which models of airplanes may be made to spin and the effects of the controls noted. The N.A.C.A.

now has in operation a vertical tunnel with a special balance on which the moments of various control surfaces may be measured while the model is in a spinning attitude. An investigation in flight has also been continued throughout the year, including the use of smoke to indicate the nature of the flow over the control surfaces.

The solution of the inadvertent spin appears to lie merely in better control and stability at low speed, and a great improvement in control seems now within reach. The control of the voluntary spin appears at present to lie in such design and arrangement of the tail surfaces as will give adequate yawing moments under all circumstances.

Airships. The airship program of the Navy Department has given rise to considerable research work on airships during the past year. Most of this work has been centered around better control of the airship and a better understanding of air flow over the hull. Investigations by Seiferth, of the California Institute of Technology, and Freeman, of the N.A.C.A., on models of the *Akron* have contributed valuable information on these problems, including data of great interest on the boundary layer over the hull. Flight acceptance tests of the *Akron*, in which the Goodyear Zeppelin Corporation and the N.A.C.A. cooperated, have also yielded valuable data.

Rotating-Wing Aircraft. The autogiro offers many attractive features for certain purposes. A number of improvements are being developed, including the change from an open cockpit to a cabin and the elimination of cables on the rotor. The N.A.C.A. is conducting an investigation of the autogiro and has published some of the results during the year. The development of the gyroplane, which differs from the autogiro in that the blades are made to feather instead of flap, indicates another possibility in this type of aircraft.

New Equipment. Of interest in connection with new equipment for aeronautical research, is the dedication during the year of the Guggenheim Airship Institute in Akron. In England, the new compressed-air wind tunnel and a new seaplane channel have been put in operation, and the new 24-ft wind tunnel is understood to be under construction. The newer items of equipment in this country, including the N.A.C.A. tank and the N.A.C.A. full-scale wind tunnel, have been in operation throughout the year, and all have turned out valuable results looking toward more efficient and safer aircraft for the future.

AER-55-2

Lightning Investigation as Applied to the Airplane

By A. O. AUSTIN,[1] BARBERTON, OHIO

The hazard to aircraft from lightning is very small, but has received much attention. Lightning itself has caused but few fatalities, and many cases of trouble have been unjustly charged to it. Once the effect of lightning upon the pilot or plane is recognized, steps can be taken to materially reduce the hazard, if not to eliminate it. Some interesting information has been gathered from tests as to protection for free balloons and planes and other equipment.



WHILE the hazard to airplanes or to any aircraft is very small as compared to many everyday risks, this hazard receives much attention. Although lightning has caused a few fatalities, it would seem that many cases of trouble due to other causes have been unjustly charged to lightning. Once the effect of lightning upon the pilot or plane is recognized, steps can be taken to materially reduce if not eliminate the hazard, although this may be small.

The studies made in the high-voltage laboratory of the Ohio Insulator Company for Ward T. Van Orman in testing out his protection for free balloonists and the series of tests run on planes and other equipment provide some rather interesting information.

Since the tests run in connection with free balloons illustrate certain phases of the subject to better advantage than those upon airplanes, this matter will be treated first.

In running the series of tests, balloon baskets, small-size balloons, and model airplanes, as well as full-size airplanes, were placed between a large electrostatic condenser and the ground on the test field. The condenser may be regarded as a charged cloud, and a discharge taking place between this cloud and the ground can be made to strike the equipment under test. Fig. 1 shows a discharge taking place from this condenser to the ground, the photograph being taken by a synchronized camera.

In general the hazards from lightning may be divided into two classes:

(*a*) Electrical or physical shock that may affect the pilot and passengers

(*b*) Those hazards that damage the plane or aircraft.

It would seem that the hazard due to the direct electrical or physiological effect of lightning upon the pilot or passengers should receive first consideration, rather than the damage to the aircraft. Unfortunately, it is very difficult to obtain information upon this point owing to the variation in the personal element itself and the hazard of making studies that may be exceedingly dangerous. It may be said, however, that as the size of aircraft increases, the direct danger to pilot or passengers tends to decrease, providing ordinary precautions are taken in the design and construction of the aircraft. This is due to the greater distance of the crater or point of contact of the lightning from pilot and passengers.

Hazards to the Plane or Aircraft

These hazards may be classified as follows:

1 Fire hazard due to ignition of combustible material used in the construction of the plane
2 Fire hazard due to ignition of explosive gases
3 Weakening or destruction of metal or other parts due to current in the discharge
4 Breakdown of insulation in the ignition system
5 Backfire or preignition
6 Damage to instruments
7 Damage to rotating parts due to passage of current
8 Sudden change in pressure on adjacent surfaces.

Electrical or Physiological Shock to Pilot or Passengers

The experience of free balloonists throws considerable light upon the hazard due to shock, and upon the means of protection. The fatalities in Belgium and in Pittsburgh of recent years caused Van Orman to give this matter considerable attention.

In the Pittsburgh race Van Orman's balloon was struck by lightning while at an altitude of 3000 ft and was set on fire, and Morton, who was with him in the basket, was killed by the stroke. The No. 26 R.C.A. portable loop radio set, which was between Van Orman and Morton, appeared to be badly damaged. However, an investigation showed that the set was not damaged electrically. Van Orman apparently was conscious for a short time after the stroke. He then lost consciousness and apparently remained in a stunned or dazed condition for five or six hours after the parachuting balloon struck ground.

Wollam and Cooper, in the balloon *City of Cleveland*, had a somewhat similar experience, the stroke of lightning apparently passing over the surface of Cooper's leather jacket or suit, which was wet at the time. The shunt path provided by the suit apparently saved Cooper's life, although he was badly burned by the stroke. Wollam escaped uninjured, and he attempted to lift Cooper and his parachute out of the basket, but was unable to do so. Cooper apparently is the only known person to have suffered a heavy direct stroke and live to tell about it.

These incidents show that a stroke of lightning may be very close to a person without causing any serious injury. Had the pilots been subject to the same conditions while flying an airplane, it is possible that they might have lost control of the plane, causing it to crash.

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Presented at the Fifth National Aeronautic Meeting, Baltimore, Md., May 12 to 14, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Fig. 2 shows the basket used on the *City of Cleveland* balloon, which was struck while the balloon was at least 5000 ft in the air.

The photograph shows the basket with a dummy in place being subjected to an artificial-lightning discharge. The steel cables from the ring to the basket were not bonded to the reinforcing wires that were used in making up the basket. It is possible that had bonding been carried out, the results might have been quite different.

In the test shown in Fig. 2, the collapse of the electrostatic field caused an induced potential of over 25,000 volts between the dummy and the lower part of the basket. While the energy is not large, the discharge might tend to startle one in a manner similar to the shock one receives in coming in contact with a metal object after picking up a charge in walking over a carpet in cold dry weather.

Fig. 3 shows the metal reinforcing wire fused, due to the discharge of lightning. It will be noticed that the horizontal wire crossing the vertical wire has been burned in two, as well as some of the basket material. An examination of the basket showed a large number of burns where the reinforcing wires crossed each other.

Fig. 4 shows a test on Van Orman's lightning protecting scheme. The protection consists essentially of a cage formed by conductors hanging freely. These conductors are bonded at the top and part way down, but are left entirely free at the bottom so as not to interfere with landing or parachute jumps from the basket. The conductors forming the shunt path are moved outward from the basket a short distance. The cable supporting the basket to the ring has been replaced with rope. It would seem that had this device been used in the flights in Belgium and in Pittsburgh, the fatalities due to direct stroke would have been avoided.



FIG. 1 DISCHARGE FROM LARGE CONDENSER TO GROUND AT THE HIGH-VOLTAGE LABORATORY OF THE OHIO INSULATOR CO.
(Photograph taken by synchronized camera, making it possible to photograph discharge in full daylight.)



FIG. 2 BASKET USED ON "CITY OF CLEVELAND," WITH DUMMY IN PLACE BEING SUBJECTED TO ARTIFICIAL LIGHTNING
(Klyndonograph showed 25,000 volts between dummy and basket.)

Apparently there is little difficulty in shunting the main discharge, but there is some question as to how far the discharge should be from the pilot so that he will not be stunned or frightened to the extent of losing control. The baskets in racing balloons are rather small, and any one in the basket must of necessity be very close to any discharge striking the basket. Should the Van Orman cage give the necessary protection for direct hits, it would go to show that the protection to pilot or passengers in an airplane may be easily provided by shunting the discharge a short distance to one side.

DANGER OF DIRECT HIT

Many people believe that a balloon or airplane could not be subject to a lightning stroke while flying, since there is no direct ground connection. Any object, however, that has a greater conductivity than the air or a greater electrostatic-flux carrying capacity will tend to disturb the electrostatic field and will cause a discharge in the immediate vicinity to take a path along the object.

EFFECT OF CONSTRUCTION UPON A STROKE OF LIGHTNING

In running the tests on the Van Orman cage, an attempt was made to protect the balloon. If the gas in the balloon is free from air, lightning will not cause it to explode, although the balloon may be set on fire. If an explosive mixture is present in the balloon, an explosion may take place due to the ignition of the mixture.

Difference Between a Wet and a Dry Balloon

In the study of high-voltage phenomena, particularly those applicable to transmission-line structures using wood, it was believed that there would be a considerable difference between a wet and a dry balloon. The tests were very interesting in this

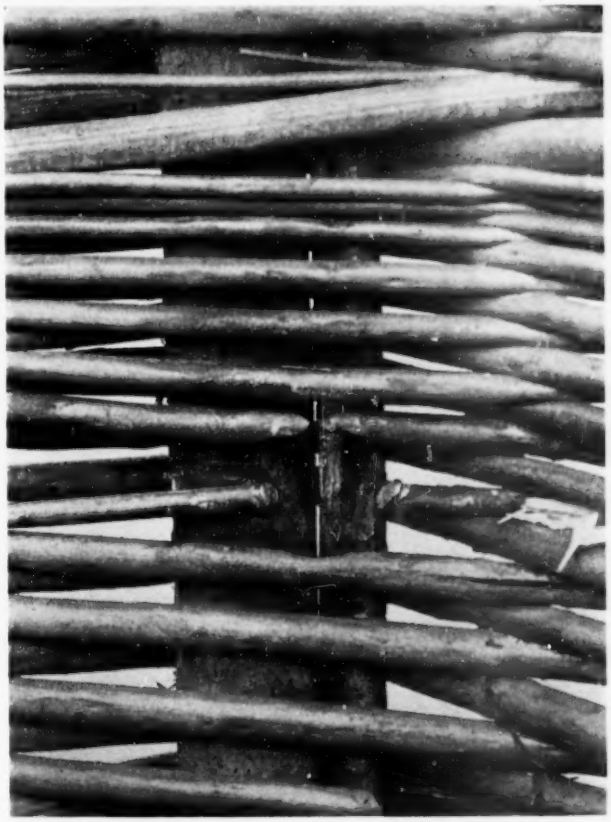


Fig. 3 View of "City of Cleveland" Basket Showing Reinforcing Wire Burned in Two Due to Stroke of Lightning

connection, as they go to show that while there might be a considerable difference in an aircraft that is constructed of metal and one that is primarily of non-conducting material when dry, there would be little difference where the latter was wet. This difference is undoubtedly due to the increase in electrostatic capacity due to the presence of water.

Fig. 5 shows a hydrogen-filled balloon being subjected to an artificial stroke of lightning. The balloon has a conductor attached approximating a long antenna, hanging from the lower side. A number of discharges were applied to the balloon similar to that in Fig. 5. All of these discharges struck the upper part of the conductor attached to the balloon, and then passed from the lower end of this conductor to ground.

As soon as the balloon was wet, however, conditions were entirely changed, the performance being similar to that shown in Fig. 6. The discharges, instead of going directly to the antenna and then to ground, struck the surface of the wet balloon. The discharge in Fig. 6 is apparently passing along the surface, which destroyed the balloon in all cases. The discharge invariably burned the rubber and set the escaping gas on fire. Had the photograph been taken a fraction of a second later, the balloon would have been shown collapsed, the duration of the arc shown in the photograph being less than one-half microsecond.

These tests go to show that there is little or nothing to be gained electrically by the use of non-conducting material in the aircraft structure where the material becomes wet, as the electrical fields set up are not materially different.

A discharge striking conducting material adjacent to inflammable material is likely to set the latter on fire. There are many cases that illustrate this point. Owing to the tendency of a discharge to strike a wet balloon, it was evident that it was necessary to remove the point of contact of any conducting system some distance from the balloon. While this is a distinct disadvantage to racing balloons owing to increased weight, the tests are very interesting as they show the tendency of the wet fabric in diverting or attracting a discharge.

Fig. 7 shows a discharge to ground some few feet distant from a balloon equipped with a cage or lightning rods. In Fig. 8 the balloon has been wet. While there was some tendency for the balloon while dry to attract the arc, the starting point of the discharge was too great, so that the arc struck to ground. In Fig. 8, however, the field set up by the wet balloon was of sufficient magnitude to attract the discharge. The discharge followed the shunt path provided by the conductors of the cage, and then continued from the lower end of the cage to ground. Without the shunt path, the wet balloons were invariably set on fire by the direct stroke. In some cases the fabric seemed to be only slightly damaged. The escaping gas, however, was frequently ignited by the stroke, causing destruction of the balloon unless quickly extinguished.

Probability of a Direct Hit

In the case of free balloons or blimps there would undoubtedly be a considerable difference between the wet and dry conditions



Fig. 4 Basket Equipped With Van Orman Cage for Protection Against Lightning

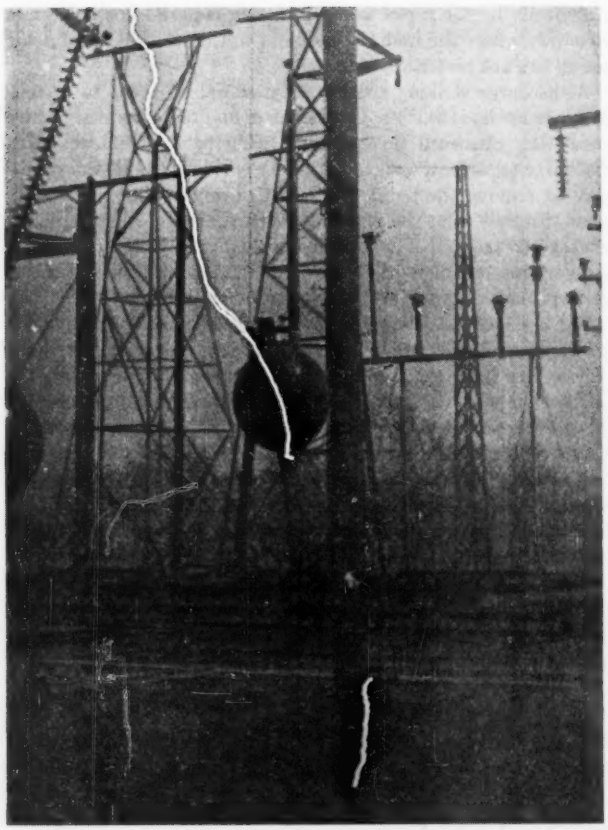


FIG. 5 DRY HYDROGEN BALLOON WITH ANTENNA ATTACHED, SUBJECTED TO LIGHTNING DISCHARGE WITHOUT DAMAGE TO BALLOON
(Discharge passes around the dry balloon to the antenna and then jumps from the end of the antenna to the ground.)

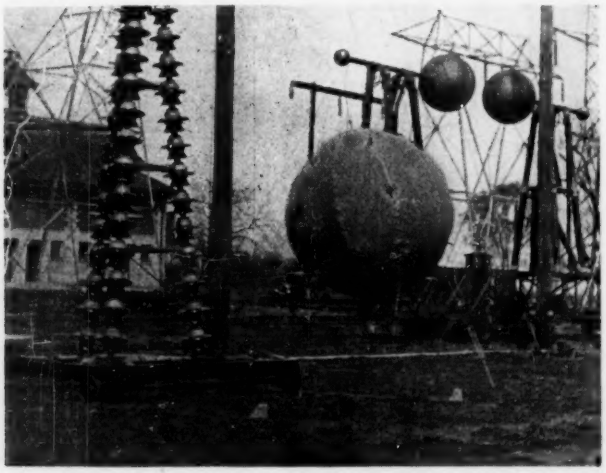


FIG. 7 DISCHARGE TAKING PLACE TO ONE SIDE OF DRY BALLOON EQUIPPED WITH CAGE AND LIGHTNING RODS

in attracting a discharge in the vicinity. In other aircraft as now constructed, however, there would be little difference wet or dry. In the tests on models placed near the path of discharge, it was evident that the path of discharge was diverted an appreciable distance by the presence of the plane. It would seem that the probability of being struck would increase approximately as the square of the greatest linear dimension.

The nature of the electrical field, the polarity of the discharge, and the direction of the axis of the aircraft relative to the general path of discharge are all factors that make it difficult to predict the effect of size in increasing the probability of direct hits. Figs. 9, 10, 11, and 12 show typical discharges to model planes, showing the probable points of contact of the stroke.

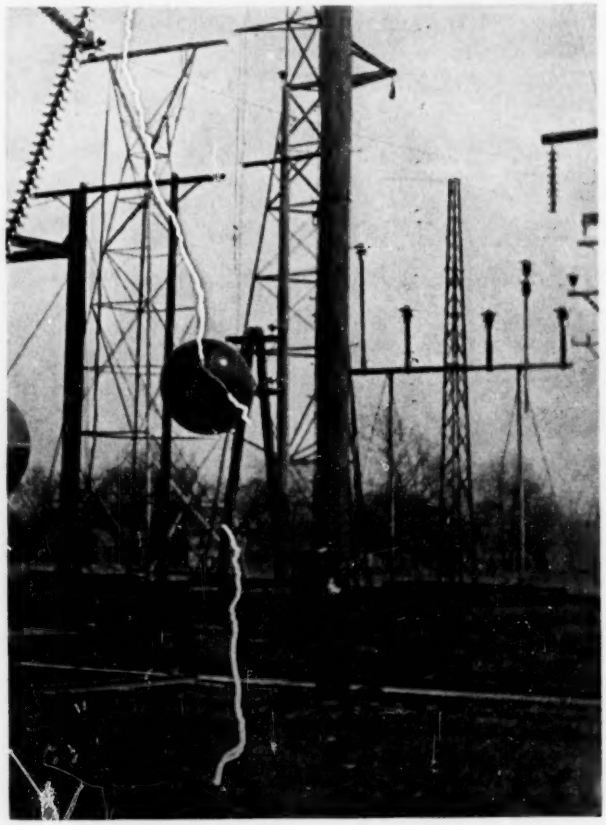


FIG. 6 PHOTOGRAPH SHOWS DISCHARGE TO THE WET BALLOON, CAUSING ITS DESTRUCTION
(The wetting of the surface caused the discharge to go directly to the balloon, burning a path along the surface and setting it on fire.)

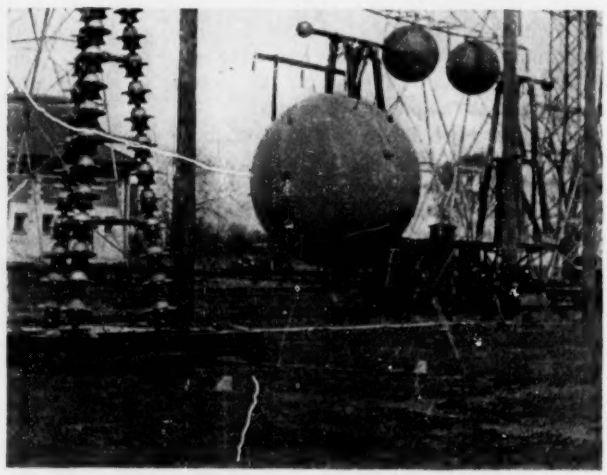


FIG. 8 THE FIELD SET UP BY THE WET BALLOON ATTRACTED THE DISCHARGE WHICH TERMINATED ON LIGHTNING ROD RAISED FROM THE SURFACE
(The effective area of the electrostatic field produced by an object is an important factor in attracting a positive stroke.)



FIG. 9 DISCHARGE ENTERING NOSE OF PLANE AND LEAVING BY TAIL STRUT

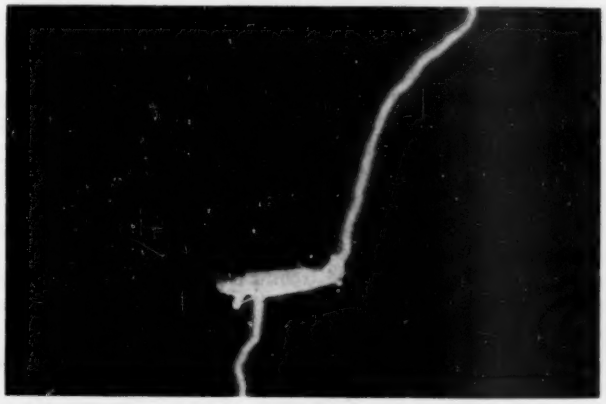


FIG. 11 DISCHARGE ENTERING RUDDER AND LEAVING FROM LANDING GEAR



FIG. 12 (LOWER LEFT-HAND CORNER) DISCHARGE ENTERING AT NOSE AND LEAVING FROM TAIL OF PLANE

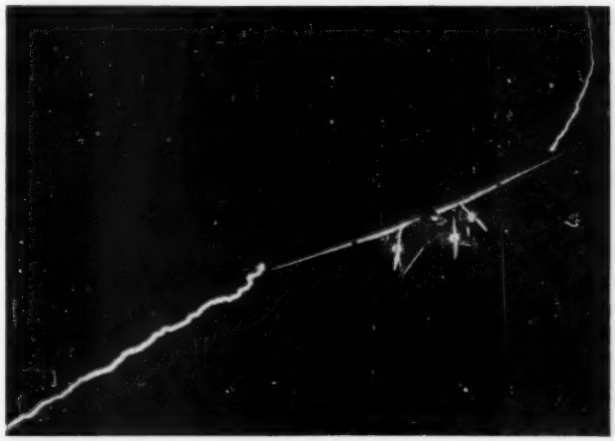


FIG. 10 DISCHARGE ENTERING AND LEAVING FROM WING TIPS

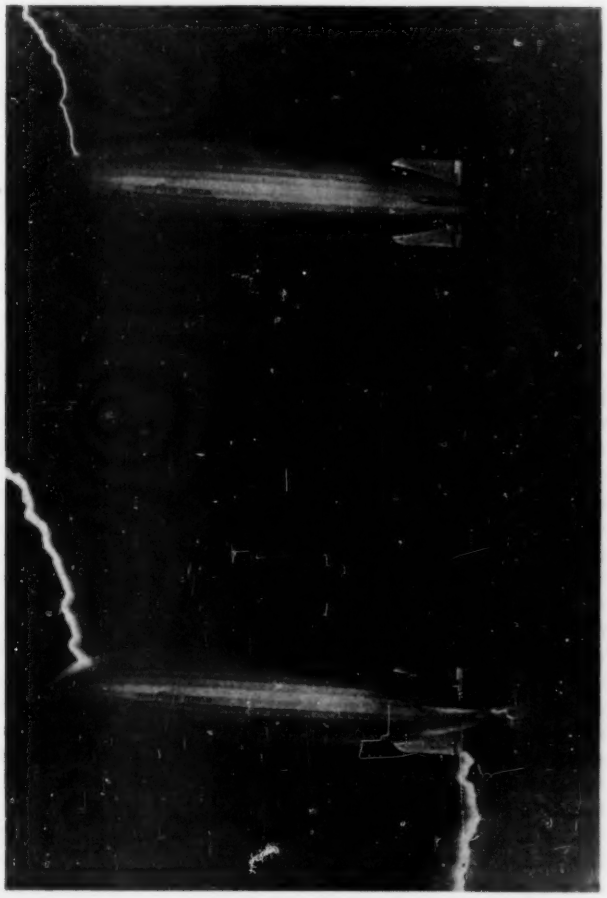


FIG. 13 (UPPER VIEW) DISCHARGE OF LIMITED CAPACITY STRIKING MODEL ZEPPELIN. (LOWER VIEW) HEAVY DISCHARGE STRIKING ZEPPELIN AND CONTINUING TO GROUND

Fig. 13 shows a positive discharge of limited capacity striking a model zeppelin. The discharge was not sufficient to cause the arc to continue to ground, but illustrates the effect of a large body free of ground. Immediately following the first discharge, another discharge was applied of sufficient magnitude to cause a discharge not only to the model zeppelin but from the rudder to ground.

It is interesting to note that in all of the tests the fabric used on the zeppelin and that on the fabric-covered duralumin airplane were not ignited by the discharges. This goes to show that the fire hazard is negligible where a path of high electrical conductivity is provided.

It would seem that the effective increase in size and the use of metal in the present construction of airplanes should do much to minimize the lightning hazard, even though little or no attention is given to protection.

A direct hit to the plane may possibly affect the pilot or passengers in one of the following ways:

(*a*) Direct hit
(*b*) By forming a path for the discharge between conducting objects
(*c*) Shock from induced charge
(*d*) Sudden change in air pressure
(*e*) Severe sound or pressure waves
(*f*) Currents induced in the body by an electromagnetic field
(*g*) Hazard due to the effect of intense light upon the pilot.

While the danger from some of these hazards may be absent in many planes, they can be largely reduced if not eliminated in others by proper attention to details of construction or by applying a protecting scheme that will establish the path of discharge at a distance from the pilot.

(*a*) *Direct Hit.* In general the possibility of a direct hit to the pilot is exceedingly small, even in the low-wing monoplane with open cockpit. The tests showed that the discharges would enter or leave through the propeller or nose, rudder, wing tips, or landing gear. A lightning rod projecting above and to one side of the pilot would insure the diverting of the stroke, even though the pilot's head projected well above the fuselage. In large planes or zeppelins, the points of contact of the stroke would be considerably removed from the pilot or passengers, so that it would appear that the danger from direct stroke may be even less than in the ordinary dwelling during an electrical storm.

(*b*) *By Forming a Path for the Discharge Between Conducting Objects.* A discharge of lightning may carry a current far in excess of that available in any of the laboratories used for the production of artificial lightning. The fused wire in the basket shown in Fig. 2 cannot be duplicated with the heaviest lightning discharges in the laboratory. Records taken at the forest ranger's stations show that a stroke of lightning may be sufficient to fuse a No. 14 copper wire. Tests on aerials have shown that wire of larger size has been fused, all of which indicates currents exceeding several hundred thousand amperes.

The impedance afforded by conductors with this very high rate of discharge will cause a considerable drop in potential. This drop in potential causes the current to divide into multiple paths, the drop in voltage being sufficient to cause the bridging of appreciable air gaps between parallel circuits, even where the difference in the impedance of the metallic paths is not very large. It is therefore essential that the most careful attention be given to bonding between parts to eliminate discharge. In order to lower the impedance (so as to reduce the difference in potential tending to cause discharge) it is well to distribute the conductor in several parallel paths. This reduces the reactance and the voltage drop along the conductor caused by the high rate of discharge.

It is evident that should the pilot come in contact at two points along a conducting member, he is likely to be subject to shock. The thorough bonding, the use of low-impedance multiple paths as far away from the pilot as possible, together with a single point of contact with conducting material, will eliminate the danger of shock from drop in potential due to the passage of exceedingly large currents.

(*c*) *Shock From Induced Charge.* A pilot in an open cockpit may be subjected to an induced charge due to the collapse of the electrostatic field. It would seem that this charge would amount to but little, provided the discharge did not cause the pilot to become frightened so as to lose control. The complete shielding afforded by metal cabin planes completely eliminates the effect of the induced charge due to the collapse of the electrostatic field. The same applies to zeppelins. Conducting parts separated by insulation may discharge across the insulation unless the distance is sufficient to prevent breakdown of the air. Where the area or capacitance of the parts is small, the discharge may be negligible, but as the area or capacitance increases, the discharges may be such as to require the bonding of any isolated metal parts to prevent discharge due to the collapse of the electrostatic field, which may have an appreciable effect although the lightning is some distance from the plane.

(*d*) *Sudden Change in Air Pressure.* The intensity of lightning varies greatly for different strokes. It would follow, however, that the very severe discharge causes a rapid heating of the air and an increase in pressure in the immediate vicinity. The pressure set up where the air is free to expand is not as great as originally supposed. The fact that three balloonists have come through storms where the stroke was within a foot or two of them would indicate that the hazard from this source is not serious, even for the very heaviest discharges where the path can be moved out a few feet from the pilot.

(*e*) *Severe Sound or Pressure Waves.* The effect of the severe sound and pressure waves may be responsible for the shock suffered by pilots and others who have been within a few feet of lightning strokes. The duration of the effect seems to vary considerably with the individual and may cause effects somewhat similar to those suffered from shell shock. Many of the factors producing shell shock are present, although there are others in addition. Fortunately the severity reduces rapidly with distance. It would therefore seem that using a construction that removes the point of contact between lightning and plane well away from the pilot will do much to eliminate any serious effects that may cause the pilot to lose even temporary control. The pilot can, of course, be easily protected from sound or pressure waves by providing a suitable compartment or other sound-absorbing equipment.

(*f*) *Currents Induced in the Body by an Electromagnetic Field.* A stroke of lightning may consist of a single discharge or of several discharges within a very short space of time. The strength of the electromagnetic field will vary directly as the current in the discharge and inversely as the distance. A current of 400,000 amp in a stroke will produce a field of 2620 maxwells at 1-ft distance, 262 maxwells at 10-ft distance, and 131 maxwells at 20-ft distance.

Any change in the electromagnetic field passing through either high- or low-resistance material will induce a voltage and current. The induced voltage and current will depend upon the rate of change in the magnetic field. It is evident that the lines of force passing through a person will induce currents, the action being similar to that in a high-frequency furnace. Several strokes in rapid succession may induce a greater current or potential than a single stroke of higher maximum magnitude but having a slower rate of change.

While it is possible to screen the electrostatic field, it is prac-

tically impossible to effectively screen the electromagnetic field so as to prevent the lines of force passing through objects in the vicinity of the field. The field may be set up by the current of a portion of the discharge in the air or in a conductor in the aircraft. It would seem that the heaviest discharges taking place close to a person may induce enough potential and current to affect the system, or at least to paralyze the nervous system temporarily.

A study of the factors affecting the induced potential and current go to show that the electromagnetic field may be materially reduced by removing the shunt path from the immediate vicinity, and by forming multiple paths around the pilot or passengers so that the field set up by the current in one path tends to neutralize the field set up in the other. The fact that some people have not been affected, although within a few feet of the discharge, would indicate that a reduction of the field strength by an appreciable amount would entirely change conditions and provide protection even for the most severe strokes. Parallel conducting paths on opposite sides can be used to advantage to reduce the effective field strengths caused by heavy discharges.

Further investigation may show that the induced current does not constitute a hazard. The magnitude of the voltage generated and the current induced for heavy strokes in the immediate vicinity, however, would indicate that the possibility of affecting at least the nervous system cannot be ignored without very definite proof to the contrary.

Although little has been accomplished in determining whether the shock or stunning effect is primarily due to electrical causes or to physical conditions similar to those producing shell shock, the same method will improve conditions for either case. This consists primarily in initiating the point of contact of the stroke to the plane as far away from the pilot as possible. Antennas or conductors projecting from the aircraft should therefore be a safeguard rather than a hazard.

(*g*) *Hazard Due to the Effect of Intense Light Upon the Pilot.* The light from a stroke, particularly at night, may have the effect of blinding the pilot when passing through the line of vision. At night the iris is wide open, and it is possible that the ultraviolet light might have some effect upon the pilot from a discharge striking the nose of the plane. In most planes, however, the discharge would be to one side and above or below the direct line of vision. It is believed, however, that the hazard from this source is small. While intense light may have the effect of blinding the pilot for a short time, any serious effect due to ultraviolet light may be easily eliminated by the use of glass in the windows or goggles that would absorb any injurious rays.

While the effect of lightning upon the pilot or passengers undoubtedly causes the greatest and most uncertain hazards, there are other hazards to the plane or aircraft that need careful consideration.

1 *Fire Hazard Due to Ignition of Combustible Material Used in the Construction of the Plane.* The fire hazard to the metal plane is negligible, although intense heat exists at the point of contact with the lightning and the plane. The fire hazard is practically negligible even where material that will support combustion is used. Where an inflammable covering is used in contact with metal, the fire hazard is also apparently negligible. Inflammable material that is readily ignited may be set on fire at the crater. However, the severe rush of air following the stroke has the effect of extinguishing the flame. The discharge being in the nature of an explosion, the rapid expansion of the gases, even in the fabric, apparently absorbs sufficient heat, which, together with the blast of air following the discharge, prevents ignition. This is particularly true where a fabric is used to cover duralumin or other metal. The heat conduction of the metal and the low resistance of the path afforded tend not only to absorb the heat but to reduce the energy dissipated. Heat generated by the discharge through a combustible material so situated that the radiation is cut off constitutes the greatest fire hazard.

Fig. 14 shows the effect of artificial lightning upon the rudder. It will be noticed that the fabric has apparently been exploded at the points of contact, the fibers being torn apart, the effect being similar to that of popping corn.

A rather severe series of tests run on a model zeppelin covered with fabric showed that it was impossible to ignite this fabric with the artificial lightning. The history of zeppelins has also indicated that this danger is small, unless conditions approximate those of the gas-filled free balloons. The framework and outer covering projecting some distance beyond the gas bags affords protection against the ignition of the gas by a direct stroke even where hydrogen is used.

2 *Fire Hazard Due to Ignition of Explosive Gases.* A discharge may take place between conducting surfaces not properly bonded or from isolated tanks or conductors separated from the bonded structure by small air gaps. The discharge may be produced by the release of a bound charge following the collapse of the electrostatic field, or due to the impedance drop in conductors



Fig. 14 Craters or Punctures Produced by Discharges to Rudder

making up the structure, or by the change in the electromagnetic field. These induced discharges are similar to those noticeable in the ordinary dwelling where a discharge takes place between wiring and conduit or fixtures following a stroke of lightning nearby. Should these discharges take place in a pocket of gas, an explosion may result. It is therefore important either to thoroughly ventilate all pockets that may accumulate gas or to adopt a construction so that a discharge cannot take place. Vents or openings for scavenging explosive gases should be protected with a screen similar to that used in a safety lamp.

3 *Weakening or Destruction of Metal Parts Due to Current in the Discharge.* In aircraft having a metal structure there is ample conducting capacity, so that a serious temperature will not be reached in any of the metal parts. However, should the construction be such that the current is confined to small members or to important connections having a high resistance, it is possible that the heating may seriously affect the structure and lower the mechanical strength. This is important, as some of the alloys are materially affected by a temperature far below the fusing point.

While the varnishing of the surfaces and joints, together with the oxidation of the surface, goes to produce a joint of high resistance, it must be remembered that the riveting cuts through the edge and forms a path of low resistance. A number of point contacts formed in this way results in a joint of very low resistance, so that little or no damage need be feared even though the cross-section of metal in good contact is small. Where metal parts simply come in contact or are held a short distance apart, considerable energy may be dissipated in the joint.

It would seem that the greatest hazard due to heavy current affecting the strength of metal parts is that due to control cables or staywires in non-metal planes. Discharges direct to the rudder showed a very appreciable rise of voltage on the control cable, although in the plane that was tested only a small percentage of the current was carried by the control cable, the major part going through the hinge and metal framework.

Fortunately the steel control cables usually have a high impedance compared to metal parts paralleling them. This high impedance tends to prevent a heavy flow of current in the control cables, although a very appreciable voltage may be induced under some conditions. An insulated section placed in the control

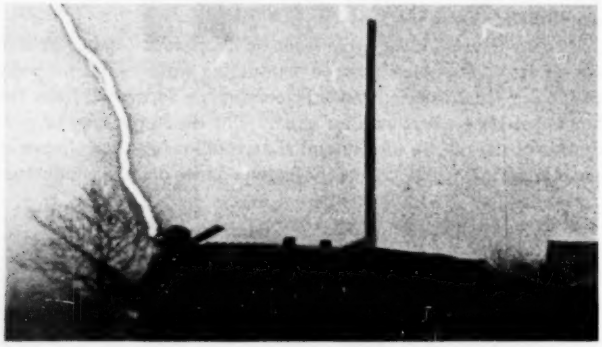


FIG. 15 DIRECT HIT TO IGNITION SYSTEM EXCEEDING 1,500,000 VOLTS



FIG. 16 DISCHARGE TO PROPELLER WHILE IN OPERATION
(Discharge strikes from termination of metal on propeller to ignition system opposite.)

cables can be used to prevent a high induced voltage on the controls, but this method should not be used to take the place of a parallel conducting path. Unless a parallel path is used the insertion of an insulator would tend to cause the discharge to leave the control cable. Should this discharge take place from the cable in place of the fitting, the heat generated at the crater might cause more damage to the cable than if the current were allowed to flow without restriction. The disadvantage of inserting the insulation may be easily offset, however, by providing a shunt between the control cable and any possible point of discharge, which would still permit the insertion of the insulator between the cable and controls.

4 *Breakdown of Insulation in the Ignition System.* Most of the ignition systems are fairly well insulated and have to withstand the relatively high voltages generated in the normal operation of the engine. Care should be taken so that conductors of the ignition system do not form a shunt path. In addition the conductors should be exposed as little as possible. The ignition systems apparently are more immune from trouble than would generally be expected. This seems to be due to the fact that the rise in voltage is limited by the discharging of the spark plug either inside or outside the cylinder.

Fig. 15 shows the ignition system being subjected to a discharge of over 1,500,000 volts. The discharge is striking the spark plug and lead on one of the cylinders. A number of similar discharges apparently had no effect upon the magneto or other parts of the ignition system. A small shield placed over the plug easily diverted the discharge.

5 *Backfire or Preignition.* It is evident that a discharge striking a plug or lead may cause a backfire or cause the engine cylinder to fire out of turn. A series of tests were made with the engine running so that the discharge would strike the propeller or ignition system. The propeller used was wood with a metal shielding which ended some distance from the hub.

Fig. 16 shows a discharge striking the end of the propeller. The discharge is then shown jumping from the metal shielding on the propeller to one of the spark plug terminals, as these happen to end about in line with the end of the shielding. A number of discharges apparently had no effect, although it appeared that the impulse from one cylinder was lost. The effect, however, was so slight as to leave this point in doubt. A small shield placed in front of the rod effectively drew all of the discharges. While the hazard is exceedingly small, it would therefore seem to be good insurance to prevent a discharge of this kind to the ignition system that might break down the insulation of the leads or the magneto.

6 *Damage to Instruments.* During one of the endurance flights at Cleveland the plane was struck by lightning, and the instruments were so damaged that a landing was made. An examination of the instruments did not show any electrical damage due to the conduction of current. There was an indication that the diaphragm in the instrument that was connected to the pitot tube used for determining velocity might have been damaged by the pressure set up. In the tests that were run some time later, it was evident that the pitot tube might be a point of contact for the stroke, in which case sufficient pressure might be set up so as to destroy the rather delicate diaphragm. A sudden air pressure set up by the stroke in any other way might of course have produced the same effect.

Should it appear that there is danger to instruments from the pressure set up, it would be a comparatively easy matter to prevent this by placing a baffle in the line between the pitot tube and the instrument, preferably with a surge chamber between the restriction and the instrument. Other instruments could be enclosed in a comparatively tight chamber having a window.

The electrical fields set up by a heavy discharge in the immediate vicinity may damage magnetic instruments or those in which an induced current may cause a breakdown of insulation. While electrostatic screening is comparatively easy, screening for the electromagnetic field is very difficult, and it would follow that instruments that are likely to be damaged by a strong magnetic field should be so placed that the path of discharge in the frame or surrounding objects is removed as far as possible.

7 *Damage to Rotating Parts Due to Passage of Current.* The damage to rotating parts may be serious where a heavy current must pass through an important bearing. Discharges with currents as high as 100,000 amp showed no appreciable damage on the main bearing or upon ball bearings. Since this discharge lasted for only a fraction of a microsecond, the tests do not prove that damage from this source can be entirely ignored. In general, however, it would seem that a discharge striking a

metal propeller would flash from the shaft to the face of the engine case without causing any damage. However, should the discharge take place in the bearing, it is possible that trouble would result. Owing to the resistance of the bearing, it is comparatively easy to provide a shunt path between the propeller shaft and housing so that the oil film will not be punctured by a discharge.

While the energy stored in the test condenser is of the order of 10,000 watt-sec compared to 10,000,000 watt-sec in the discharge of lightning, it must be remembered that the energy is dissipated almost entirely in heating the air, so that the energy dissipated in a bearing may not be any more with a lightning stroke than that under laboratory conditions, unless the discharge consists of a number in rapid succession liberating a considerable amount of energy.

8 *Sudden Change in Pressure on Adjacent Surfaces.* The sudden change in air pressure following a heavy stroke will probably not exceed that frequently occurring under normal flight. Should a discharge be close to and parallel to a surface, it is possible that a heavy effective pressure may be set up tending to cause a collapse. It is this sudden change in pressure that probably accounts for the tearing of fabric.

It is evident that the larger the spread of the conducting surfaces, the greater will be the danger of a stroke including the airplane in its path to ground or from cloud to cloud. While the use of an aerial extending some distance below the plane will tend to increase the danger of a stroke, it must be remembered that this will at least keep one of the points of contact of the discharge at some distance from the plane. This advantage may more than offset the increased probability of a stroke. The use of a loop set does not change the hazard in any way over that where no radio set is used. The use of a strut or mast extending above the plane for an aerial would seem to be an added protection, as it would tend to keep the point of contact at a distance.

The insertion of a resistance or impedance between the antenna and instrument with a shunt path to the frame of the aircraft will provide ample protection, the protection probably being much more effective than that provided for the ordinary house radio set using an outside aerial.

While hot gas is a good conductor, the question of the engine exhaust forming a conducting path that would tend to carry a stroke to the plane does not seem to be an appreciable hazard. In all of the tests the effect of the exhaust gases could not be noticed. In the many tests made there was no indication that a low resistance or conducting path was created by the hot gases. In fact, it would appear that the dielectric strength of the air was apparently increased by the wind or pressure produced by the propeller. While the gases are conducting for a very short distance from the exhaust, this hot gas is soon cooled by the mixing of the strong air current produced by the propeller, so that no effect upon a discharge can be expected.

Conclusion

In conclusion it may be said that while information is lacking as to the effect upon the person, much can be accomplished to remove the danger of this effect by giving attention to the various factors involved. While the hazard is exceedingly small, it is possible that still further improvements may be effected by taking advantage as opportunities in design and construction present themselves.

Further information upon the effect of shock will doubtless show that the hazard is not very great, although it may appear to be necessary to protect the pilot from sound or other conditions during a storm. Where the pilot has a fear of lightning, it is possible that tests or checks might be devised that would remove this fear.

Anything that will permit the safe landing of the plane or that will provide automatic control while the pilot is stunned would do much to eliminate the hazard that now exists. Owing to the increased reliability of aircraft, less attention will be paid to storms. While this will tend to increase the lightning hazard, it would seem that the present hazards can be more than offset by careful attention to the various factors tending to produce reliability.

AER-55-3

Photogrammetric Measurement of Airplane Landing Speeds

By V. C. FINCH,[1] AUBURN, ALA.

The paper gives a method of airplane landing-speed measurements. Wing loadings have mounted steadily in the past few years, with consequent increase in landing speed. Within the past two years wing areas have been so ruthlessly clipped that some of the 1932 models of conventional commercial airplanes are carrying weight per unit of wing area far in excess of the fastest wartime pursuit airplanes. To fit the government requirement of 65 mph as the limit for passenger-airplane landing with full load in still air, several manufacturers of stock models are assisting in the development of a method of actual measurement of landing speeds. The photogrammetric method gives an accurate and permanent record.



THE question of landing-speed measurement will not down. Wing loadings have mounted steadily since the war, with consequent increase in landing speed, and within the past two years, when economy of airplane operation has been so eagerly sought, wing areas have been clipped and trimmed with such ruthlessness that today some of the 1932 models of conventional commercial airplanes are carrying weight per unit of wing area far in excess of the fastest wartime pursuit planes. Certainly the use of high wing loadings and high landing speeds is conducive to economy and to the high cruising speeds so much desired. Cruising at 125 mph in an airplane with a landing speed of 55 mph costs several times as much per unit of distance and pay load as cruising at the same speed with a plane whose landing speed is in excess of 75 mph. If airplanes had advanced to a stage where forced landings were unknown, and if airplanes were never called on to make a landing at airports other than those with sufficient runway length to gain an A-1-A rating, the tendency toward even higher wing loadings and further economy might be encouraged to go on unchecked toward the final goal, the flying engine.

Unfortunately, under existing conditions, other things being equal, the airplane which lands most slowly is the safest, and the Department of Commerce has therefore set 65 mph as the limit at which a passenger-carrying airplane may land with full load in still air. Faced with the necessity of closely approaching this limit and not desiring to enter into production of stock models exceeding it, several manufacturers have seriously encouraged the development of a method of actual measurement of landing speeds. Their encouragement and their generous cooperation are chiefly responsible for the investigation described in this paper. While the work has been in progress, contacts have revealed that many present-day air-transport operators and even a few private owners are as much interested in the correct landing speeds of their airplanes as they are in the maximum and cruising speeds. Landing speeds obtained by approximate methods of calculation and by uncalibrated air-speed meters are frequently accepted as accurate by purchasers of new airplanes. In general, landing speeds obtained by these methods are appreciably lower than actual landing speeds, and the purchaser who intends to operate from a small field, and to this end selects the airplane of the type desired which is advertised as having the lowest landing speed, is almost invariably disappointed.

Keen competition in the field of airplane sales and the lack of a capable commercial flight-testing organization equipped to make standardized performance tests has led to the advertisement of some very optimistic landing-speed figures. Fig. 1 illustrates these discrepancies. The curve shows the average relation of wing loading to landing speed and is based on the results of the photogrammetric measurements made on airplanes employing medium-lift airfoil sections. The actual landing speeds of airplanes with medium-lift sections should fall on or near this curve. Thirty airplanes with medium-lift sections were selected at random, and their advertised landing speeds are plotted as points on this figure. The range of optimism varies from about 5 mph to nearly 20 mph.

The most common method of measuring airplane-landing speeds used in the past has been by the employment of a calibrated air-speed indicator. Air-speed indicator calibration is accomplished by flying low over a measured course and taking the time of covering the course with a stop-watch. This method has been found subject to the following inaccuracies: (*a*) Deviations in the flight path of the airplane, which are increased in the single-seater sport or pursuit airplane, where the pilot has to handle the stop-watch, sight the markers at beginning and end of the course, read instruments, and record data. (*b*) Air-speed indicator errors, the greatest of which are those caused by faulty pitot-tube installation. If the pitot tube is not parallel with the flight path or is located so close to a wing or strut that it is operating in disturbed air, the air-speed indicator reading will be much lower than the plane's actual air speed. (*c*) Personal errors in pressing stop-watch, sighting markers, etc.

The pitot bomb, suspended below the airplane, eliminates the installation errors, but its position renders it impracticable for determining actual landing speeds.

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Presented at the Sixth National Aeronautic Meeting, Buffalo, N. Y., June 6 to 8, 1932, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 DATA OBTAINED ON A THREE-PLACE MONOPLANE OF CONVENTIONAL DESIGN WITH FULL RATED LOAD

Test No.	Distance A ft	Distance Z ft	Distance Y ft	$x = \frac{AY}{Z}$	Camera speed, frames per sec	Frames counted	Computed ground speed	Wind velocity, mph	Observed air speed	Air temp., F	Barometer, in. Hg	Density factor	True air speed reduced to std. atm.	Remarks
1	2	26.25	416	31.7	68	47	31.3	9.1	40.4	72	30.05	1.009	40.05	3-point landing
2	2	26.25	536	40.85	70	62	31.5	9.3	40.8	72	30.05	1.009	40.45	3-point landing
3	2	26.25	492	37.5	67	48	35.72	10.0	45.72	72	30.05	1.009	45.32	Level landing
4	2	26.25	510	38.9	68	40	45.15	9.0	54.15	72	30.05	1.009	53.65	Take-off, normal
5	2	26.25	550	41.9	69	42	47.00	10.0	57.00	72	30.05	1.009	56.45	Take-off, tail high
6	High-speed course			519.5	68	190	126.8	9.1	135.9	72	30.05	1.009	134.6	Into wind
7	High-speed course			519.5	68	166	145.0	9.1	135.9	72	30.05	1.009	134.6	With wind

Calibration runs should, for accuracy, be made at or below an altitude of 50 ft. The wind should be parallel to the course and for accurate results should not exceed 14 per cent of the high speed of the airplane. The difficulty and danger of flying low over a measured course at low speeds may be obviated by pacing the airplane which is being tested with a slower airplane

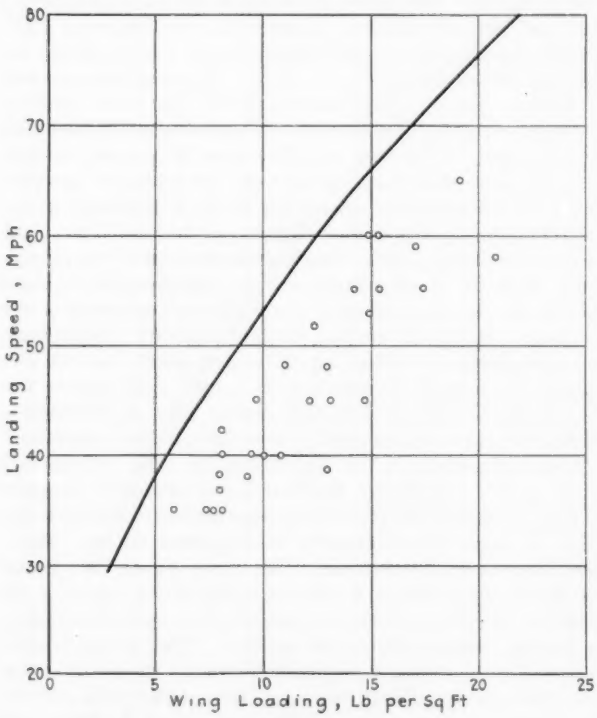


FIG. 1

equipped with a calibrated air-speed meter. The slower airplane should lead the way over the course at a safe altitude. Personal errors are, as stated, at a maximum in single-seater airplanes and are reduced, but not entirely eliminated, when an observer is carried. The air-speed-meter calibration curve is usually continued in a straight line from the lowest accurately observed speeds. True air speeds are obtained from the observed air speeds by correcting for atmospheric pressure and temperature, and for convenience and ready correction of subsequent air-speed-meter readings, they are plotted against indicated air speeds.

Another method which has been used for obtaining landing and take-off speeds (1)[2] consists of attaching an electric recording instrument to the landing wheels which graphically records their rate of revolution while in contact with the ground. From this and the known circumference of the wheel the landing or take-off speed may be readily calculated. This method also has been found subject to large errors from jouncing of the mechanism, partial deflation of the tires, lag of the wheels in landing, and friction.

The form of photogrammetric method of obtaining airplane landing speeds used at the Alabama Polytechnic Institute in tests during the past two years appears to eliminate most of the errors common to the methods described. Its greatest advantage lies in the fact that the pilot is left free to concentrate on landing the airplane. Instrument errors and personal errors are lessened to a great degree, for the reason that all observation is accomplished from the ground. In the tests run and the studies made the primary purpose has been to develop a simple method of landing-speed measurement that could be utilized easily and cheaply by airplane manufacturers and operators. In testing the airplanes submitted for landing-speed measurements, the aeronautical department of the college did not enter into an agreement to furnish certified performance data, but out of courtesy to the manufacturers submitting planes and in the desire to develop an accurate method, experienced engineering observers of the faculty performed the actual tests with graduate aeronautical engineering students assisting. The results of most of the tests were treated as confidential.

The data in Table 1 were obtained with a 16-mm motion-picture camera with standard lens and the expenditure of only 100 ft, or one loading of film. Economy of the process is further promoted by reversing the film in development from negative to positive, thus eliminating the expense of making a negative and taking positive prints from it. The small camera and film are not expensive and are entirely suitable for this work. The 16-mm film is particularly suited to the steady high camera speeds necessary for the slow-motion photography used because of the fact that this size gives 40 frames or pictures per foot of film as against 16 frames per foot of standard 30-mm film. This allows a lower film velocity through the shutter for a given number of frames per second and consequently gives greater accuracy in speed analysis.

The normal speed of motion picture cameras is 16 frames per second, and on the small cameras this speed is governed by a spring motor which maintains a surprisingly accurate rate. The camera used in obtaining the data in Table 1 has two speeds, the normal rate of 16 frames per second and the slow-motion rate of 68 frames per second.

The arrangement of the apparatus used is shown in Fig. 2. The camera is set up on a tripod at a distance that permits six standards to show in each frame on the film. These standards are $^3/_4$ in. by $^3/_4$ in. and are accurately located 2 ft apart, in a line parallel to the direction of the wind. The distance to the camera on a line perpendicular to the line of standards was for this particular "set-up" 26 ft 3 in. This distance will vary with the class of lens used and with the number of standards it is desired to show in the pictures. The distance from the camera to the path of the plane in landing and the spacing of the standards must be accurately known in order that the distance traveled by the airplane in its path between standards may be calculated. Assuming a distance from the camera to the path of the plane of 390 ft, we have, with 2 ft between standards, $26.25 : 2 = 390 : x$, or $x = 29.71$ ft.

[2] Numbers in parentheses relate to references at the end of the paper.

Briefly, the procedure for obtaining landing speeds is as follows: The set-up of the apparatus is made and checked, and white lines are laid down, as shown in Fig. 2. The plane is signaled to land, and the landing is made into the wind and parallel to the lines laid off on the field, which are parallel to the line of standards and the direction of the wind. These lines assist the pilot in landing directly into the wind and aid the observers in identifying the marks made by the wheels and tail skid when they take measurement Y for each landing. The character of the landing is noted, and readings are taken on anemometer, barometer, and thermometer. The time is noted, and later these three readings are checked with those of the recording instruments in the aerological tower of the airport. It is necessary for the plane to land at a point within the angle of the camera, and for this reason two white lines intersecting at the camera and passing through the bases of the end standards are projected out across the landing runway. A landing at any point between these boundary lines is satisfactory, providing the path of the plane is parallel to the wind direction. The camera is started just before the airplane enters the range, and the airplane is photographed as it passes across the range. Before stopping the camera after each landing, a close-up of a stop-watch is taken for 5 sec or more to give a check on the camera speed. A seconds pendulum swinging in front of the lens is used as an additional check. A slate with a number on it is then photographed to identify the film for future reference.

After the film is developed, it is run in a projector to check the observer's notes on the character of landings and take-offs. The slow-motion views are very interesting. The slightest motion of the control surfaces and the telescoping of the shock absorber unit as the plane comes in contact with the ground are plainly visible. After projection, the counting of the frames is accomplished under a magnifying lens. This enlargement assists in determining the exact frame where the landing or take-off occurred. In landing, the tail skid strikes the ground, and dust usually shows in the frame where the initial contact is made. If this point of contact is between the two standards, a count of the frames between the standards (i.e., from the frame in which the nose of the plane passes the first standard to that frame in which it passes the second standard) is used to obtain the correct ground speed. If the point of contact is in line with a standard, a similar count of frames is made from the frame in which the nose passes the preceding standard to that in which it passes the following standard. Equally accurate results can be obtained in the latter case if the length of the airplane is known and the number of frames counted from the time the propeller hub passes the reference point (the standard) until the tip of the rudder passes the same point. Camera speed is readily obtained from the stop-watch dial and the seconds pendulum as shown on the film. With the frame count thus made for the previously computed distance x, which the airplane travels between markers, the ground speed is computed from the following formula:

$$0.6818 \frac{(x \text{ times camera in frames per second})}{\text{frames counted for distance}} = \text{airplane speed in mph}$$

The wind velocity as taken from the anemometer is added to this, giving observed air speed. The observed air speed is corrected to standard atmosphere by dividing by the density factor $\sqrt{\rho_o}/\sqrt{\rho}$ obtained from $\rho/\rho_o = 17.32\ (\rho \text{ in. Hg})/T\ °\text{F} + 459.4°$. The standard atmosphere is an arbitrary variation of temperature, pressure, and density, given in Reference 2. Standard air, at 29.92 in. pressure, and 59 F or 518.4 abs, has a density of 0.0764 lb per cu ft as a basis for comparing performance.

Measurements of the take-off speeds and maximum speeds of the airplanes submitted for landing-speed measurements were also made with the 16-mm camera. The same arrangement of apparatus is used for measuring take-off speeds that is used for measuring landing speeds. In measuring take-off speeds, the airplane starts at a point estimated to be the correct distance outside the camera angle to allow the airplane to leave the ground at a point within the angle. This distance will vary with the speed of the wind, and usually several preliminary trials will be necessary before a normal take-off will be made within the camera angle. The observers note the character of the take-off and record the data previously mentioned for landing-speed measurements.

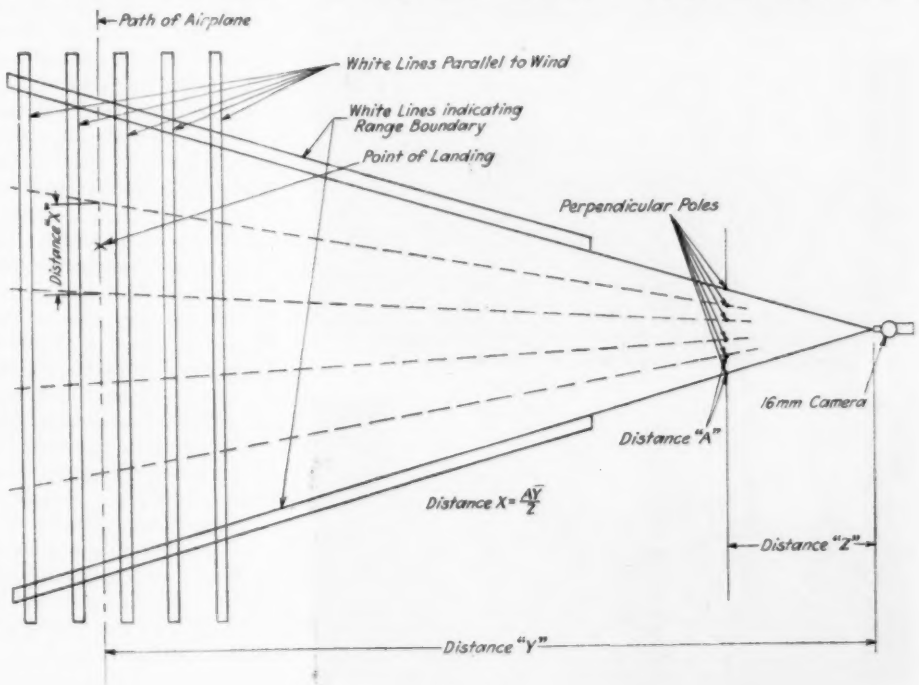


FIG. 2

The measurement of take-off speeds has not been stressed, as length of take-off run and rate of climb are figures that are easily obtainable and are considered more vital to the airplane operator than the rate of speed at which the airplane leaves the ground. The performance requirements for take-off imposed by the Department of Commerce on a landplane are that it shall take off within 1000 ft at sea level in still air with full load, and that the climb in feet during the first minute after take-off shall be at least eight times the theoretical stalling speed in mph, but it must in no case be less than 300 ft.

For maximum-speed tests the camera is secured in the airplane, focusing down at right-angles to the thrust line. The plane is flown level at full throttle over a measured distance, first into the wind and then with the wind. A course parallel

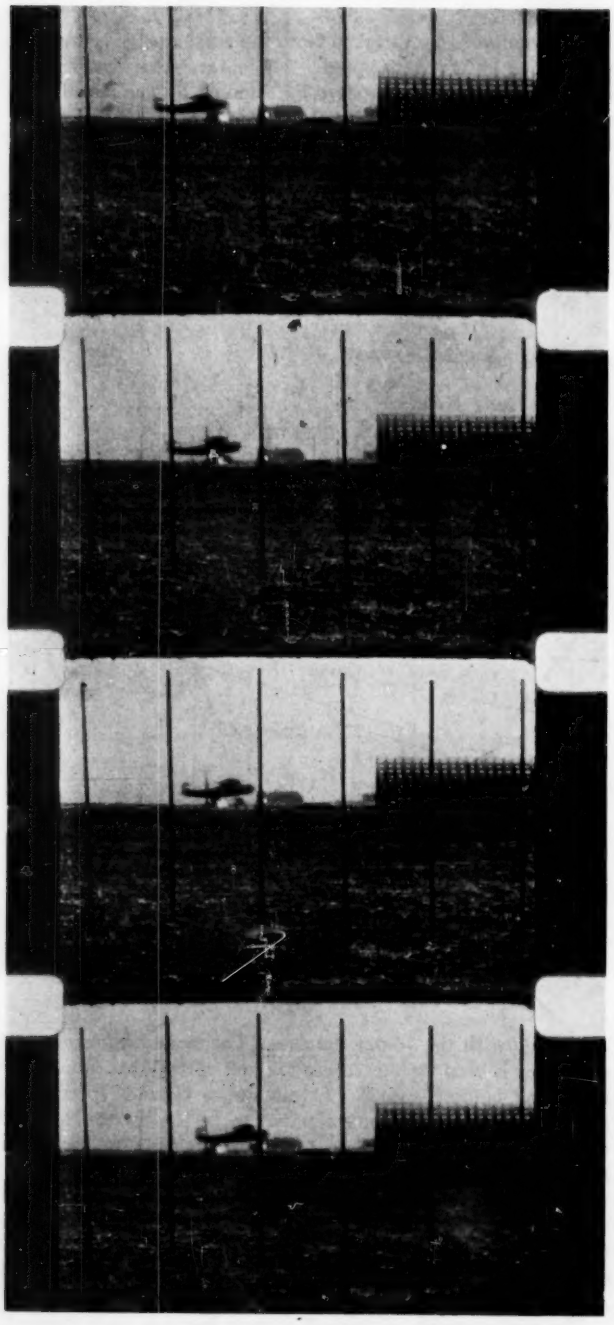


FIG. 3

with the wind direction and about 500 ft in length over level ground is adequate. This test requires careful piloting, and a number of runs must be averaged to insure accuracy. A sensitive level-flight indicator is of considerable aid to the pilot in making this maximum-speed test. Runs are observed from the ground, and where there is any deviation from level flight, the run is eliminated from the computation. The difference between the average of the speed in both up-wind and down-wind directions and the speed in either direction equals the wind velocity. A comparison of the wind velocity obtained in this manner with the anemometer reading gives a check on the accuracy with which level flight has been maintained. In this high-speed test the camera is started before the line marking the beginning of the measured course comes in range and is stopped when the line marking the end of the course is out of range. The pilot must fly at a constant altitude (50 ft has been used in our tests) and at a horizontal distance from the markers that will permit them to fall within the camera angle as mounted in the airplane.

An alternative method of measuring high speed is exactly similar to that used for measuring landing or take-off speeds with the camera placed low, and, if necessary, somewhat closer to the standards. An objection to this method is that the measuring course is very short and that piloting errors are therefore magnified. With a longer course, errors in piloting are usually noticed by the pilot and are slightly over-corrected. Thus they tend to compensate each other. Outlying markers at a distance of about 500 yd from the standards should be established over which the pilot must pass in level flight.

Photogrammetric measurements of maximum speeds have been found to compare closely with speeds obtained by timed runs over a 2-mile course, but these close comparisons were found in tests where the pilot making the photogrammetric runs conscientiously tried to fly at a constant level over the course. The majority of the maximum speeds found by the photogrammetric method were high because of the fact that on one or more of the runs the pilot, inadvertently or otherwise, would start across the short course at a greater altitude than that at which he was flying when he reached the end of the course. This almost imperceptible "downhill" flying has occurred very frequently in the maximum-speed tests, and for this reason photogrammetric measurements of maximum speeds are not recommended. Runs over a 2-mile course timed with a stop watch appear to offer better conditions for accuracy and should be employed for maximum-speed measurement whenever possible.

The important application of this form of the photogrammetric method of obtaining airplane speeds is in the measurement of landing speeds. It gives an accurate and permanent record which with careful checking can be certified as being correct within 1 per cent.

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Discussion

A. M. PRIDE.[3] The method presented is of considerable value in that it provides a complete history of the landing operation and enables the observer to study rebound, changes in attitude,

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etc., as well as to obtain a continuous record of speed. One of the first requisites for accuracy seems to be an exact knowledge of the distance from the camera to the airplane, and experience along these lines leads to skepticism as to the dependence that can be placed on a pilot's ability to fly parallel to the indicated lines. A positive means for determining the path of the airplane is probably necessary, and a trailing rod that will scratch the ground suggests itself.

Several years ago, in measuring airplane speeds in landing on board a carrier, by means of a camera that was fixed normal to the airplane's path, it was found that an error of about 2 per cent occurred due to blur in the image on the film. It is possible that a similar condition would occur with this method if photographic material capable of "stopping" very rapid motion with sharp definition were not used. It is suggested that the requirement for this class of material be stressed.

Timing by counting frames with a spring-motored camera is considered highly satisfactory, since experience involving a great many measurements and check measurements in which photographs of split-second watches were made in connection with the landings on board ship, mentioned above, has indicated that the maximum time error is less than 2 per cent, and this with ordinary aerial camera guns and commercial hand-held cameras selected at random. If the equipment were to be selected especially for uniformity of film motion, the error could undoubtedly be reduced to less than 1 per cent.

In any measurements of this nature, the complication of errors due to rotation of the airplane as the tail rises or falls must be accounted for if a high degree of accuracy is to be obtained. Since by "speed" it is assumed that the velocity of the center of gravity is meant, it is obvious that measurements referred to any other part of the airplane are not directly applicable, if pitching occurs. The superposition of the effect of rotational velocity on the lineal velocity obtained in such a case is difficult, and it is suggested that the center of gravity be marked on the fuselage and that all measurements be taken directly from this mark. Due allowance must be made for the lateral distance from the fore-and-aft center line to the side of the fuselage.

C. F. Taylor.[4] The value of the paper would be enhanced by data showing the degree of accuracy of the readings. A table might be given of the readings and the computed landing speeds for a large number of landings by two or three different pilots with a given airplane, other conditions being held as nearly the same as possible for the whole group of tests. Such a table would reveal the amount of error to be expected in using this method.

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Pressure in Airships

By W. A. KLIKOFF,[1] DETROIT, MICH.

The use of pressure in the so-called "pressure airships" and the methods used for maintenance of pressure are described. The effects of "gas head" and of excess internal pressures are considered as design factors, showing the advantages derived from a strength standpoint. It is pointed out that, by utilizing the pressure, the low factors of safety used by airship designers for critical loading conditions can be eliminated, and the ultimate design of the pressure rigid airship is discussed. Recognizing the disadvantages of manual control of pressure employed in the present airships, it is emphasized that pressure control can be made perfectly automatic if pressure regulators used in the modern power-plant installations are adopted for airship-pressure control. The proposed pressure control system for large metalclad airships is described. To utilize the advantages of pressure as a design factor, the materials used in the present rigid airships should be more efficiently disposed around the cross-section, again pointing to ultimate metalclad design.



THE type of lighter-than-air craft whose shape is maintained solely by internal pressure is often called the "pressure airship." In this type of airship the contraction and expansion of the gas with changes of atmospheric conditions are compensated for by the air-filled ballonets. The internal air pressure is maintained by blowing air in the ballonets, thus raising the pressure slightly above atmospheric. The air pressure acts on the movable ballonet diaphragm, thereby increasing the pressure in the gas container and producing tension in the outer cover.

For purposes of analysis this tension is usually considered as having two components: longitudinal, acting in the direction of the ship's axis, and transverse (or hoop), acting in the plane perpendicular to it If airships had been made as perfect spheres, then the longitudinal component of pressure would be equal to the transverse, but as long as aerodynamic requirements dictate a slender, elongated shape with much larger radius of curvature in the longitudinal direction than in the transverse, the longitudinal component of pressure becomes smaller than the transverse. Each one of these components performs a well-defined function. The transverse component of tension retains the shape of the cross-section, which will be circular in the case of an airship subjected only to internal air and gas pressure and loads distributed uniformly around the circumference. The longitudinal component of tension, due to pressure, maintains the longitudinal shape of an airship. When the airship is built of materials that cannot resist compression loads, such as the non-rigid airship, this longitudinal tension component should be always of such magnitude that the tension due to it exceeds the compression imposed by either static or aerodynamic loads acting on the hull. This consideration determines the value of the internal pressure to be used in the hull.

It may appear that the internal pressure should be of considerable magnitude to resist these loads, but actually, due to large cross-sectional areas, the pressure required in the case of non-rigid airships is, in general, of the magnitude of $1^1/_2$ in. H_2O, or approximately only 0.37 per cent higher than the standard atmospheric pressure. Such a small pressure can be very easily maintained mechanically, but is also easily subjected to atmospheric variations.

This small increase of pressure corresponds approximately to only 2 F change in temperature and 0.11 in. Hg change in pressure, or 100 ft change in altitude. This comparison shows the sensitiveness of pressure to outside atmospheric conditions and the necessity for careful handling.

Maintenance of Pressure

The excess of pressure is handled by automatic valves which tend to release the air from the ballonets in case of gas expansion, and at the present time they have reached such a degree of development that no particular attention is attached to this phase of the problem. The capacity of the valves is usually designed for the maximum desirable rate of ascent, and the valves are set in such a manner that the air valves open first and the gas valves follow, so as to preserve gas as much as possible. Manual control of the valves is provided to take care of the possible failure of the automatic features.

The means of supplying internal pressure have not reached such a degree of perfection as the means of releasing it, and at the present time are rather cumbersome. In non-rigid types the pressure is usually maintained by scoops which are lowered manually in the slipstream of the propellers. This manual control has the disadvantage of constant attention on the part of the pilot. The drop in pressure to or below atmospheric will cause the collapse of the envelope and the necessity of abandoning the forward speed, and will put the airship at the mercy of the winds like a free balloon. The maintenance of too high a pressure produces high tensions in the fabric envelope, causing a more rapid deterioration.

The semi-rigid airship was evolved to relieve the tension load in the fabric, and part of its ability to withstand outside bending moments is achieved by the introduction of a rigid keel capable of carrying a very high percentage of the bending load.

In the case of semi-rigid airships the pressure is often maintained by means of dynamic scoops which are located in such places on the hull that they automatically scoop up air in the case of forward motion, producing higher internal pressures in case of higher speeds. This arrangement has the disadvantage that if the scoops are not properly located they may be blanketed

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

somewhat in some of the maneuvers, reducing the delivery of air. In addition the scoops become inoperative when the forward speed of the airship drops to zero. In this case the pilot has to rely on mechanical blowers, which should always be ready to be started for this purpose. In the hangar or when maneuvering on the field, the blowers are the only means of maintaining pressure. Being manually controlled, they require constant attention. These factors tend to make the maintenance and operation of the present fabric-covered pressure airships troublesome.

The opinion is often expressed that rigid airships are independent of pressure. This opinion is erroneous, because the rigid airships are not independent of pressure. They tend to fly only under such conditions that pressure inside is near atmospheric, and a series of openings are provided to keep the internal pressure as close as possible to atmospheric. Failure to do so may lead to disaster. Some of the rigid airships tend to utilize the effect of pressure for the support of fabric or other purposes. For instance, the latest British airship, the *R-101*, was normally operated under slight positive pressure to reduce flapping of the fabric covering. Openings in the envelope were provided to satisfy these conditions.

Pressure as a Design Factor

The question arises, if pressure is such a cumbersome factor and one requiring so much attention, why use it at all? Why not eliminate it entirely and build airships perfectly independent of pressure? The answer is that internal pressure is always present, even in the case of the conventional rigid airships, where it is often not used to any advantage and with the present types of design may constitute even a liability. The pressure that is referred to in this case is the gas-head pressure. The pressure within a gas container increases due to the difference of the densities of air and gas. This property of gas is called "gas head," which is exactly similar to the "water head" in hydraulics, but of an opposite sign.

If h represents the head of gas measured upward and k the unit lift of gas, the pressure due to gas head becomes evidently kh. This pressure produces a constant longitudinal force, which in the case of a fully inflated circular airship is $k\pi r^2 h$, where r = the radius of cross-section.

The pressure due to gas head increases toward the top of the airship; therefore the resultant of this longitudinal force acts above the center line, which usually coincides with the neutral axis of the airship. Consequently, it will tend to produce a hogging bending moment, which for a fully inflated ship is equal to $k\pi r^3 h/4$.

The values of longitudinal forces and bending moments for sections not completely filled can be found in textbooks on airships.

This hogging bending moment and this longitudinal force due to gas-head pressure are present in all airships. In rigid airships there exists another factor due to gas pressure. Whereas in non-rigid types the transverse component of pressure produces uniform transverse tension in the covering, in rigid airships this transverse component acts as a side load on longitudinals, complicating their design by loading them with side load combined with direct stresses due to bending of the whole airship. This loading condition of longitudinals tends to explain why gas pressure is often called a liability in the case of conventional rigid airships.

The present large airships reach such large proportions that the gas-pressure factor is becoming of primary importance. The diameter of modern airships has already reached 132 ft. In this case the pressure due to gas head at the top of the hull at the maximum section becomes, for a helium-filled ship, 8.20 lb per sq ft, or 1.57 in. H_2O. The total longitudinal force becomes 57,000 lb, and the hogging moment due to it 950,000 ft-lb.

These gas-head pressures due to the properties of lifting gas produce forces and moments reaching such magnitudes that the airship designer should undoubtedly try to utilize them as much as possible to his advantage. The longitudinal force is the most helpful one because it tends to produce a uniform tension throughout the structure, and all materials used in airships can carry much higher tensile loads than compression loads. It will be of interest to study the help that could be derived from the longitudinal tension on the hull due to pressure.

Consider an airship whose cross-sectional area of metal used is A, which is distributed uniformly around the cross-section; then the moment of inertia of this cross-section is $Ar^2/2$. If this airship is subjected to a certain longitudinal force P and bending moment M, then the maximum compressive longitudinal stress will equal

$$S = \frac{P}{A} - \frac{2Mr}{Ar^2}$$

Assuming, as in the case of the non-rigid airship, that the outer covering is incapable of carrying any compressive loads or limiting the value of stress $S = 0$, then solving the equation, we obtain the critical value of bending moment in relation to longitudinal force $M/P = r/2$. If the non-rigid airship is full of gas and is subjected to pressure due to gas head only, then the relation of gas-pressure bending moment to longitudinal force due to gas pressure is:

$$\frac{M}{P} = \frac{kr^4}{4kr^3} = \frac{r}{4}$$

or the value of the hogging bending moment can be doubled before compressive loads will occur in the fabric, and the airship could carry a sagging bending moment equal to three times the gas-pressure moment without having any longitudinal folds in the fabric.

Introducing an internal pressure higher than the pressure due to gas head, such a rigidity of the hull can be obtained that a very severe bending moment can be carried by the hull, which is exactly the practice followed in non-rigid airships.

It will be of interest to arrive at actual figures to determine what is the value of the internal pressure required in modern airships to resist bending moments. Take, for instance, the case of the British airship *R-101*, which was destroyed with great loss of life in France. During the inquiry of causes of this disaster, and previous to that, the designers released more reliable information about this airship than about any others recently built.

This airship was 732 ft long, 132 ft in diameter, and had an air displacement equal to 5,600,000 cu ft. The arbitrary value of the maximum aerodynamic bending moment to which this size of airship would be subjected is:[2]

$$M = 0.02 \frac{\rho}{2} V^2 (\text{vol})^{2/3} L$$

or at 75 mph at sea level,

$$M = 0.0000237 \times 110^2 \times 31{,}000 \times 732 = 6{,}640{,}000 \text{ ft-lb.}$$

Assuming that the gas-pressure bending moment resists all the static loads, then to counterbalance the effect of this aerodynamic bending moment the longitudinal force due to internal air pressure should be:

$$P = p\pi r^2 = \frac{M}{r} \times 2$$

and internal pressure

$$p = \frac{6{,}640{,}000 \times 2}{3.14 \times 663} = 14.7 \text{ lb per sq ft} = 2.82 \text{ in. } H_2O$$

[2] Technical Report No. 325, National Advisory Committee for Aeronautics.

It can be noticed that by increasing the internal pressure only 180 per cent above the pressure that already is present on the top of the ship due to gas head, all the compressive stresses which would appear in the hull due to the action of this severe aerodynamic bending moment will be eliminated. This computation does not mean that all the stresses will be eliminated—they will appear on the tension side, but as long as the materials used for the present airships have such properties that they can resist tension from 50 to 100 per cent better than compression, the advantage of utilizing the internal pressure can clearly be seen.

The fact that materials should be utilized always on the side on which they are most efficient has long been recognized by the structural engineer, who in suspension bridges utilizes the high tensile value of the cables and in concrete structures utilizes the high compressive value of concrete.

Design Conditions and Factors of Safety

In the present design of rigid airships a rather peculiar system of factors of safety is adopted. Factors of safety of 4 and higher are used for static loads, but when the aerodynamic loads are superimposed, then the designers do not increase the structure in proportion to the increase of load, but increase the structural strength only to some extent which causes decreasing of the factors of safety. This practice is justified by the fact that conditions of superimposing both types of loading occur less often and the effects of higher loads on the structure will be less. For this reason airship designers are satisfied to drop their factors of safety to as low as 2, and sometimes even smaller for the worst loading conditions. This method of design may give the operating personnel a false sense of security, making them overconfident in the strength of airships under normal flying conditions, and in case of emergency they may treat the airship without due caution, causing perhaps a breakage of structure and severe disaster. Several airship accidents were traced to this cause by some of the experts.

It must be kept in mind that the nature and magnitude of gusts and other aerodynamic forces are not completely known and probably never will be definitely known, just as in shipbuilding the forces due to the impact of waves are just approximated from the results of years of experience. Airships while floating in a fluid have the advantage of being more easily analyzed for aerodynamic loads due to the fact that air is a more perfect fluid than water, but air has a peculiarity of having very steep rising and descending currents which may not be visible to the eye of the pilot.

Therefore a somewhat more logical system of airship design would be not to design the airship for too high a factor of safety for average conditions and too low for an extreme condition, but to design an airship, for instance, for the following conditions:

1 Adopt a reasonable factor of safety for the worst static conditions, combined with certain usually encounterable aerodynamic loads.

2 Provide enough compressive strength in the airship to be able to carry these loads safely.

3 In flight when the airship is subjected to severe aerodynamic loads, provide the excess strength and maintain the same factors of safety by introducing certain other positive factors of strength; for instance, one which is the most readily obtainable, pressure.

The operating personnel will then know that when the airship is operating under internal pressure the compressive strength is not endangered, the loads are taken through materials in tension, which are known to have much higher strength value than in

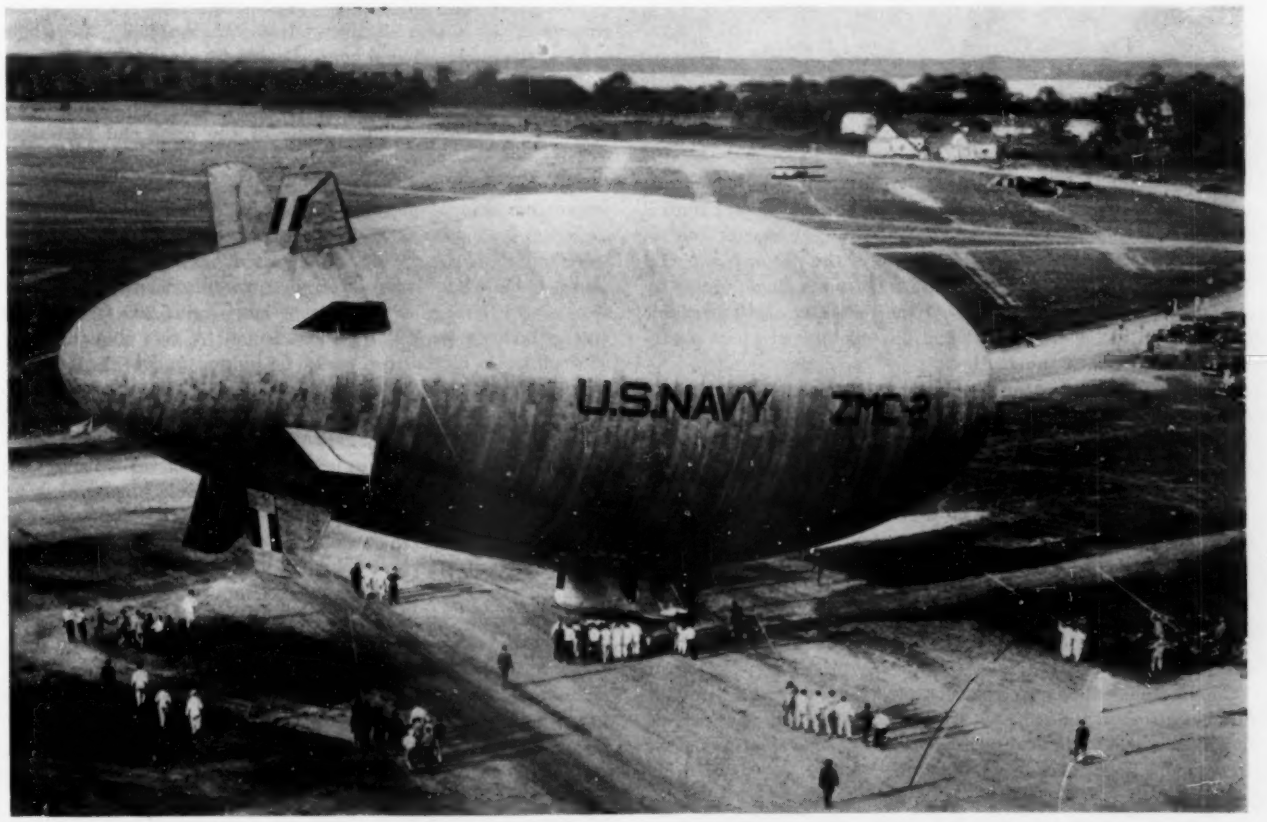


Fig. 1 First Successful Pressure Rigid Metalclad Airship

compression, and the airship can be safely flown through any atmospheric disturbance and can withstand any maneuvers.

When the pressure system is not functioning for some reason or other, then the operator knows that the airship is not strong enough to withstand severe loads and should be handled carefully. It should be slowed down, like an ocean liner partially disabled at sea, no excessive maneuvers should be made, and aerological disturbances should be avoided if possible.

If the pressure system is designed with the required amount of care, then the last condition should really never occur in the life of the ship.

Pressure Rigid Airship

The designers of the large metalclad airship have always kept in mind the policy that the system of design just outlined is a more logical one and have always considered it as an ultimate goal of metalclad airship design and development.[3]

As was mentioned previously, conventional pressure airships are so dependent on the pressure that a drop in pressure means a loss of shape and inability to proceed at a forward speed, thus transferring the airship to the free-balloon class.

This condition is undesirable, and it should be the aim of designers of pressure airships to design the airships in such a manner that they can maintain their shape and proceed at slow speeds at atmospheric internal pressure, using the internal pressure as an auxiliary means of extra strength and rigidity in case of flying at high speeds.

The question arises as to the possibility of controlling the internal pressure. All the disadvantages of the pressure supply system as used in present airships can be eliminated easily if airship designers would only try to follow in the footsteps of the mechanical engineer, who has already solved the problem of pressure control very admirably in power-plant installations.

Pressure Control in Power Plants

It is seldom realized to what extent the control of power plants has been taken out of the class of the uncertain human element by the automatic regulator. In a typical automatically controlled power-plant installation, the regulators control the fuel feed, air supply, furnace pressure, and forced- and induced-draft fans. All of these regulators are connected to a master control panel, from which they can be controlled. The master control panel is thus the central control station of the power plant, and the operator can control the whole plant from a single source by simply adjusting the setting of all the regulators. Remarkable decreases in fuel consumption and an economy of plant operation are achieved by this method.

Sometimes the regulators are called upon to regulate accurately pressure and pressure differentials to within an error of 0.001 in. of water column and in other instances to regulate the steam pressures running into hundreds of pounds to within a limit of a few pounds.

Automatic combustion control is used in the most modern and large power plants, such as the Brooklyn Edison Company's station, Hell Gate Station, Detroit Edison Company's stations, and Bethlehem Steel Company's power plants. The application of automatic control is not limited to power stations, but it also is applied to coke ovens, open-hearth furnaces, and blast furnaces.

The mechanical engineer has accepted the automatic regulator and is using it in the industry and is relying on it in complicated installation where slight disarrangement of the system will cause tremendous losses and disorganization.

[3] R. H. Upson, "Metalclad Rigid Airship Development," delivered before the S.A.E., February, 1926, gives the original ideas on internal pressure control as applied to metalclad airships.

Application of Pressure Control to Airships

The airship engineer, by taking one item out of the complicated power-plant-control system and installing it in the airship, may solve the whole problem of pressure control on airships. This item, the pressure regulator, can be designed to a range of accuracies exceeding considerably the airship requirements and would tend to make the control of pressure entirely automatic.

Several automatic pressure regulators could be installed in such a manner that a drop of internal pressure would cause either the immediate opening of the propeller scoops, in case of flight under power, or would start the blower motor, which would be kept constantly in warm condition ready to start. It might even be desirable to provide a regulator which in case of failure of the pressure system would switch off some of the engines, thus reducing the speed of the airship to a point safely allowable for operation without pressure.

The operating personnel, with automatic pressure regulators installed, would set the regulators at a certain selected value of internal pressure. The regulators would take care of the variation of pressure by operating pressure-supplying units. All of these pressure-supplying units could be connected to the control car, and any change in their position would then be shown by a signal on the board installed in the control car. Thus all the elements of the internal pressure-control system could be checked periodically by releasing the pressure and observing the functioning through the signals. Any uncertainty of the system could be immediately adjusted so as to have the whole system in perfect condition in case of emergency.

Proposed Pressure Control in Metalclad Airships

It is recognized by the designers of large metalclad airships that the pressure system should approach perfection as closely as possible and that automatic pressure regulators should be incorporated as a part of the system. Undoubtedly the whole body of the airship should be made airtight.

This can be attained by providing all the means of access with doors that are pressure-tight. This is not considered a difficult engineering problem. Then some of the crew and passenger quarters, as well as corridors and means of access to the structure, could be placed in the hull in the air space, which under certain conditions would be under slight pressure. This also can be easily achieved, and the small internal pressure inside of the airship of, say, 3 in. water would not be uncomfortable, because this slight increase in the atmospheric pressure corresponds to a change of altitude of only about 200 ft, and the human body is known to be unaffected by this change. The whole interior of the ship could be lighted and be kept under slight pressure when necessary.

If the pressure rises above the design value, then an excess of it could be released by the air valves, the design of which would be very similar to the valves used at the present time.

If the pressure drops, then some means should be provided for its replenishment. The main pressure-supplying units would undoubtedly be the dynamic scoops, which are really inward-opening valves, located at several points on the hull where the external dynamic pressure due to the forward motion of the airship will exceed the internal pressure required for the longitudinal strength. The design of dynamic scoops is a very simple problem, and they would normally be used as the only pressure-supplying units. Their action can be made perfectly automatic, and with increase of speed, air at higher pressures would be supplied.

In case it should be desirable to supply more constant pressure, irrespective of changes of speeds or in case of descents under power, when a large quantity of air must be supplied, the propeller scoops located in the slipstream of the propeller can be used. It is proposed to control these scoops by automatic pressure regu-

lators. If the amount of air supplied by the dynamic scoops proves to be insufficient, then the pressure inside of the hull would drop, and the automatic pressure regulators influenced by this drop of pressure would open the propeller scoops.

Mechanical blowers also should be installed to supply air for rapid descent without forward speed and for ground handling. The blowers should always be kept in such condition that they can be started at short notice and develop the maximum pressure. The blowers also would be controlled by the automatic pressure regulators, which would start them when the system of dynamic and propeller scoops might prove to be inadequate under extreme conditions.

In case of complete failure of the pressure system, recourse might be had to the present rigid arrangement, i.e., opening of certain apertures in the hull, tending to maintain inside pressure equal to atmospheric. Then the airship would assume the same condition as when maintained in the hangar and also would be able to fly at reduced speeds, as previously mentioned.

Manual control of all the pressure-control items could be provided also as an auxiliary to the automatic control.

It can be clearly seen that the proposed installation would take out of the hands of the operating personnel all necessity for constant attention to the pressure system except for routine checking and inspection, and would remove the most important objection to the use of internal pressure as a strength factor in airships.

Types of Airship Structures

After considering pressure as a factor in airship design, it may be desirable to review the types of structures used in the present airships. It must be admitted that the present rigid airship structure is very similar to the familiar airplane wired-frame system used in the wartime days. The airplane fuselage built according to this system had four longerons acting as major structural members, braced by shear wires and covered by doped fabric, which, to produce a shape of low resistance, was supported by fairing strips. Only the longerons and shear wires acted as structural members, the remainder of the cross-section being added only for streamline effect.

The present rigid airship employs a very similar arrangement, except that it even complicates it by adding the gas cells and the gas-cell wiring. Therefore, if all the elements of the cross-section of the rigid airship are considered, it will be found that a large percentage of the total weight is not efficiently utilized from a strength standpoint, but serves auxiliary functions such as gas holding and preserving exterior shape.

Assuming a certain rigid airship and applying to it an arbitrary value of bending moment, it will be found that the stresses on the tension and compression side will be equal, and for comparison it may be assumed that the factor of safety is 4 on the compression side and 7 on the tension side. Suppose the longitudinal force is now introduced of such magnitude that it eliminates all compression loads, then there will be zero stress on the compression side, but on the tension side the factor of safety will drop to 3.5.

From this consideration alone it appears that it is not advantageous to utilize the effect of pressure on rigid airships. This is due to the fact already mentioned that rigid airships do not utilize all the materials disposed around the circumference as load-carrying factors. If all the materials employed could be utilized, then the cross-sectional area available for load-carrying purposes would be increased and the unit tensile stresses would drop. That is exactly the principle used in non-rigid airships, where the whole cross-section acts as a load-carrying member. This statement is proved by the fact that theoretically in non-rigid airships the same overall efficiency can be reached as in rigid airships. The proper utilization of the cross-sectional area, although handicapped by inability to carry any compressive loads, can be made as efficient as the rigid system of design.

It would appear that the most efficient system of design should utilize the compressive and tensile ability of the cross-section to the utmost advantage. Taking the case of the airship just considered, the most advantageous value of internal pressure will be such that the factors of safety on the compression and tension side are equal, or 5.5 for the loading aforementioned. It will be hard to balance the pressure exactly at this point, but this investigation points to an ultimate design of airship that will tend to utilize completely the whole value of the cross-section as a stress-carrying member capable of carrying high tensile loads as well as certain compression loads.

As has been mentioned, the ultimate metalclad airship design would try to approach these requirements. In it the high transverse stresses due to pressure, which are so detrimental to fabric non-rigid airships, are easily carried by monocoque metal covering. The hull of such a ship also may be sufficiently reinforced in the longitudinal direction to carry appreciable compressive loads. The perfection of this method of construction, employing the proper combination of internal structure with stressed skin covering, will produce the very efficient large-size airship capable of the high speed essential to commercial application.

Conclusion

It cannot be denied that pressure is one of the major factors in airships. It is always present as a gas-pressure force. It is a question whether it would be advisable to utilize the pressure as a strength factor or to try to combat its effects as much as possible. So long as airships are built of materials that are more efficient in tension than in compression, and pressure provides a factor which allows carrying all the stresses on the tension side, then it is certainly advantageous to utilize pressure as a strength factor. This can be accomplished by raising the internal pressure higher than the already-present gas-head-pressure value.

The question of control of internal pressure can be solved by an application of the modern methods of control already adopted by mechanical engineers.

It is undeniable that from a safety standpoint airships should be built not fully dependent on pressure. The ultimate airship must be designed to be rigid enough at atmospheric pressure for maintenance in the hangar and for slow-speed flying and to utilize the pressure only as a positive factor of excess strength to withstand extreme loading conditions. Such an airship in counterbalancing the excessive external loads by internal pressure coincides with the logic of Count Zeppelin, who many years ago said, "The forces of nature cannot be eliminated, but they may be balanced one against the other"—to serve man's needs.

AER-55-5

Air Compression With Temperatures Above Adiabatic, With Special Reference to Airplane Superchargers

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In compressing air or other gases by means of reciprocating or centrifugal compressors of comparatively large size, with large cooling areas and often water-jacketed, an appreciable amount of heat is removed which helps to decrease the power input. This cooling, however, is by no means essential, and under many circumstances decrease of power for compression by cooling is not very great. Furthermore, conditions are often encountered which make cooling inadvisable. In the present paper the author outlines the mathematical theory to be used when due consideration of all of the circumstances indicates that a compressor with a small amount of cooling is the most advantageous.



COMPRESSION of air or other gases originally was accomplished by means of reciprocating or centrifugal compressors of comparatively large size and with large cooling areas, often water-jacketed. An appreciable amount of heat was removed during compression, which helped to decrease the power input. The usual mathematical treatment of compression implies such possibility of appreciable cooling. However, this cooling is by no means essential. The upper curve of Fig. 5 shows that, under many circumstances, decrease of power for compression by cooling is not very great. Frequently conditions are encountered which make cooling inadvisable.

Figs. 1–4 give examples where consideration of all of the circumstances has led to selection of machines without water jackets. The machines shown in Figs. 1, 2, and 3 have rotational speeds from 14,000 to 33,000 rpm in order to decrease the overall size of the casing. This results in a casing area so small that there is relatively little cooling, even with use of radiating fins. The requirements regarding weight and space are such that multi-stage compressors are not desirable, and there must be appreciable pressure rise in a single stage. Pressure ratios of 1.5 to 2.0 with air are quite common, and machines have been built with a pressure ratio as high as 4.0. With air at sea-level inlet pressure, this gives discharge pressures of from 7.5 to 15, and in the extreme case 45 lb per sq in. gage, in one stage. Fig. 4 shows a larger unit of 13,000 hp capacity for quite a different service, but to which the same mathematical considerations apply, on account of the final air temperatures, which are arranged to give the best heat balance for the plant.

It is the purpose of this paper to outline the mathematical theory to be used when consideration of all of the circumstances indicates that a compressor with a small amount of cooling is most advantageous. As explained in detail in the Appendix, all of the losses of compression, as well as the theoretical power for compression, are added to the fluid and finally appear as heat causing temperature rise. Usually there is removal of heat due to whatever cooling may occur with the casing area used. However, this is comparatively small as compared with the heat added, for the compressors here considered, so that the temperatures are above the theoretical ones for adiabatic compression instead of below them. With some superchargers there is even unavoidable addition of heat from adjacent hot engine parts, or from the nozzle box in an exhaust-gas-turbine supercharger.

The type of compression here discussed may be called "hyperadiabatic" compression as opposed to the "subadiabatic" compression supposed to occur in some types of compressors. (As an actual fact, there often have been illusions regarding the amount of heat removed when subadiabatic compression is expected. Many an indicator card of a reciprocating compressor shows a compression line well below the adiabatic and approaching the isothermal line, not due to efficient cooling but to piston and valve leakage during compression.)

The removal or addition of heat during compression changes the volume of the fluid being compressed. The power for compression depends largely on the actual volume, so that, as already mentioned, fluid compressed to a given pressure will require somewhat more power if the volume is increased during compression than if it is decreased. Therefore the power depends directly on the temperatures which occur during compression, and these depend directly on the facilities or lack of facilities for removing heat. This is an entirely separate consideration from the shaft efficiency e with which the compression is carried on. If compression with certain existing temperatures attained no matter how and requiring addition or removal of a definite amount of heat, were carried on without losses of any kind, there would be needed P_n the theoretical power for compression with existing temperatures, values for which are given in the compressor formulas of Table 1. This amount of power is therefore the criterion for a perfect compressor operating with the existing temperatures. An actual compressor with definite losses and a resulting shaft efficiency e will require a greater amount of power. However, the standard on which the efficiency is to be based is this theoretical power for compression with

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[1] In charge of Mechanical Engineering Research, Thomson Research Laboratory, General Electric Co. Mem. A.S.M.E. Dr. Moss began his engineering career by serving a four years' apprenticeship as a machinist in San Francisco, Calif., afterward spending some years as a draftsman. He entered the University of California and was graduated from the College of Engineering in 1896, afterward receiving a Master's Degree from the University of California and a Doctor's Degree from Cornell University. This college work was interspersed with professional work as a draftsman and engineer. He became associated with the General Electric Company in 1903 and has since been engaged on mechanical engineering research work connected with gas turbines, steam turbines, centrifugal compressors, and superchargers for internal-combustion engines. He has been granted a number of patents for apparatus produced by the General Electric Company.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

existing temperatures, or theoretical power for hyperadiabatic compression, and not the power for adiabatic or isothermal compression. In order to ascertain whether or not a given compressor is good or bad, from the point of view of efficiency, we must therefore find the actual temperatures which occur during compression regardless of the manner in which these temperatures are attained, which fixes P_n, the theoretical power when these temperatures exist. Of course, the losses which occur during the compression, the theoretical power for hyperadiabatic compression, the amounts of heat removed by casing cooling, and

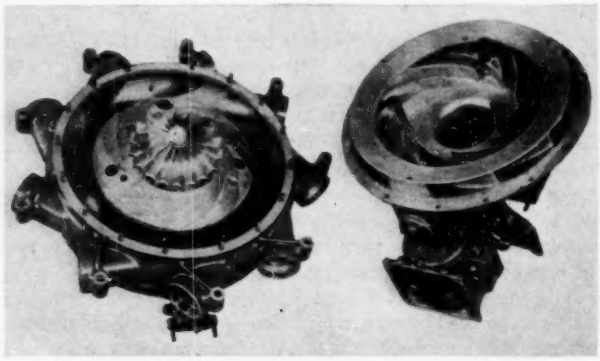


FIG. 1 BUILT-IN GEARED SUPERCHARGER IN PRATT & WHITNEY RADIAL AIR-COOLED AIRPLANE ENGINE. SPEED, 14,000 to 33,000 RPM

added by hot engine parts or the like, are all factors in determining the temperatures during compression. However, as far as the power for hyperadiabatic compression is concerned, the way in which the temperatures are attained is immaterial, as they are fixed by considerations external to the compressor itself and largely independent of it. The temperatures having been fixed, there results a theoretical power for hyperadiabatic compression which is the standard by which the compressor must be judged. To state this matter in another way, the compressor handles fluid with certain volumes, corresponding to certain existing temperatures, and the power which is theoretically required to handle fluid with such volumes is the criterion on which to base the shaft efficiency, not the power for adiabatic or isothermal compression, where the volumes would be less than in the existing case.

However, if it is desired to determine the effect of a change in internal design of a given type of compressor, the amount of cooling remaining fixed, the theoretical power for adiabatic compression would be a logical standard on which to base the comparison. This gives a somewhat better idea of the relative merits of two designs under such conditions, since the losses are added as heat and the shaft efficiency based on existing temperatures would not sufficiently penalize the more inefficient machine. The ratio of theoretical adiabatic power to shaft power is called adiabatic shaft efficiency.

Furthermore, while for most purposes the shaft efficiency rather than the adiabatic shaft efficiency is of importance to the designer, the latter may be of interest from a commercial standpoint. If the user of a compressor does not need high temperature of the discharged fluid, he wishes a machine which will deliver a given pressure with a low power input, and whether a gain comes from better cooling or improved design is immaterial to him. The adiabatic shaft efficiency is a measure of the combined gain due to both factors.

This matter of computation of power for existing temperatures is well known, on the basis of an equation of the compression curve, $pv^n = c$. The general formula for theoretical power for compression with such compression curve with the general exponent n often has been given in the exact form presented later. It usually has been applied to the case here called subadiabatic compression where n lies between 1.00 and k or c_p/c_v. However, the same formula is shown to apply to the case of hyperadiabatic compression with n greater than the adiabatic value k or c_p/c_v. Hence the theoretical power for compression with existing temperatures is the same thing that has been called theoretical power for polytropic compression, or theoretical power for compression with exponent n. For the present case, with temperatures above adiabatic, whence n is greater than k, it is called the theoretical power for hyperadiabatic compression.

There are therefore two entirely independent variables which fix the performance of a hyperadiabatic compressor, one relating to the heat removal and the temperatures which occur during the compression and which fix the theoretical power for hyperadiabatic compression, and the other relating to the shaft efficiency with which the actual power input is applied to supply this theoretical power for hyperadiabatic compression. It is always understood, as already mentioned, that all of the losses due to the fact that the shaft efficiency is less than 1.00, are added to the fluid and so form one element in the heat which is added during compression and which contributes to fix the temperatures during compression. The bearing friction is always negligible for the compressors here considered. Sometimes the fluid passes around the bearings and absorbs their losses. In any case, bearing loss not absorbed by the fluid is neglected.

The ratio of the theoretical power for hyperadiabatic compression to the actual shaft input is defined as the shaft efficiency, denoted by the symbol e. The better the blade and diffuser

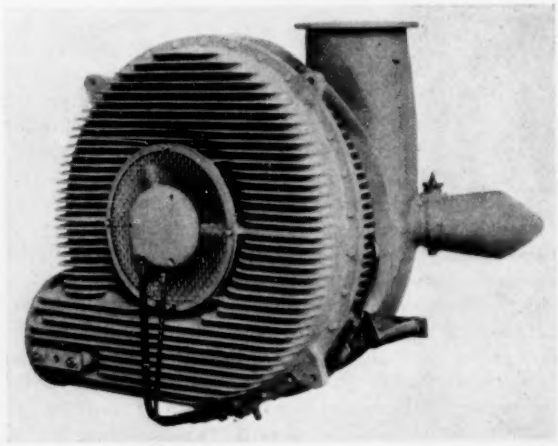


FIG. 2 EXHAUST-DRIVEN SUPERCHARGER USED BY U. S. ARMY AIR CORPS

angles, shapes, and passages, and the less the rotation losses, the greater the value of e.

The temperatures during compression may be taken account of in several ways. The most common one is the use of the value of the exponent n in the relation between pressure and volume during compression, $pv^n = c$. As is well known, n is 1 for isothermal compression, about 1.4 for adiabatic compression, and 1.4 to 1.7 for the hyperadiabatic compression here discussed. Another way in which the temperatures during compression may be taken account of is by the value of the ratio, called e_n, of the theoretical power for hyperadiabatic compression, P_n, to the net power added in the form of heat during compression, causing the temperature rise, P_T. This net power P_T is the shaft power P_s which gives the gross power added in the form of heat, minus the power P_R corresponding to cooling. Of course, the power corresponding to any addition of heat from hot engine

parts or the like also must be counted. This ratio e_n is called the exponential efficiency, and a very simple mathematical expression for it is deduced later

$$e_n = \frac{P_n}{P_T} = \frac{(k-1)/k}{(n-1)/n}$$

e_n is a direct function of n and the relation between the two is shown graphically by the lower curve of Fig. 5. The value of e_n occurs frequently in the mathematics of hyperadiabatic compression, and this is the principal reason why it is given a name.

If the cooling during compression is such as to give adiabatic temperatures, P_n becomes P_k, the theoretical power for adiabatic compression. P_T, the power corresponding to the temperature rise of the fluid, also becomes P_k since we have adiabatic temperatures. Therefore, for cooling to adiabatic temperatures, e_n becomes 1.00. For no cooling at all, P_T becomes P_s, the shaft power, and e_n becomes e, the shaft efficiency. The exponential efficiency e_n therefore varies from 1.00 with cooling to adiabatic temperatures, to e with no cooling, and the actual intermediate values in any given case denote how much cooling is accomplished. e is about 0.70 for a usual case, with a maximum of about 0.75 when circumstances are favorable, and values down to about 0.65 when compromises have to be made in the design. These are therefore the values of e_n with no cooling. In designing a hyperadiabatic compressor for a given pressure rise and flow, values for e_n or n between the adiabatic case and the case for no cooling may be estimated from experience. There can be no general relation between speed, capacity, etc. which serves to fix e_n or n, since they depend primarily upon the heat dissipated from the casing. Values may be computed by finding a tentative value of P_s, and a tentative value of P_R from an estimate of the casing temperature and Robinson's paper mentioned later. Then

$$e_n = \frac{e}{1 - P_R/P_s}$$

and

$$n = \frac{e_n}{e_n - (k-1)/k}$$

P_R need not be known very exactly to give a good value of e_n. A second approximation may be made if desirable by using the first value for e_n to compute n, P_n, P_s, and T_1, from which a second estimate of the casing temperature may be made and a second value of P_R computed.

Various other functions of n or e_n give other aspects of the amount of cooling, and some of these are listed in line IV, column 3, of Table 1. However, no matter how we express the amount of cooling which occurs during compression, there are two entirely independent variables, one such as the exponential efficiency e_n, giving the amount of heat removal or addition with given losses, and in some way fixing the temperatures and hence P_n the theoretical power for hyperadiabatic compression; and the other, the shaft efficiency e, fixing the efficiency with which the shaft input P_s is applied to supply this theoretical power P_n. Of course, the higher the exponential efficiency e_n, indicating better cooling, the lower will be P_n, but for given temperatures P_n has a given value which is the ideal so far as shaft efficiency is concerned. It is by keeping clearly in mind these two distinct variables e and e_n that we shall best be able to attain the best values for each of them, and so obtain the best compressor. The shaft efficiency e is to be made as great as possible by properly propor-

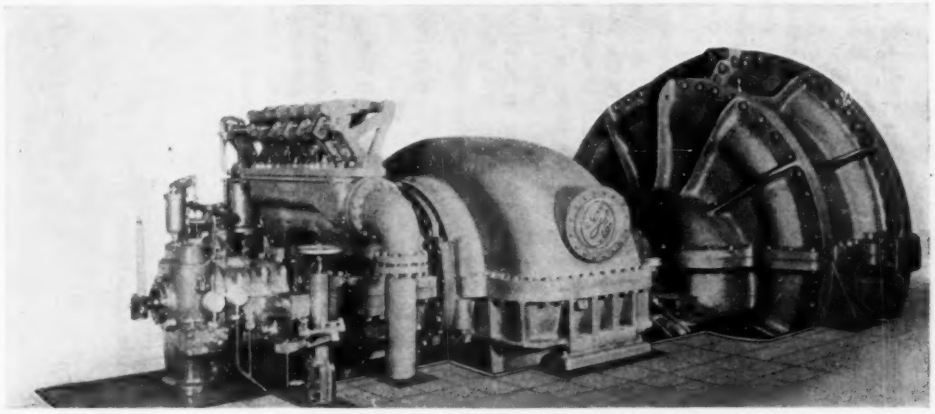


FIG. 4 CENTRIFUGAL COMPRESSOR (WITHOUT WATER COOLING) FOR BLAST-FURNACE SERVICE

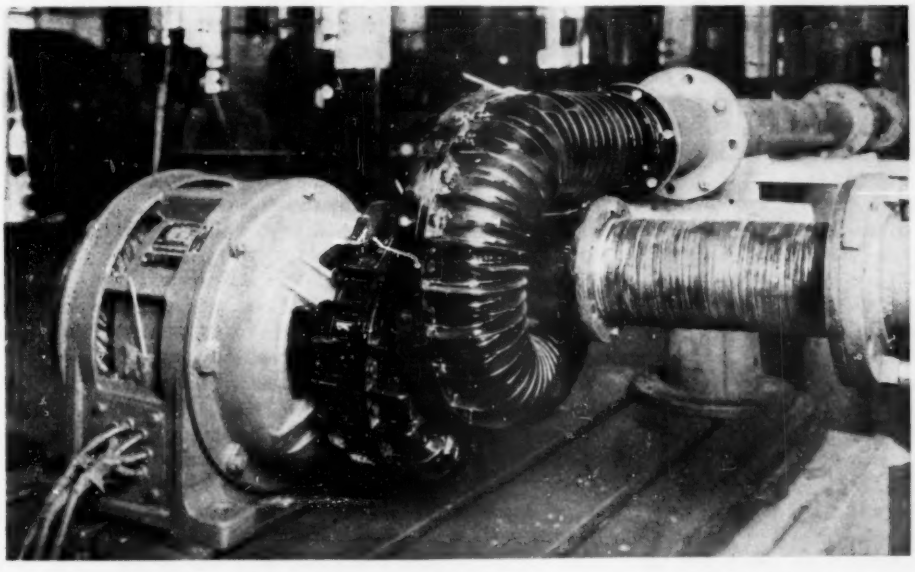


FIG. 3 MOTOR-DRIVEN CENTRIFUGAL COMPRESSOR FOR USE IN SEWAGE DISPOSAL
(Pressure, 8 1/3 lb per sq in.; capacity, 1000 cfm; speed, 18,800 rpm.)

TABLE 1 COMPRESSOR FORMULAS WITH SPECIAL REFERENCE TO HYPERADIABATIC COMPRESSION

(The values are for any perfect gas unless otherwise specified)

Formula	(1) Isothermal Compression. Cooling which removes theoretical power for compression as well as losses so that there is no addition of heat	(2) Adiabatic Compression. Cooling which exactly removes losses, or perfect compression without loss	(3) Hyperadiabatic Compression with cooling or addition of heat. The temperatures are above adiabatic and below or above those of the next column according to the amount of cooling or addition of heat	(4) Hyperadiabatic Compression without cooling or external addition of heat. The temperatures are those due to addition of heat corresponding to entire shaft power which includes theoretical power for hyperadiabatic compression as well as losses due to a given shaft efficiency e. This is the case where the surface is so small in comparison with the power that cooling or addition of external heat is negligible. For the case where such cooling or addition is comparatively small, the given formulas may be used as approximations
I—Theoretical power for compression with existing temperatures, expressed as horsepower	$P = 0.004364 p_2 q_2 \times (\log_e p_1 - \log_e p_2)$ $(0.004364 = 144/33000)$	$P_k = \dfrac{0.004364 p_2 q_2 Y}{(k-1)/k}$ For air and other diatomic gases this becomes $P_k = 0.01542 p_2 q_2 Y$	$P_n = 0.004364 \dfrac{T_1 - T_2}{T_2} \dfrac{p_2 q_2}{(n-1)/n}$ For a design where e, n, and e_n are known, $P_s = P_n/e$. For a test, P_s may be observed or may be found from the computed power P_T corresponding to the temperature rise and the computed casing cooling P_R, by the following formula: $P_T = 0.004364 \dfrac{(T_1 - T_2)}{T_2} \dfrac{p_2 q_2}{(k-1)/k}$ $P_s = P_T + P_R$ n and e_n are found from the observed temperatures and the formulas below Then $e = P_n/P_s$	$P_n = 0.004364 \dfrac{T_1 - T_2}{T_2} \dfrac{p_2 q_2}{(n-1)/n}$ $P_T = 0.004364 \dfrac{T_1 - T_2}{T_2} \dfrac{p_2 q_2}{(k-1)/k} = P_s$ For a design e is known and the temperatures are computed from the formula for e below. For a test the temperatures are observed
II—Theoretical energy for compression, ft-lb per lb of fluid	$w = J c_p T_2 \dfrac{(k-1)}{k} \times (\log_e p_1 - \log_e p_2)$	$w_k = J c_p T_2 Y$	$w_n = J c_p e_n (T_1 - T_2)$	$w_n = J c_p e_n (T_1 - T_2)$
III—Relations between initial temperature T_2 and final temperature T_1 abs F	$T_1 = T_2$	$T_1 = T_k$ = final temperature with adiabatic compression $\dfrac{T_k}{T_2} = \left(\dfrac{p_1}{p_2}\right)^{(k-1)/k} = Y + 1$	$\dfrac{T_1}{T_2} = \left(\dfrac{p_1}{p_2}\right)^{(n-1)/n} = Y_n + 1$ $\log T_1 - \log T_2 = \dfrac{n-1}{n} (\log p_1 - \log p_2)$ $n = \dfrac{1}{1-(n-1)/n} = \dfrac{e_n}{e_n - (k-1)/k}$ For a design n is known and T_1 is computed. For a test T_1 is observed and n is computed	Use formulas of preceding column. For a design, e is known and $(n-1)/n$ is computed from the formula for e below. Then T_1/T_2 and Y_n may be computed from the formulas of the preceding column. For a test $(n-1)/n$ is computed from the temperatures and the formulas of the preceding column and e is computed from the formula below

IV—Efficiencies and ratios

Exponential efficiency = $\dfrac{P_n}{P_T} = \dfrac{(k-1)/k}{(n-1)/n}$

$= \dfrac{k-1}{k}\dfrac{\log p_1 - \log p_2}{\log T_1 - \log T_2}$

Formulas for P_k and T_k in column 2.

Adiabatic power = Theo. power with exist. temps. $\dfrac{P_k}{P_n} = \dfrac{1}{e_n}\dfrac{Y}{Y_n} = \dfrac{1}{e_n}\dfrac{T_k - T_2}{T_1 - T_2}$

Adiabatic temp. rise = Existing temp. rise $\dfrac{Y}{Y_n} = \dfrac{T_k - T_2}{T_1 - T_2} = \dfrac{T_k/T_2 - 1}{T_1/T_2 - 1}$

Adiabatic power = Shaft power $= e e_k = \dfrac{P_k}{P_s}$

Fraction of shaft power removed by cooling = $\dfrac{P_s - P_T}{P_s} = 1 - \dfrac{e}{e_n}$

Fraction of eddy and friction losses removed by cooling = $\dfrac{P_R}{P_L} = \dfrac{P_s - P_T}{P_s - P_n} = \dfrac{1 - e/e_n}{1 - e}$

For a design, $e_n = \dfrac{e}{1 - P_R/P_s}$

Shaft efficiency = $e = \dfrac{P_n}{P_s} = \dfrac{(k-1)/k}{(n-1)/n}$

$= \dfrac{k-1}{k}\dfrac{\log p_1 - \log p_2}{\log T_1 - \log T_2}$

$n = \dfrac{e}{e - 1 + 1/k}$

e is given by e_n of the preceding column. Use formulas of preceding column, substituting e for e_n

tioning vanes, areas, angles, and other items of the design. The cooling is to be made as great as possible to decrease the theoretical power for hyperadiabatic compression, as well as to decrease the final temperature in case it is desirable to have the final fluid as cool as convenient. In case of a supercharger some temperature rise assists vaporization, but after this, the cooler the air, the greater the amount of charge forced into the engine cylinder. It is easily possible for a supercharger to nearly defeat the purpose for which it is installed if the final temperature is so high that the weight of charge in the given volume of the cylinder displacement is unduly decreased. This occurs if the efficiency is unduly low, and if heat is added instead of being removed during compression. Therefore the supercharger casing should be separated as far as possible from the engine crankcase or parts heated by hot oil, and protected by baffles from radiation from a hot nozzle box. Cooling fins, as shown in Fig. 2, should be provided, and whatever arrangements are possible should be made for circulation around these fins.

In case the compressor speed and power input are so great and the casing area so small that the cooling is a negligible percentage, we have the case of column 4 of Table 1, with no cooling or heating, so that the power P_T corresponding to the temperature rise gives the shaft power P_s, and the exponential efficiency e_n has its minimum value, which is the shaft efficiency e. A test of such a compressor with measurement of temperatures and pressures only thus gives the value of shaft efficiency without necessity for power measurement. This sort of test is so simple that it has been used in cases where appreciable dissipation of heat from the casing makes it improper, and shaft efficiency has been computed from a test on the basis that the shaft power was given by the temperature rise of the air, when really the latter was appreciably less, owing to heat dissipation from the casing. This amounts to using our exponential efficiency e_n as if it were shaft efficiency for this case. An example of the error is shown in Fig. 7 where some points have a shaft efficiency of 0.70 and an exponential efficiency of 0.78. If the value of 0.78 is supposed to be the shaft efficiency, the compressor is given much more credit than it deserves. A demonstration of the fallacy of neglecting heat dissipation from the casing is the fact that some centrifugal compressors whose efficiency was computed by temperature rise, operated at no load without much increase of the casing temperature. In this case the entire shaft power at no load, about 20 per cent of the full-load power, was dissipated from the casing because there was no flow of fluid. An amount of power of the same order of magnitude was of course dissipated at full load, and there was therefore appreciable heat dissipation. However, owing to the use of high speeds, casings of small area cannot safely dissipate the no-load power, and machines of the type of Figs. 2 and 3 become unduly heated if operated at no load by completely closing the inlet for any length of time.

The shaft power P_s for a test may be accurately computed by adding the power P_R removed by cooling, computed by methods outlined by Robinson,[2] to the power P_T shown by the measured temperatures of the air.

Fig. 7 shows the application of the preceding principles to a test at sea level of a particular airplane supercharger. Test I was first made with the entire casing covered with heat-insulating lagging so as to make the heat dissipated from the casing negligible, as in column 4 of Table 1. Then $e = e_n$, and the shaft power and shaft efficiency are given by measurement of the temperature rise of the air. This method of measurement of shaft power and shaft efficiency has been used in a number of cases and has been found to be trustworthy when the temperature rise is great enough so that it may be determined with accuracy. The lagging was then removed and the run marked IV (Fig. 7) made with usual cooling. The interior of the machine was the same as in test I, so that the shaft efficiency was the same for both tests and was given by e_n for Test I.

Fig. 7 shows values for the various efficiencies and factors. It is to be noted that the maximum cooling which can be provided gives an appreciable decrease of temperature rise, but does not do very much toward decreasing P_n, the power for hyperadiabatic compression. However, all of the cooling has been done which seems possible with the small, compact machine which the conditions demand. The power actually delivered during the test at the larger volumes is 117 hp with a casing of 20 in. outside diameter which does not permit of an appreciable percentage of cooling.

Notation for Formulas

v = specific volume

p_1 = high pressure, lb per sq in. abs. This is the final pressure for a compressor

p_2 = low pressure, lb per sq in. abs. This is the initial pressure for a compressor

[2] "Heat-Balance Method of Testing Centrifugal Compressors," Trans. A.S.M.E., vol. 47 (1925), p. 1179.

T_1 = high or final temperature, abs F
T_2 = low or initial temperature, abs F
T_k = high or final temperature for adiabatic compression
q_2 = flow, cu ft per min at inlet conditions p_2, T_2
k = c_p/c_v = ratio of specific heats. $(k-1)/k$ is taken as 0.283 for air and perfect diatomic gases
n = exponent in $pv^n = c$
Y = $(p_1/p_2)^{(k-1)\ k} - 1$
Y_n = $(p_1/p_2)^{(n-1)/n} - 1$
P_s = shaft horsepower

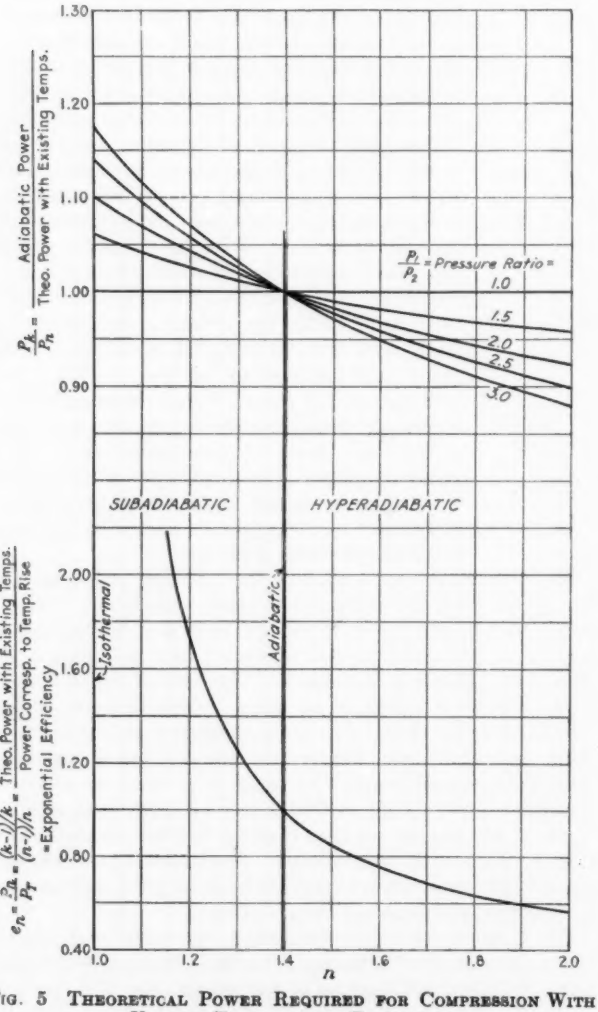


FIG. 5 THEORETICAL POWER REQUIRED FOR COMPRESSION WITH VARIOUS TEMPERATURE CONDITIONS

P_n = theoretical power for hyperadiabatic compression or compression with existing temperatures or with exponent n, expressed as horsepower
P_R = cooling or radiation from the compressor casing, expressed as horsepower. It may be computed from the temperature and area of the casing if there is no possibility of addition of heat as from hot engine parts, nozzle box, etc.
P_k = power for adiabatic compression
P_L = eddy and friction losses of compressor, as horsepower
P_T = horsepower corresponding to temperature rise of the air
w_n = theoretical energy for compression with existing temperatures, ft-lb per lb of fluid
w_k = energy for adiabatic compression
e = P_n/P_s = shaft efficiency
e_n = $\dfrac{P_n}{P_T} = \dfrac{(k-1)/k}{(n-1)/n}$ = exponential efficiency. This is the ratio of differential temperature increments, being that for adiabatic temperatures divided by that for existing temperatures. It also comes out as the ratio of the theoretical power for hyperadiabatic compression to the power corresponding to the temperature rise
e_k = adiabatic power divided by theoretical power for hyperadiabatic compression. $e_k = w_k/w_n = P_k/P_n$
η = hydraulic efficiency, or ratio of theoretical energy for hyperadiabatic compression required for the pressure produced, to the theoretical energy added to the fluid by virtue of the wheel angles and speed.

In order to determine the size of an impeller to produce a given pressure, there must be known from experience the hydraulic efficiency η, which is the ratio of w_n the theoretical energy for hyperadiabatic compression to the theoretical energy input for a given wheel speed. The latter expression has the form KV^2/g, where V is the peripheral speed of the impeller in feet per second, and K is a constant dependent upon the directions of the fluid, in space at the impeller inlet, and relative to the impeller at the exit. The values of w_n given by Formula II are to be inserted in the relation

$$\frac{KV^2}{g} = \frac{w_n}{\eta}$$

In a usual supercharger, the fluid enters the impeller with its absolute velocity in space in a direction normal to the surface of the inlet edges of the impeller vanes, and the vanes at impeller exit give the fluid a velocity relative to the impeller which is normal to the surface of the exit edges of the impeller vanes. For this case $K = 1.00$ in the above equation. Under some circumstances the fluid at the impeller exit does not follow

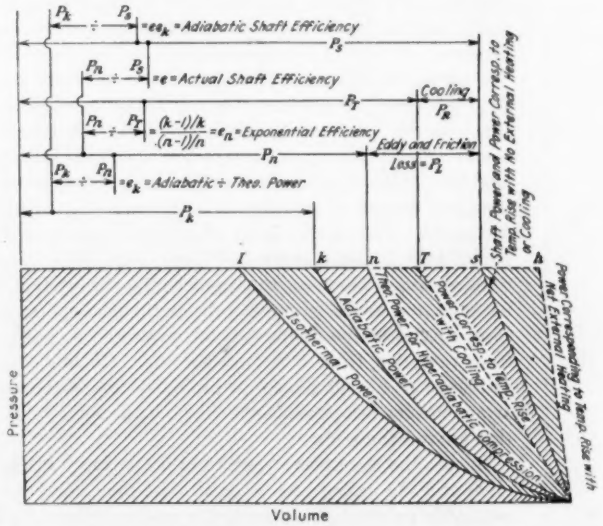


FIG. 6 PRESSURE-VOLUME DIAGRAM FOR VARIOUS TYPES OF COMPRESSION

(This figure is purely diagrammatic and not drawn to scale. Dimension lines refer to area of diagram included between the two sloping or vertical lines indicated. The area between k and n shows the extra power which must be supplied because the cooling provided is not sufficient to give adiabatic temperature.)

the blade angle, and a factor from experience must be introduced to take account of this effect, called "slip." Sometimes it is most convenient to refer to w_k, the adiabatic energy. Both

purposes may be accomplished at once by using another factor from experience, defined as follows:

$$\text{Compression coefficient} = \frac{w_h}{V^2/g}$$

Appendix

DEDUCTION OF FORMULAS

SUPPOSE a pound of fluid passing from a region A, such as the atmosphere, through a region C, such as a compressor, where mechanical energy, heat, etc. may be transferred, and into a region D, such as the discharge pipe of the compressor; with conditions p_2, v_2, and T_2 at the junction of A and C and p, v, T at the junction of C and D. Let $U - U_2$ ft-lb be the change in the molecular energy resident in the fluid from A to D. The passage from A to C means that energy 144 p_2v_2 is added to C by A, and similarly energy 144 pv is added by C to D. Hence the net energy added to the fluid during its passage through C is

$$(U + 144\,pv) - (U_2 + 144\,p_2v_2)$$

Of this added energy, only a portion $U - U_2$ remains in the fluid, and the remainder is added to the external universe, but nevertheless the whole expression represents energy which must be added when the fluid passes through C. The value in each parenthesis, counted above some suitable zero, is well known as the "total heat" or "enthalpy" for any fluid. Air and similar gases, for engineering purposes, may be treated thermodynamically as perfect gases. For these it is well known that no energy is involved when the molecules are brought closer together as at the higher pressure, since even then they are so far apart that their mutual attraction is negligible. Hence a given weight of gas at a given temperature contains a given amount of internal energy, regardless of the pressure. It follows that the heat equivalent of all of the mechanical energy actually added to the fluid causes temperature rise just as if it were added directly as heat without causing any change of pressure. Energy in ft-lb per lb added to cause temperature rise of fluid in a closed vessel is expressed by

$$J\,c_v(T - T_2)$$

and this is therefore the value of $(U - U_2)$ for a perfect gas. By substituting this value in the above expression and also making use of the well-known relations for a perfect gas, 144 $pv = RT$, and $c_p - c_v = R/J$ (which is really the statement that the internal energy depends on the temperature only), we have as the value of the above expression for the energy which must be added to the fluid either as heat or mechanical energy as it passes through the compressor

$$w_T = Jc_p\,(T - T_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

The mechanical energy for compression is applied to the fluid in a centrifugal compressor in a way somewhat different from what it is in a reciprocating compressor, although of course the result is the same. At any point in the compression, the volume v passes from a region of pressure p to a region of pressure $p + dp$. The mechanical energy in ft-lb per lb which must be supplied by the compressing machine to accomplish this is 144 vdp.

The total energy for compression from p_2 to p, therefore, is

$$w_n = 144 \int_{p_2}^{p} vdp \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

This is obviously the area of the pressure-volume diagram, Fig. 6, and vdp is the area of a differential horizontal strip. Such a diagram is also the "indicator card" for a reciprocating compressor. If the energy added to the fluid as given by [1] is the exact amount of the mechanical energy given by [2], so that no heat as such is added or taken away, we have what is called

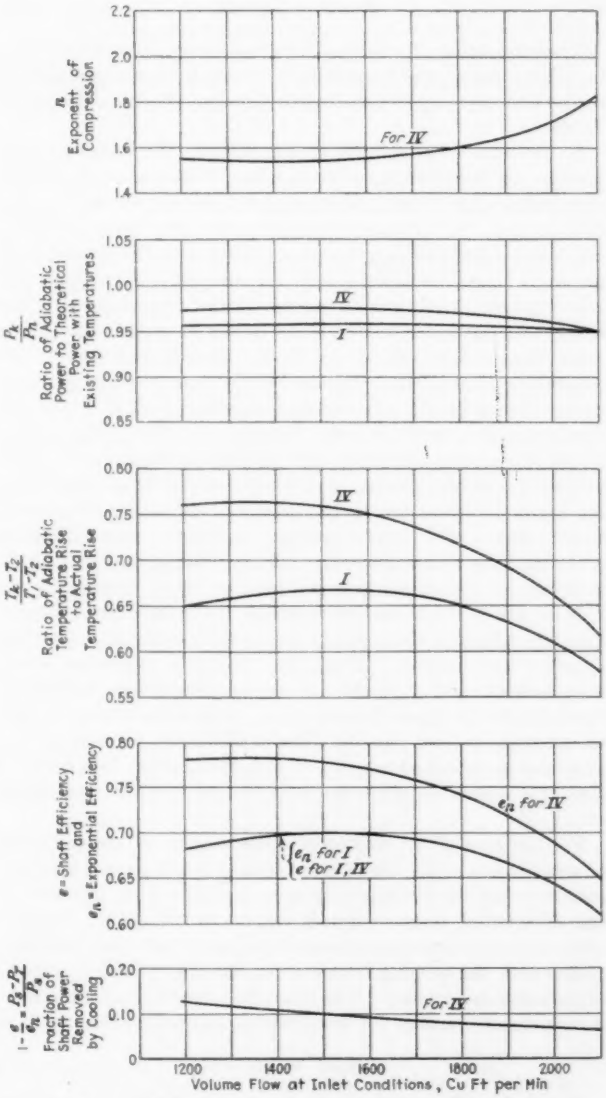


FIG. 7 EFFICIENCIES FROM SUPERCHARGER TESTS
(I—Casing lagged with asbestos. IV—Casing bare and cooled.)

"adiabatic compression," defined by the equality between [1] and [2]

$$Jc_p\,(T - T_2) = 144 \int_{p_2}^{p} vdp \quad \ldots\ldots\ldots\ldots [3]$$

Such a thermodynamic change, without external addition or subtraction of heat, is called adiabatic in the general case of expansion where there may or may not be some internal transformation of mechanical energy into heat, as by energy loss due to fluid friction. The particular case of adiabatic expansion with no internal transformation of mechanical energy into heat is called "isentropic," and this term could properly be used for the same type of compression, which is the present case. However, the reverse of the general case of adiabatic expansion would be compression with internal transformation of heat into mechanical

energy, which is not possible. Hence adiabatic compression can only be isentropic compression. As the term "adiabatic" is very widely used, the author will adhere to it. Equation [3] gives as the differential equation for adiabatic compression

$$Jc_v dT = 144\, vdp \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

The integration of this equation gives the well-known relations between pressure and temperature for adiabatic compression (of course also applying to isentropic expansion) given in column 2 of Table 1.

In the case of a centrifugal compressor, the losses of compression are due to eddies at blade inlets, friction along the blades of impeller and diffuser, and rotational friction of the impeller in its casing. All of these, which may be called hydraulic losses, are added as heat, causing temperature rise of the fluid being compressed. Since the useful energy for compression, $\int vdp$, is also added as heat, it follows that in a centrifugal compressor all of the energy corresponding to the shaft power P_s finally appears to cause temperature rise of the fluid. That is, the energy corresponding to the losses plus the energy shown by [2] gives the energy shown by [1]. In the mathematical work it is assumed that there is a constant ratio between the differential energy added to cause compression, the differential heat added or removed by external sources, and the differential heat added from the hydraulic losses during compression. Of course, this is not strictly true. The heat removals, additions, and efficiencies probably have values somewhat different during that part of the compression occurring along the impeller from those along the diffuser, etc. The losses due to eddies at the impeller or diffuser inlet are added at these points rather than uniformly along the impeller and diffuser. There are no data regarding these irregularities, and it would not be mathematically feasible to take account of them if there were. Hence the observed final temperature is used and the differential energy changes due to all causes are assumed to proceed proportionately. It is believed that this procedure gives the final results without appreciable error.

For the case of hyperadiabatic compression, where the losses of compression minus the effect of cooling give a greater temperature rise than for the adiabatic case, we define e_n as the ratio of the differential temperature increment which would occur if there were adiabatic compression, to the increment which actually occurs with the existing temperature conditions, taking account of the losses and cooling. The adiabatic increment corresponds to the mechanical energy for compression, 144 vdp, so that we have

$$e_n Jc_v dT = 144\, vdp \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

This is the differential equation for hyperadiabatic compression corresponding to Equation [4] for adiabatic compression. The integration is readily performed and gives

$$p_1 v_1^n = p_2 v_2^n \qquad \text{and} \qquad \frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{(n-1)/n}$$

where n is defined by the relation

$$e_n = \frac{(k-1)/k}{(n-1)/n}$$

This assumption of a constant ratio e_n of adiabatic to existing differential temperature increments therefore gives final results identical with the well-known formulas for polytropic compression with exponent n. The energy for hyperadiabatic compression comes out

$$w_n = 144 \int_{p_2}^{p_1} vdp = e_n Jc_p (T_1 - T_2)$$

The energy corresponding to the temperature rise is

$$w_T = Jc_p (T_1 - T_2)$$

It follows that

$$\frac{w_n}{w_T} = \frac{P_n}{P_T} = e_n$$

which gives a definition of e_n which may be used instead of the original one. From the relations above given may be deduced the various relations of column 3 of Table 1. If there is no cooling, P_T becomes P_s and e_n becomes P_n/P_s, which is defined as the shaft efficiency e. This gives the formulas in the last column of the table.

In the transformations involved in the various formulas there is often used the well-known relation

$$c_p - c_v = c_p(k-1)/k = R/J = \frac{144pv}{JT}$$

where R is the constant in the relation $144pv = RT$ with p in pounds per square inch. The fundamental constants should be selected so that they are reputable in view of modern experimental work, as well as to exactly satisfy this equation.[3]

Discussion

C. F. Taylor.[4] The writer is glad that Dr. Moss has presented the theory of air compression under high-temperature conditions in such a clear and useful form. While this theory has been well known, this is the first time that the writer has seen it presented for publication in a form so convenient for the engineer.

There is one point to which attention might be called, and that is that all the calculations and formulas are based on the assumption that the initial volume of air, per unit time, remains constant.

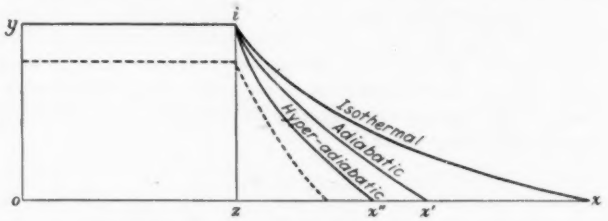


Fig. 8

In the case of the internal-combustion engine, however, it is the final volume rather than the initial volume which is held constant, assuming the engine speed not to vary.

Referring to Fig. 8, let us assume that the cylinder volume of the engine per unit time is represented by the length *o-z*. The power for isothermal compression would be represented by the area *o-x-i-y*. Now if the temperature increases during the compression process, and we compress to the same final pressure, the initial volume must change in order to attain the same final volume. In such a case the power for compression is reduced, and is represented by the areas *o-x'-i-y* and *o-x''-i-y* for adiabatic and hyper-adiabatic compressions, respectively. Thus the increase in compression temperature actually reduces the power absorbed by the supercharger, under the usual conditions obtaining in the internal-combustion engine. As a matter of fact, the case is usually even more exaggerated, because when the temperature of the air delivered to the engine increases, the pressure to which supercharging may be carried is often reduced on account of deto-

[3] "Engineering Computations for Air and Gases," by S. A. Moss and C. W. Smith, Trans. A.S.M.E., vol. 52 (1930), paper APM-52-8.

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nation. The actual compression diagram under such conditions would be represented by the dotted line of Fig. 8.

The power for compressing a given *weight* of air to a given pressure, increases with increase in compression temperature, as the author states. From Fig. 8, however, we can easily see that in the case of supercharging the internal-combustion engine the weight of air decreases as the compression temperature increases, to such an extent that the power required by the supercharger goes down rather than up.

In the case of a blast furnace, a constant weight of air is required. This means a constant initial volume, and Fig. 6 would be representative of such an application except that the final pressure would vary with the varying resistance to flow of the furnace charge.

For most commercial uses of compressed air, such as for driving small tools, the requirement is a constant pressure after the air has been stored and cooled practically to its initial temperature. In this case, Fig. 6 would apply.

The Mooring and Ground Handling of a Rigid Airship[1]

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Problems in mooring and handling rigid airships are treated. Differences of opinion resulting from the relatively limited airship operating experience are discussed. Prediction is made that successful mooring and handling of rigid airships will center around the low mast. When the problems of airship terminals reach a satisfactory status, as it is asserted they soon will, it will be found that extremely important contributions will have been made by the airship personnel and engineering talent of the United States.



EVERY successful type of carrier requires extensive auxiliary facilities to permit it to serve usefully and efficiently. Such facilities have seldom been anticipated or originally conceived with the carrier, but are achieved through progressive development after extensive and sometimes expensive practical experience has been gained. Even a hasty glance at the histories of railroads, steamships, and automobiles reveals the multitude of essential auxiliaries such as servicing stations, roundhouses, repair shops, elaborate signal systems, communications, tunnels, drydocks, wharves, bridges, fleets of tugs, and so on, of fundamental and indispensable need. Furthermore, automobiles and railroads have required hundreds of thousands of miles of elaborately prepared paths for their use. In other similar essentials and except for the matter of prepared paths, aircraft, and particularly airships, are at present in the status analogous to that of other successful carriers in their pioneering days when their auxiliaries were just being found necessary and accordingly began to be developed. The greatest need today in the airship project is that of providing mooring and ground-handling equipment commensurate with the ships themselves.

It is to be regretted that the development of mooring and handling methods and equipment did not proceed apace of airships themselves, as probably no other single factor has been so effective in retarding the airship project as the lack of adequate and efficient terminal facilities. This is not an occasion for citing the reasons why such development has lagged, but suffice it to say the situation has been realized in this country from the start of rigid-airship operation here; and as a result of determined steps along this line, highly improved equipment is now in use and further continuous progress is being made. It is the opinion of those of us who have been and are now engaged in the operation of rigid airships that the attainment of the full potentialities of this remarkable carrier is dependent on and will be materially expedited principally by the solution of the mooring and handling problems. Most important of all, it is our conviction also that these problems are not at all insurmountable, but are capable of solution upon the expenditure of suitable effort.

In taking up airship operation in this country we were forced to begin with inherited methods of operation, and naturally were largely dependent on manual methods for the landing and ground-handling of our ships. The several years of operation of the *Shenandoah* and the *Los Angeles* taught us much. With only one active airship, progress has necessarily been slow, but we are now more fully cognizant of the requirements and specifications for more generally applicable mooring and handling methods. We have produced a mechanical handling system for the *Los Angeles* whereby that ship has been handled in and out of the Lakehurst shed with as few as 60 men. Further improvements are necessary and are on their way to early realization.

Likewise we have realized the weak points in the construction details of past airships which have impeded the application of mechanical-handling methods to them. Future ships will incorporate features for making ready application of improved handling equipment. The old manual methods, however, involve the basic fundamentals and have been valuable in impressing the principles involved. These manual methods will probably for some time continue to be available for emergency or last-resort use just as hand-steering equipment is still available in surface vessels. Modern airships are of such size as to preclude regular dependence upon man power for ground handling, and if a suitable mechanical equivalent is not found, it will be very difficult indeed for the airship project to succeed.

The main problems involved in providing adequate airship terminals (exclusive of repair facilities) may be classed under three headings:

(1) Landing the airship from flight by mechanical means
(2) Servicing the ship and mooring it out in the open
(3) Housing and unhousing the ship when necessary, by mechanical means that will reduce the danger and increase the utility of the operation; this includes moving the ship between the mooring location and the dock.

I shall endeavor to point out briefly some of the fundamentals of the remaining problems and then how I believe we can meet them practically.

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[1] The opinions and assertions contained herein are private ones of the writer and are not to be construed as official or as reflecting the views of the Navy Department or the Naval Service at large.

[2] Lieutenant Commander, U. S. Navy, in command of U.S.S. *Akron*, home port Naval Air Station, Lakehurst, N. J. (NOTE: Early in 1932, Lieut.-Com. C. E. Rosendahl was transferred to sea duty on the battleship *West Virginia*.) The author was born on May 15, 1892. He was graduated from the U. S. Naval Academy in June, 1914; served in various line officers' billets in the Navy continuously; in April, 1923, volunteered for airship duty and began training in that branch at the Naval Air Station, Lakehurst, N. J.; was a member of the original crew of U.S.S. *Shenandoah* and a survivor of her loss in September, 1925; then became executive officer of U.S.S. *Los Angeles*, becoming commanding officer of U.S.S. *Los Angeles* in May, 1926, and retaining that command for three years; then became commander, Rigid Airship Training and Experimental Squadron, for a year; about one year in Bureau of Aeronautics, Navy Department; flew trials of U.S.S. *Akron* and became first commanding officer of that ship, relinquishing that command in June, 1932; made first transatlantic crossing of *Graf Zeppelin* in October, 1928, as U. S. Navy observer; made world flight of *Graf Zeppelin* in August, 1929.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

In the development of the improved equipment, we have kept in mind the desirability of combining in each item as many functions as possible and of keeping the number of items small, if such could be done without materially complicating the units involved. While such an objective might at first seem a little ambitious, consideration of the matter shows it to be largely practicable. Experience has shown that a mooring mast having certain essential characteristics can be made to serve as the nucleus for all such systems and equipment.

As is generally known, the mooring mast or tower is a post-war product devised by the British. The main factor contributing to its conception was the vital need for servicing and mooring out airships in the open when weather or other conditions made it impossible to house them. There exists also the additional need for having outlying servicing stations which are not so pretentious or expensive as those possessing hangar and repair facilities, etc. In other words, airships require not only navy yards or docking yards, but harbors as well. The facilities at airship harbors are those comparable to wharves, moorings, and anchorages for surface vessels, including fueling, watering, and provisioning arrangements. For an airship, these will consist of a mooring mast and certain servicing facilities that can be conveniently grouped around it.

During the temporary cessation in the British airship program their progress in mooring masts also was retarded somewhat; but with the revival of their airship activities an elaboration of the original tower was produced, and several examples of this design were constructed throughout the British Empire. During the British inactivity, the U. S. Navy made great progress in mooring masts and expanded their use very materially. The mooring mast is established as a permanent factor in airship operation.

Three general types of masts or mooring towers are recognized:

(*a*) The high mast or tower
(*b*) The low or "stub" mast
(*c*) The mobile mast.

"Semi-portable" and "expeditionary" masts of either the high or stub varieties have been put into use after quick erection at outlying places. The "floating" mast and base, as represented by the installation on the airship tender *Patoka*, have been in use for a number of years.

The low or stub mast is a U. S. Navy development. The high tower is the type developed and still used by the British. No high masts have been erected in the United States for several years, and it is the author's opinion that the high mast is being superseded by the stub mast and is rapidly going out of style. The stub mast is a very versatile member of the airship project, and the principal concern of this paper is with it. To show the trend leading to the stub mast, it is necessary to describe briefly the present means employed for securing a rigid airship to a mast.

The longitudinal members of a ship's structure come together in a strong bow-cap, in which is secured an axial steel tube housing a strong spindle mounted in bearings. A steel mooring cone is mounted on the outboard end of the spindle and hinged by an athwartship pin. At the top of the mooring mast, a steel cup that is the counterpart of the ship's cone is mounted on a radial bearing. When the ship is secured to the mast, the cone is locked in the rotary mast cup, and the ship has three degrees of freedom: she can swing around the mast in the horizontal plane like a weather vane in response to fluctuations in the direction of the wind; her stern can oscillate vertically about the hinge pin of her mooring cone; and she can roll about her longitudinal axis. The original type of mooring mast was of such height that contemporary ships were permitted a downward angle of 12 to 15 deg before the stern was in danger of striking the ground. With modern airships, this freedom would require a structure some 250 ft high. Restriction of the vertical motion of the stern, particularly upward, presented an exceedingly interesting problem. In a wind of appreciable velocity, some dynamic control at the mast is available from the flow of air past the ship's surfaces, and a certain amount of static control, although somewhat sluggish at times, is possible by use of ballast. Due to fluctuations in wind direction, to rapid atmospheric temperature changes, and to gusts and vertical air currents, a ship moored to a high mast is in motion about the mast much of the time and must be "flown" while so moored.

Vertical movement of the stern, under certain conditions, has assumed dangerous proportions. Various efforts at automatic ballasting and trimming have been suggested and some tried, the principle being to permit easy horizontal movement of the ship in azimuth, but to check the vertical oscillations by automatically adding weight during the tendency to rise and by reducing the weight when the ship tends to fall. A seemingly ideal form of tail drag (or automatic weight) is a suspended chain branching into parts and each successive part branching out again, to increase the weight increment with increase of upward angle of the stern. The most practical type consists of one or more sets of heavy wheels that tow on the ground in horizontal motion, but add their restraining weight as the ship tends to lift. For use at the mast of the *Patoka* a string of partially filled floating drums of water serves quite well. However, any form of drag yet devised for use at a high mast has considerable surface friction, with the result that there often occurs a tendency (which can be of dangerous proportions) to "spring" the stern downward. The use of several individual drags presents difficulty in their attachment and detachment under the gusty conditions in which they are most needed.

During the passage of wind shifts in barometric depressions that are sometimes as great as 180 deg in direction, and often accompanied by considerable and almost instantaneous drops in temperature, there may be danger for the ship which is moored at a high mast in adjusting herself to the new wind direction, due to having a temporary false buoyancy caused by lag of her gas-temperature changes and to vertical currents and gusts. Although it might be expected that storm conditions, if any, would be those that would put a moored ship through dangerous contortions, another situation of quite different characteristics actually caused concern over this on at least one occasion. On a peaceful day when the barometric gradient was weak and the wind consequently light and variable, the *Los Angeles* lay moored to the high mast at Lakehurst. Secured to her stern, as had been customary in several years' operations, was a tail drag consisting of chain and heavy wheels weighing altogether about 2500 lb. The mean wind direction had been northwesterly. Early in the afternoon and without warning except for a series of dust whirls seen approaching a short distance away, the sea breeze with noticeably colder air came sweeping in from a southeasterly direction. With this breeze striking the ship directly from astern, and with the buoyancy augmented temporarily by the sudden immersion of the ship in the colder air, the stern began to rise and continued to almost a vertical position. Just before reaching the top of the arc, the ship began to turn about her longitudinal axis and into the new wind direction. Shortly she came to rest on an even keel without any injury to personnel and with practically no damage whatsoever to herself. As a matter of fact, she could have flown at once in total safety. Although this was the most remarkable demonstration of unsuspected strength ever staged by an airship, it can hardly be called a desirable stunt for repetition. I can conceive of storm conditions that might not treat a ship at the high mast so considerately. This performance corroborated our previous views as to this fault

of the high-mast mooring-out system. It is my opinion that proposed elaborate tail drags and automatic ballasting systems cannot adequately correct this fault of the high mast without introducing complications and perhaps other dangers.

While this incident spurred us on to develop the lower type of mast that would permit full control of the stern movement, there were also other important contributory reasons. Included in these are the knowledge that wind velocities nearer the ground are frequently much less than those at the level of the so-called "high" masts; the fact that gustiness is greater a few hundred feet off the ground than at the surface; and that while vertical currents often exist at the level of the high mast they cannot blow into or out of the ground. The lower position should therefore be relatively safer.

The stub-mast equipment, in general, consists of a mast structure carrying a cup into which the nose of the ship is secured and a stern carriage to which the after part of the ship is attached. A path of some sort for the stern carriage is involved. The simplest form of stub mast consists merely of a wood or steel pole carrying on top, above a small operator's platform, an ordinary mast cup. The heel of the mast may be so hinged as to permit lowering it. Suitable steadying guys and erecting gear are provided. Servicing connections for such needs as electricity, fuel, water, and helium are located at the base of the mast. It is interesting to note that the Germans, who for many years had no mooring-mast convictions, adopted this low type as their first mooring-mast venture.

At outlying stub masts, up to the present, the ship has used a pneumatic wheel and a tail drag at the stern. However, at more complete bases we have employed a circular standard-gage railroad track about the mast and a holding-down carriage fitted to this track by means of rollers which prevent its motion up or down. Suitable means have been devised for the attachment of the rear car of the ship to this holding-down carriage. The latter is so installed that in an emergency a take-off is possible by automatically disconnecting from the stern carriage when the ship is free forward; yet under normal conditions the ship responds to fluctuations in wind direction freely, while the stern is positively restrained from vertical oscillations. At first it was believed that any upward vertical load should not be taken by the car itself, and hence the track was made to resist the expected vertical load. However, it has been found by experimentation that it is safe to make the holding-down carriage itself of such weight as to resist most of the tendency toward lifting vertically. Early criticism directed at this restricted stern movement pointed out that the high friction expected from such a device would retard and deprive the ship dangerously of a freedom of movement in azimuth that it should be allowed. Actual experience has shown this fear to be unfounded.

It has been pointed out that the use of a path or track of fixed radius around the stub mast will necessitate such a path for each of the various sizes of airships that might at some time desire to moor there. While this may be true, it is not considered of any serious consequence. The matter of variable height to accommodate ships of various diameters is readily taken care of by incorporating an extensible feature in the mast. A remaining consideration of equal importance is that of standardizing mooring cones and mast cups, and international negotiations bearing on this matter have been under way for some time.

For the moored ship, the low riding position at the stub mast undoubtedly possesses marked superiority over the high mast. However, to the inexperienced or to those who have not studied the matter, there may at first appear to be certain disadvantages connected with mooring or riding to the short mast. Actually, however, in my opinion, the principal feature pointed out by such persons as a disadvantage of the low mast represents a shortcoming common to both types as they exist today. The introduction of the stub mast merely called closer attention to it.

The gist of the mooring operation is of course the pulling of the ship's cone into the corresponding cup at the top of the mast. To accomplish this, the airship about to moor proceeds into the wind to a point to leeward of the mast and drops to the ground the end of a steel cable fastened to a swivel in the ship's cone. This cable is then coupled to a similar cable that has been led from a winch at the base of the mast, up through the mast, and out of the cup to the ground. Operation of the winch in winding in this coupled cable pulls the nose of the ship into the cup. As the wind is seldom steady in direction and velocity, additional wires become necessary. A steel cable called a "yaw" or nose-steadying wire is led from each side of the ship's nose to the ground, these two cables being then coupled to corresponding wires that are led out on the ground at an angle of 60 deg on either side of the wind direction. When at the proper length, these wires prevent lateral motion of the ship and also keep her from overriding and fouling the top of the mast during the mooring operation. The ship is thus held at the apex of a pyramid of wires, the buoyancy of the ship providing an upward force to make the wires taut, the ship's engines and the wind preventing undue forward momentum, and the yaw wires preventing lateral displacement and overriding. We thus establish in space at the ship's nose as rigid a point as we can with wires; with the ship fast to it we then transfer this point to the mast top.

In the past, the anchorages for the ground yaw wires have been fixed, but the use of a rigid track around the mast will make the introduction of floating yaw-wire anchorages easily feasible. This is another advantage of the stub-mast stern-track system that can be well understood when it is realized that if, during the mooring operation, the wind changes direction to any considerable extent, there is danger of having the wires foul the hull of the ship and also of diminishing their usefulness if their anchorages cannot be readily corrected to the new wind direction. Also it has happened on occasions that the wind direction at the levels of approach for landing is diametrically or widely divergent from that on the surface, so that the ship approaching into her own wind direction actually spirals down as she approaches nearer to the surface.

As indicated earlier, the introduction of the stub mooring mast called attention to an important shortcoming in the method heretofore used in mooring rigid airships. Some people believe that due to the lesser height of the low mast there would be additional danger to the ship while mooring because of the closer presence of the ground. This has resulted in considerable attention being directed toward obtaining better vertical control of the ship during the mooring operation. While the necessity for such control may be seen more readily in connection with the short mast, nevertheless an analysis of the whole problem shows unmistakably that adequate vertical control is a necessity at any mooring mast regardless of its height. Since it has been my fortune to make or attempt to make moorings under bad conditions as well as good, my own experience has pointed out forcefully that the mooring system as employed in the past will not suffice. By "bad" conditions is not meant merely high wind velocities, for a steady high wind would facilitate the mooring. It is the gusty irregular wind, involving lateral and vertical disturbances, that complicates the operation.

The mooring operation has been considerably improved by the adoption of higher speed and more responsive winches and of distant control of the mooring winches from the operating platform. However, no mooring method has been able satisfactorily to overcome purely vertical gusts or the vertical components of gusts that would tend to drive the ship downward during the mooring operation. To make either the high or low mast sufficiently

serviceable, it will be necessary to take care of these vertical fluctuations by an addition to the present mooring system. It is only the buoyancy of the ship that now provides the present upward force and makes a fairly rigid pyramid of the mooring wires. In itself, this has been found to be insufficient. There are several other ways of providing a sufficient force, and included among them are the following:

(1) Inclination of the ship to obtain upward dynamic effect
(2) Use of ship's propellers
(3) Vertical thrust from propellers
(4) Excess buoyancy by ballasting
(5) Addition of new means to provide upward force.

Each of these will be discussed briefly, as they apply to one of the very important remaining mooring problems; and frequently engineers in other lines are able to recommend, from the symptoms, cures that their parts of the engineering world may be able to supply.

Every one knows that airplanes and airships obtain lift from air speed and the inclination of their surfaces. An airship such as the *Los Angeles* can, at top speed, obtain a dynamic lift of between 4 and 5 tons. However, since this lift varies as the square of the air velocity, it will be seen that the dynamic lift falls off rapidly at low or moderate speeds, or rather, does not amount to much until quite high velocities are reached. While no one challenges the fact that there is some slight advantage in maintaining the bow-up, stern-down inclination during the mooring operation, nevertheless its value, in my opinion, is highly overrated. American airships have made a great majority of the moorings that have been made in the world, and these operations have generally been made on an even keel or even at a slightly stern-up inclination. Reference to the dynamic-lift curves for an airship will show that a high wind velocity is required before the dynamic effect obtainable from a few degrees of inclination amounts to any considerable quantity. Actually, some of the most trying conditions for mooring are found in winds with a velocity of only 20 to 25 miles, or even less, with gusts considerably higher, and the dynamic effect at such air speed is of course small. British mooring doctrine is based on the approach to the mast with the ship trimmed bow up, so that downward gusts would have to overcome this slight dynamic lift to cause downward motion of the bow.

It is not a simple practical matter to keep a ship trimmed stern down throughout a mooring; for one thing, passage of air strata of different temperatures may alter the buoyancy temporarily, but compensation for such changes must be made at once and not postponed. Next, while we do not know the exact structure of a downward gust, it does represent the resultant air flow. Coming from above, rather than its normal horizontal flow which produces the dynamic lift on the inclined ship, it would appear that the downward flow of air in the gust not only deprives the ship of the dynamic effect at the time it is needed, but actually tends to drive it downward in addition.

In the American stub-mast mooring it is of course not possible to maintain the stern-down attitude during the entire operation. As a result there is one school of thought that has advocated the construction of a telescopic mast so that the actual mooring operation might be conducted at a sufficient altitude to allow the stern-down inclination, and then by telescoping the mast obtain the advantages of riding to the stub mast. However, in view of the slight assistance to be expected from the inclination and the resultant dynamic lift, this proposition represents practically negligible advantage and has the considerable disadvantage of complicating for all uses an otherwise simple mooring mast. It overlooks entirely the fact that the high mast in itself lacks any vertical control that the low mast does not possess. It is therefore my opinion that the building of a telescopic mast to secure the stern-down inclination or for any other reason now apparent is not warranted.

The use of the ship's rudders and elevators to counteract the effect of lateral and vertical gusts while mooring is generally of little value. The use of these movable surfaces during the mooring operation in gusty conditions is generally sluggish and tardy, and on the whole unprofitable. Human instinct is not able to detect or interpret gusts sufficiently in advance to prevent their effects; it is difficult also to know when to shift the controls to counteract for the opposite gusts. Furthermore, it must be remembered that in trying to lift the bow by means of the horizontal rudders to counteract a downward movement of the bow, the whole ship moves bodily downward for a short period. One must draw the deduction therefore that the use of the elevators to counteract vertical gusts during the mooring operation is generally too limited to be of any practical value.

The *U.S.S. Akron* is equipped with propeller mountings which enable the ship to obtain vertical thrust both in the upward and downward directions. It would appear that, among other virtues, this vertical thrust might be of considerable value in overcoming vertical gusts; experience may substantiate this. However, from the suddenness with which gusts strike and from the relative sluggishness of human reactions, I believe the use of vertical thrust during the mooring operation in gusty weather will be a ticklish procedure.

One school of thought proposes to obtain the necessary upward vertical force from excess static buoyancy by the generous use of ballast. In my opinion this has definite limitations, although under ordinary conditions, such as those in which possibly 90 per cent of all airship moorings of the past have been made, the buoyancy of the ship itself suffices. It is probable that in 75 per cent of the moorings to be expected, even in wider operation, a reasonable amount of buoyancy and the normal method of mooring as we know it today would suffice. It is the other 25 per cent, however, that must be provided for, and in order to be able to use the corrective measures when they are required under these severe conditions, the system used daily should be a complete one. Nature seldom, if ever, guarantees that the good conditions at the beginning of a mooring will not rapidly degenerate into unsuitable ones before its completion. Although a ship equipped with water-recovery apparatus will, at the end of a long flight, have a great amount of water ballast available for dropping to obtain excess buoyancy, such buoyancy adds materially to the load on the mooring winches, which would have to be made considerably more powerful than would otherwise normally be required. Also it is not to be expected that all ships will be equipped with water recovery. I recall the arrival of the *Graf Zeppelin* at Lakehurst from her South American flight in 1930 when the sole useful ballast, either liquid fuel or water, was represented by one bag of emergency ballast dropped during her spectacular but "heavy" landing. The *Graf Zeppelin*, under the conditions, could not have made a safe mooring to a mast.

Should a very light ship be required to discontinue her efforts to moor due to failure of mooring wires or other equipment, as has been experienced, or for other reasons, the resultant loss of helium in order to land her later might be a very expensive item in ships of modern sizes. The place in the ship at which we desire the upward resistance to vertical gusts is at the bow. If ballast is dropped throughout the length of the ship, she will assume the normal stern-up attitude of a light ship; this, according to the doctrine of the high-mast proponents, is undesirable. It is ordinarily a difficult matter to determine, in variable conditions, just how light the ship should be during the mooring operation. It is perfectly apparent that the force required to overcome momentum is considerably greater than that required to prevent

the acquiring of momentum. It may therefore be seen that a ship could easily get into a state of excessive buoyancy.

In my opinion the foregoing features are not sufficient remedies to care for the matter under consideration. I regret that my deduction involves additional equipment in an operation which, within its limited usefulness, has heretofore been marked by great simplicity, but at this time I can see no other method for solving the problem than with additional equipment. To the anxious airship captain who has been helped out of a tight place by this added gear but once, the up-to-date combination required will forever after appear sweetly simple. It might be recalled that the mobile mast had its scoffers, but today no airship operator wishes to handle his ship without it. We must incorporate in the mooring system a mechanical provision which is effective at all times after connecting to it, for resisting undesirable downward movement and not one which we attempt to use intermittently after the accumulation of downward momentum. This provision we have already made to a considerable degree in the similar matter of lateral yawing, where we do not wait for the ship to acquire a yawing motion before attempting to apply a counteracting force. The yaw wires are put into complete operation at the earliest possible moment. Likewise, for vertical yawing we must make advance provision. The pyramid of wires that we establish in our mooring must have added to it an additional upward force. If it cannot be applied directly at the bow, then it should be as near as possible to that point.

There are two possibilities in this connection: one is the application of the force from below and the other from above. There have been numerous suggestions as to how to put these ideas into practical execution: glorified buffer bags approximating small balloons have been suggested; collapsible ladders and platforms providing an adjustable rigid connection with the earth have been proposed; one wag has even suggested the use of a stream of water against the bottom of the ship. Unfortunately, such proposals for utilizing the force from beneath the ship do not appear to be practicable.

On the other hand, it has been proposed that there be established above the top of the mast a point from which there can be led downward to the nose of the ship an additional wire kept under a reasonable degree of tension. This wire would thus supply continuous resistance to the ship's acquiring undesirable downward movement under the influence of gusts. In other words, I propose to increase the rigidity of the apex of the pyramid or the point in space formed by the wires at the ship's nose, by continuous mechanical assistance, and not by various means of fluctuating values. In my opinion this does present a practicable solution.

The ideal mooring system would be that in which the airship could be transferred immediately from her free-flight condition into her moored status without an intervening period. There appears to be no practicable way in sight at the present for accomplishing this ideal, and we must therefore reconcile ourselves to a transition period and make every effort to reduce this transition period to an absolute minimum. Certain practice abroad has been to let the ship rise on about a thousand feet of main cable before beginning to haul her down. It has been my experience that dangling on a single cable under gusty conditions is without a doubt one of the most annoying times in the life of an airship captain. American practice is based on high-speed winches and low approaches, so that the slack in the mooring wires may, immediately after the coupling to the ground wires, be whipped into the taut condition with the least possible delay. The ship is much more secure once she is on three taut mooring cables.

If we are to incorporate the vertical wire from a point above the ship, it would be desirable to make that point as high as possible. There seems, however, to be a feeling of apprehension over the possible danger to the ship from the presence of such a superstructure above the mast. This apprehension I do not share, as the proper method of supplying the force from above would not include the erection of the superstructure in its mooring position until after the ship had been secured to her yaw guys and was thereby restrained from overriding the mast structure. The superstructure or topmast for providing the point above must not project to leeward beyond the center of the mast, must be capable of quick erection and quick removal, and must be provided with adequate motion in the horizontal plane so that in shifting winds it may be kept opposite the ship at all times.

A stub mooring mast for a ship the size of the *Akron* will be about 80 ft high. It has been calculated to be perfectly feasible to build a structure capable of furnishing a point 200 ft above the ground for the desired vertical control, of sufficient strength for the loads expected, and still easily and readily erected and removed. Its motion in azimuth also is an entirely feasible matter. At the present time it appears to me that this is the best solution yet offered, and in the absence of something better we shall no doubt try it practically.

There is an additional phase of the mooring operation that requires correction. It has already been pointed out that the anchorages for the ground yaw wires must be mobile and that this can be easily accomplished. The remaining feature is the introduction of a method for equalization of the tensions of the yaw wires. It can readily be seen that a gust striking on the starboard side of an airship whose starboard yaw guy is of fixed length and of an unyielding tension would tend to drive the ship downward on the yaw guy as a radius. In so doing, the leeward yaw wire would become slack. Upon the dying out of the gust and the attempt of the ship to return to its previous position or, as normally happens, upon the impact of a gust from the other side, the impact load on the unyielding yaw wire might assume great proportions. As a matter of fact, yaw guys have been parted in such a manner.

It is therefore apparent that we must provide a means of maintaining the two yaw wires automatically at even tensions. This may be done in effect by veering slightly on one and reeling in on the other. Actually it is possible to devise a means for adjusting these lengths without using the winches for that purpose. The main point to be gained is never to allow slack in either yaw wire or, as a matter of fact, in any of the mooring wires.

To sum up the improved mooring operation, we may expect to find it follow these lines:

- The mooring will be made to a mobile stub mast accurately centered and firmly anchored within the circular railroad track for the stern carriage
- The mooring winches will be located at the mooring site in such a way as not to impede movement of the mobile mast
- At least four wires will be used—that is, a main cable, two yaw guys, and the fourth wire or its equivalent for vertical control
- Yaw-guy anchorages will be made mobile so as to maintain continuously the proper angle relative to the ship's structure in shifting wind conditions
- There will be incorporated a practical means for automatically maintaining yaw-guy equalization
- The approach of the ship will be made on approximately an even keel until the latter part of the operation, when a stern-up attitude will be assumed
- After the bow has been secured to the mast, the stern will be hauled down into its stern carriage by means of a cable and a winch.

I fully expect that the mooring operation to the stub mast will eventually be accomplished more easily, more economically, and more safely than can be done at the most elaborate high mast in use today.

The next important phase of our problem of adequate terminal facilities for a rigid airship involves the servicing of the ship and its mooring out under the wide range of weather conditions to be expected. Experience in the United States has pointed out that there are definite advantages in riding to the short mast with the stern restricted from vertical motion. As a matter of fact, I do not know any one in the American airship operating personnel who would shed tears at the abolition of the high mooring mast, and I confidently believe the high mast will pass out of style quickly upon successful attainment at the stub mast of the items pointed out.

Before going into the relative merits of the two masts, I wish to comment on one of the commonly alleged disadvantages of the stub mast. Proponents of the high mast point out that in case of a breakaway or a forced departure from the mast, the airship might be expected to have a better chance from the taller structure. With the importance of this feature or this idea I do not agree. In the first place, it is difficult to conceive of a modern airship breaking away from a modern mast. The British *R-101* rode out a gale in which the wind reached gusts of over 60 miles an hour over an extended period, and in which the maximum gust attained a velocity of 83 miles per hour. I believe any airship captain would hesitate to make a deliberate departure from a mooring mast of either type under average gale conditions. In any event, if he did make such a decision, he would have to depend on a very considerable static buoyancy to get his ship safely into the air. I do not believe that the hundred-foot difference or so in the mast height under those conditions would count very much. The mast structure itself is the main obstacle in the path of the ship and is at the same relative height whether it is a high or a stub mast. I cannot help wondering how an airship employing several tail drags simultaneously, as has been done with a recent ship at a high mast, would free itself of them in a breakaway or a storm take-off. It is the storm condition in which the tail drags are generally the most urgently needed. Furthermore, they would be needed up to the very instant of departure. It would appear to be a difficult task indeed to rid the ship of several tail drags simultaneously and instantly, and also to have any but the roughest idea of the buoyant condition of the ship under storm conditions with several tail drags attached. Depending on generous use of ballast and upon vertical thrust from propellers, and upon the freedom from tail drags that are not automatically and instantly detachable, I believe that in a forced departure the stub mast is equally as safe as the high type. It is a real emergency in any case, and must be met by gaining altitude with reference primarily to the mooring mast, whether high or low.

Comment on other relative features of the two types of masts would seem in order. Following are my comments:

(1) *Cost.* A modern high-mast installation costs several times as much as the most elaborate or complete stub-mast installation now under consideration. Even without added equipment for what I consider satisfactory vertical control, a recent high mast on this continent is said to have cost about $375,000. With necessary vertical control incorporated, the cost of the high mast should be materially greater than this figure. The cost of including vertical control of whatever type at the stub mast is proportionately much less than at the high mast.

(2) *Complication.* The top hamper of the most recent high masts is decidedly more complicated than our stub masts, even of the mobile type. Vertical control added to a high mast is decidedly more of a complication there than to the low mast.

(3) *Other Utility.* The high mast cannot be used for docking and handling purposes and requires an added "transporter" mast, if a mobile mast is to be used as the nucleus of a mechanical docking system. The stub mast can easily be made a mobile unit at a reasonable cost, thereby performing an additional very important function. Transition of an airship from a high mast to a "transporter" mast for docking is an added operation and may be a hazardous one. No such added features are involved in the use of the mobile stub mast.

(4) *Care of Ship While Moored Out.* While moored out at a high mast an airship is continuously being "flown," and consequently the strain on the crew is greater than at the low mast. Absence of vertical motion of the stern at the stub mast is a decided advantage as well as a safety factor in ordinary conditions, and particularly in wind shifts, rapid temperature drops, and gusty conditions. Large quantities of water are required at a high mast for continuous ballasting. At some locations this necessary water is difficult to obtain in the quantities required. With the stern supported at the stub mast, far less care is required in the case of rain or snow loads that might be great enough or sudden enough to force the stern to the ground at the high mast. Absence of tail drags at the low mast removes a great liability and a source of trouble.

(5) *Servicing and Loading.* Much more easily accomplished at stub mast, where loads can come aboard from the ground at logical points throughout the ship. At the high mast all loads and personnel come to the top of the mast via elevator and must be brought on board through the bow hatch.

(6) *Repairs.* More extensive repairs are possible at the low mast; ordinary repairs and replacements are more readily accomplished there than at the high mast.

(7) *Adaptability to Larger Ships.* Cost and size of stub masts for larger ships do not increase as rapidly as would be the case with high masts for the same ships. High mast is more easily standardized for various sizes of ships. A set of stern carriage tracks for each length of ship would probably be required at a stub mast. None is required at the high mast. Tail drag paths are required at high masts, but not at the low. More level terrain is required at the low mast.

(8) *Safety of Ship on Emergency Take-off.* Previously discussed.

(9) *Wind Velocities.* Generally markedly less at the low mast. Therefore the ship moored there should be safer.

(10) *Stresses.* Moored to the low mast, the stresses during riding are probably greater than at the high mast, but it is believed that they are entirely within safe limits.

(11) *Stern Weighing Device.* At the stub mast it is possible to incorporate a stern weighing device that, with the similar bow device, allows ready and continuous knowledge of the static condition of the ship. At the high mast it is difficult, if not impossible, to know the static condition under other than the best conditions.

(12) *Mooring Crew.* At the high mast 10 to 12 men have sufficed thus far. With vertical control added, this number may have to be increased to 16 or 18 men. No stub mast has yet been equipped for a full mechanical mooring, and therefore the number of men required by the present temporary combined manual and mechanical system cannot be fairly compared with the 10 to 12 men in the high-mast crew. With proper mechanical aids at the stub mast comparable to those at the high, it is probable that only 4 to 6 more men will be required at the low mast. This will include any extras that may be found necessary in the provision of mobile yaw-guy anchorages at the low mast. When vertical control is included in each type, the same difference of 4 to 6 men will probably obtain, but actually no more highly trained specialists are needed at the low mast than at the high.

(13) *Mobile Yaw-Guy Anchorages.* They are necessary at either type, but cannot be incorporated in the present high-mast

scheme without considerable added equipment. They are readily applicable on the stern carriage path already provided at the low mast.

(14) *Ease and Safety of Mooring and Unmooring.* This is discussed at several places in this paper. In my opinion there will be greater ease of mooring to the low mast, when corrected as indicated to be necessary, than to the present high mast. The safety of mooring will then be at least as great as to the high mast, even if the latter is provided with ertical control. In unmooring from either type, the mast structure itself is at the same height relative to the ship, and in either case is an obstacle to be hurdled. In an emergency the moored airship can be released from the low mast with more safety than is possible at the high mast due to the tail drags required.

(15) *Menace to Flying Activities.* The high mast is much more of a danger to airplane operations than is a low mast, particularly of the mobile type, which may be moved out of the way, or the expeditionary type, which may be lowered by its hinged heel.

The next phase of this paper involves that of housing and unhousing the airship mechanically. It is now realized that hangar design can be an important factor in this consideration. The shape of the building itself and of its doors, if not carefully determined, can give rise to very undesirable and even dangerous air-flow conditions at and in the vicinity of the shed and the handling point. Much more attention is being paid to this point today, and we may expect improvement in the housing operation because of the attention to aerodynamic design details of hangars.

Another interesting need in this connection is for a thorough analysis of gusts and gust structure. No one today has a satisfactory working knowledge of this feature, but undoubtedly more complete knowledge of this point could assist materially in our handling problem. We must know what nature is trying to do to us. There ought to be cheaper ways than learning from full-scale tests.

Ideas vary as to how far we should go in strengthening the airships themselves for mechanical handling and housing and unhousing. With the wind parallel to the hangar axis, it is a comparatively simple matter, dependent to some extent on the vertical loads that the ship's structure can safely withstand. In a beam or cross wind, the enormous loads on the full area of the airship hull make it a more difficult problem. In such a case there is involved not only the vertical strength but the beam strength of the ship as well. The principle of the revolving shed is still correct, but we hope it will prove much cheaper and sufficiently satisfactory to utilize a fixed shed and to revolve the ship itself into the necessary direction. Judicious selection of sites with a prevailing wind of marked directional constancy is a great natural factor of assistance.

It may be interesting to know that it was the vertical strength of the *Los Angeles'* structure aft that determined the maximum wind velocity in which that ship could be handled in and out of the shed. As a matter of fact, it was necessary to supply bracing equipment at the stern of the ship for use on handling occasions. With the mobile mooring mast and certain equipment for resisting the side-wind load, applied only around the stern, it is possible to handle the *Los Angeles* in a cross-hangar wind of 17 miles per hour. Meteorological records show that even at Lakehurst, where there is a high frequency of severe conditions, on 93 per cent of the days of the year there are periods of two or more hours a day during which the wind velocity drops to less than 20 miles per hour. Therefore any method that will enable us to handle the ship in a 20-mile cross-hangar wind would be highly acceptable, for with a safe mooring-out system it would not be necessary to wait very long for the opportunity of docking if required. However, as a result of extensive experience we have provided sufficient strength in the *Akron* and in her handling equipment to permit housing and unhousing that ship in a cross-hangar wind velocity as high as 26 miles per hour. While it is still felt necessary to have docks for airships, there is no doubt that the day is coming when airships will be capable of staying out over extended periods and will not have to resort to "dry docking" in a shed except infrequently.

By the simple expedient of making the stub mooring mast mobile, we have obtained the fundamental unit for a satisfactory mechanical-handling system. It seems firmly established that future handling systems will continue to be premised on the use of a mobile mast. At Lakehurst there has been in operation for a considerable time a triangular mobile mast 60 ft to a side, weighing about a hundred tons and mounted on crawlers. It has proved highly satisfactory. There are of course refinements yet to be made in the mast, such as the incorporation of its own propulsion plant. The crawler mast has proved entirely satisfactory and, in my opinion, has some very forceful advantages over a similar mast mounted on trucks to run on railroad tracks. However, in order to try several promising ideas, there has been constructed a second mobile mast that will operate on rails. Competitive use of the two types should develop any advantages of either, but there is no reason to believe today that the crawler type will be surpassed in its attainment of the necessary degree of practicability.

In the movement of the ship from its mooring point to the vicinity of the dock for housing, or vice versa, it is necessary to have at the after part of the ship a point of contact with the ground. Formerly men on the hand rails of the after car kept this part of the ship under control during its movement over the ground. This same function is now accomplished much more satisfactorily by the use of a large pneumatic wheel secured to the stern and capable of easy castering. This wheel is capable of high vertical loads as well as ready change of direction and affords smooth rolling contact since the ship is kept "heavy" so that the wheel ordinarily cannot lift off the ground. As an additional precaution, a tail drag is also employed at the stern. Whether this castering device continues to be a single wheel or blossoms out into a multi-wheeled unit is unimportant. The necessity and correctness of such a device have been definitely established.

Under cross-hangar wind conditions there is required apparatus for hauling the ship up against the wind into the direction parallel to the hangar for entering. Conversely, in undocking under such conditions, the ship must be kept under control while the stern is being allowed to swing into the wind direction after clearing the hangar. There are several ways of providing mechanical power for pulling the ship up against the wind or for restraining it. The idea that immediately suggests itself is that of having tractors attached by lines to the ship pull it into position. There are several reasons why this tractor scheme is not a practicable idea. One is that the pull from the tractors of today, due to their means of traction and their methods of changing direction over the ground, is extremely irregular and even dangerous to the hull of a ship. Another reason is that tractors are not built in commercial sizes that provide sufficient drawbar pull for our purpose. For example, a drawbar pull of 63,000 lb will be required in the case of the *Akron* for this purpose. There is a method of hauling up and controlling the stern based on the use of straight lengths of track at right angles to the hangar axis. Another method employs a circular track in the immediate vicinity of the hangar. While both straight- and circular-track ideas have advantages and disadvantages, it has been decided to experiment first with the circular-track idea.

There will be required a hauling-up circle at each end of the hangar. From the mooring site at which the ship was landed from flight, the mobile mooring mast will cast off its anchorage and proceed with the ship to the center of the hauling-up circle.

While the ship is still headed into the wind, a huge steel beam specifically designed for this purpose will be moved by a locomotive along the circular track to a point under the lower fin of the *Akron*, and there the ship will be attached to the beam. To the extremities of the beam, which is about 164 ft long, there will be run lines from both sides of the ship. The locomotive will then tow this stern-handling beam into the direction for entering the shed, pivoting the ship at her nose about the mobile mast, then stationary at the center of the circle. When the stern-handling beam has reached the rails parallel to the hangar, its second set of trucks will be dropped to the handling rails. The locomotive will be detached from the beam and moved off clear on the hauling-up circle. The mobile mast will then get under way and move the ship into the shed. No attempt is made to provide the stern-handling beam with its own motive power, as it may not be necessary, and it would probably be a difficult matter to synchronize the power of the mobile mast with that of the motorized beam to prevent any possible "accordion" effect on the ship. For the *Los Angeles* the mechanical-handling gear at the stern of the ship is towed by cables to the ship itself, and it is believed that a similar arrangement of equivalent cables from the stern-handling beam to the structure of the *Akron* will prove satisfactory for the latter ship.

Differences in airship construction make themselves felt in this handling problem at this point. In a ship not designed for mechanical handling, it was necessary to apply the loads at carefully selected points and always in proper directions relative to the structure. Although satisfactory means were found, nevertheless this feature complicated the design of the mechanical-handling equipment for the *Los Angeles*. For the *Akron*, with its differences in structural design and construction, the matter is not such a difficult one. Instead of having to maintain the pull always normal to the ship's length, as is the case with the *Los Angeles*, it will be safe to pull on the main frames of the *Akron* at a considerable angle. This allows on each side the leading of a wire from each of several frames near the stern of the ship to one point at a central location abreast the after part of the structure. This results in the necessity for providing but one movable docking point on each side of the stern of the ship instead of the four points now required on the *Los Angeles*, for resisting beam-wind loads. The net result is beneficial, for the length of the handling gear aft runs athwartship rather than fore and aft.

There have been various suggestions as to other ways of handling large airships mechanically, and a number of them possess merit. Among these is the suggestion to employ a mobile mast at the stern as well as at the bow. Another proposes, in addition to the bow mast, one or more mobile masts or their equivalent along each side of the after portion of the ship. By the time these individual units have been connected up and synchronized, the project almost invariably reaches the proportions and characteristics of a huge cradle or raft similar to that proposed several years ago in this country and more recently in another adaptation abroad. In our efforts we have endeavored to avoid the necessity for equipment of such magnitude. We confidently expect that the use of the mobile mast at the bow, together with the stern-handling beam at the after part of the ship, will prove completely satisfactory for handling such large and modern ships as the *Akron*.[3]

In conclusion, it is my opinion that the problems of supplying the remaining essentials in the mooring and handling project for rigid airships will be met successfully in due course of time. In the absence of more widespread airship operating experience there is naturally room for some differences of opinion in practical solutions to problems. So no doubt the high-mast school of thought will continue to have patronage. However, as may have been surmised from this paper, my own experience so far tells me that the successful mooring and handling of rigid airships will center about the low mast.

The aeronautical world moves rapidly. New developments will continue if we can arouse the proper interest in the project. Although the fundamentals remain the same, no man is safe in making his suggestions as to their applications to aeronautical matters too final and fixed in the present pioneering state of the art. However, the opinions and ideas I have expressed herein are those I hold today in the light of all the information and experience that I have been able to gather.

When the problems of airship terminals reach a satisfactory status, as they soon will, I am sure it will be found that extremely important contributions will have been made by the airship personnel and engineering talent of the United States.

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[3] For a description of recent equipment devised at the Naval Air Station at Lakehurst, N. J., for handling the *Akron*, see paper on "Mechanical Equipment for Handling Large Rigid Airships," by Lieut. C. M. Bolster (CC), U. S. N., at Sixth National A.S.M.E. Technical Aeronautic Meeting, Buffalo, N. Y., June 6 to 8, 1932.

AER-55-7

Heating and Ventilating of Cabin Airplanes

By M. GOULD BEARD,[1] FARMINGDALE, N. Y.

Comfort for airplane passengers is being seriously considered, and aeronautical engineers are designing features for the airplane cabin that follow closely the comforts of railroad drawing-room cars and the facilities of the larger motor coaches. The adaptation is easily made, with alterations in weight and size. Heating and ventilating systems are replacing the rudimentary provisions of the past. The paper states the problems of heating and ventilating of cabin airplanes, gives available data, describes methods in use or attempted, and urges the mechanical and aeronautical branches of the profession to join in solving the problems.

THE airplane is rapidly establishing itself as the vehicle for fast transportation. Its mechanical reliability is beginning to compare favorably with any of the longer-established means of transportation. Aeronautical engineers all over the world are concentrating on obtaining the maximum efficiency in performance for this carrier of commerce. Intense effort is being applied to make the airplane reliable in all kinds of weather. It is only recently, however, that the comfort of passengers has been seriously considered, as is evident by looking through the cabins of transport planes built as recently as 1930.

Since the air lines have taken up passenger transportation in earnest, far-sighted aeronautical engineers have concentrated on building models suited especially for this work and have designed into the plane as many of the comforts of travel already existing in railroad drawing-room cars and in large motor coaches as could be readily adapted to the airplane. Modification of these appliances is always necessary, particularly as to weight and often as to size, to make them usable in airplanes, but in general the adaptation is easily made. Such phases as heating and ventilating have been left with only rudimentary provisions until recently. The expansion of the air lines, calling for planes suitable for passenger carrying, put into the minds of some engineers the necessity of designing really efficient heating and ventilating systems. Thus far these problems have been tackled by aeronautical engineers within the industry. To get the best results the industry will have to utilize the training and experience in the mechanical branch of the engineering profession. The main object of this paper is to state the problems of heating and ventilating of cabin airplanes, to give whatever data are available, and to describe methods in use or being attempted, with a view to bringing together the mechanical and aeronautical branches of the profession on these problems.

The basic principles of heating and ventilating are well known, and courses in these subjects are in the catalogs of all colleges of engineering. The application of these principles to the airplane require a knowledge of:

1 The physical conditions that are encountered in flight, such as temperature limits of the upper air at all times of the year and the probability of rain, snow, and dust in the air.

2 The physiological and psychological reaction of passengers riding in airplanes.

3 The structural limitations of the airplane as to weight, space, and location.

4 The safety requirements that must be met in aeronautical design.

5 The sources of heat and air conditions available which can be put to use in heating and ventilating of airplane cabins.

6 The limitations and advantages of the various systems for heat transfer or generation and the applicability of each to any particular airplane layout.

It becomes necessary to make a clear statement of the conditions to be met; to discuss the mental attitude and physical requirement of the airplane passenger; to specify weights and establish locations for heater and ventilator systems; to state safety requirements; to point out available heat sources and locations from which fresh air may be obtained; and to compare the various systems which can be used in heating and ventilating an airplane.

Physical Conditions

In the United States, summer-air temperatures of over 100 F are seldom met; 90 F can be considered as an average high temperature for the West and Southwest during the hot seasons. Airplane cabins subjected to direct sunlight in flight will get several degrees hotter than this, unless very well ventilated. However, it is almost always possible to find lower temperatures at some high altitude, so that temperatures above 85 F are seldom encountered for more than a few minutes. The passengers' cabin can be maintained at atmospheric temperatures with suitable ventilation. When the pilot's cab or cockpit is located ahead of the passenger cabin and just behind the fire wall of a single-engine plane or a trimotored plane, it is more difficult to keep it cool. With the present use of N.A.C.A. cowls and Townend rings, with high-speed air baffles between air-cooled engine cylinders, the pilot's cab is enclosed on all sides and in front with a close-lying layer of hot air from the engine which has not been diluted by the outer air stream. The author has flown ships with insufficient ventilation in the pilot's cab when the thermometer in the cab read 112 F, with outside air at 85 F, which is an unsafe condition for any human being to endure while performing the duties of a pilot.

At the other extreme, the heating system must be designed to keep the entire passenger cabin and pilot's cockpit at 70 to 75 F with outside air temperature at —20 F. Heating systems designed for operation in Canada should be calculated for entering air at —40 F. The heater must supply heat to the cabin within five minutes after the engine is started for warming up and must be readily controllable to take care of sudden changes

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[1] American Airplane and Engine Corporation. M. Gould Beard was born on November 13, 1896, at Foochow, China. He entered Oberlin College in 1916; enlisted in Aviation Corps, December 21, 1917; flight training received at Kelly Field, and was commissioned second lieutenant, November 15, 1918; entered the marine Diesel-engine field after the war and worked up to chief engineer's position; attended University of Michigan, 1921–1925, specializing in internal-combustion engines; employed by Ingersoll-Rand Company as experimental and test engineer, 1925–1927; entered the aviation field in 1927 as installation engineer and test pilot for the Fairchild-Caminez Engine Corporation; transferred to the airplane division as chief test pilot at the Farmingdale plant until October, 1932. Mr. Beard did all the test work on the Caminez engine; all development and experimental testing on the Pilgrim 100X, 100A, and 100B; all initial test work on the fast all-metal low-wing 10-place transport now known as the Clarke GA-43.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

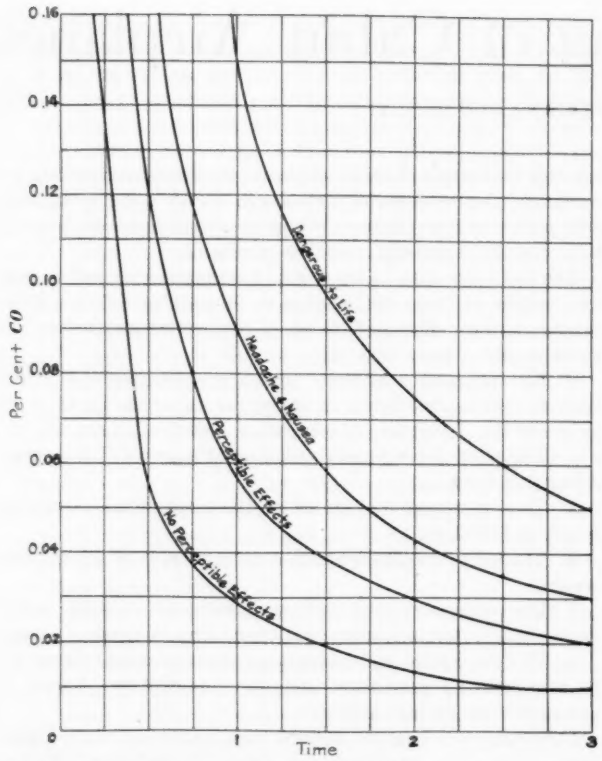


FIG. 1 EFFECTS OF CARBON MONOXIDE FOR GIVEN TIME ON HUMAN BEINGS

(Taken from Bureau of Standards Technical Paper No. 212.)

in air temperature. A modern transport plane will climb from sea level to 10,000 ft in 10 to 15 min, depending on the load. During this climb, air temperatures may change from 60 to 70 F to below 0 F. Maintaining the desired cabin temperatures with this rapid change in the air temperatures will require thermostatic control to relieve the pilot of one of the many duties that have been piled upon him in scheduled transport operation.

Rain offers the problem of water separation in some point in the ventilator ducts. Snow offers no problem, since very little flying is done in heavy snowstorms and light snow is readily melted and vaporized in the heater before reaching the heater ducts to the cabin. Dust is a problem in the Southwest part of the United States, where sandstorms are encountered in flight and where sandy fields permit the propellers to pick up dust and blow it into the ventilators. Dust separators are desirable in the ventilator ducts of planes operating through this region.

CABIN INLET AND OUTLET LOCATIONS

Effective heating and ventilating systems must admit hot and cold air from points in the cabin walls or ceiling which will give the passengers the greatest degree of comfort. Since the volume of air to be admitted is large for the volume of the enclosure, it is almost impossible to prevent drafts. Single air inlets always subject a few of the passengers to violent drafts of hot or cold air and often prevent the other passengers from receiving the benefits of heat and ventilation.

Individual inlets for hot air and cold air, correctly designed, allow each passenger to obtain temperature conditions to his own liking and eliminate strong drafts in any one part of the cabin. It has been found that cool air is most effective in keeping a passenger comfortable in hot weather if the individual ventilator is located above and in front of the seat so that it will blow into the passenger's face. The direction and quantity of the air stream being controllable, the passenger can adjust conditions to suit his needs without discomfort to the other passengers. Hot air should be admitted at a low point in the cabin through individual hot-air grids or inlets which are also controllable by the passengers.

Heater inlets or grids in the floor are undesirable because they collect dust and dirt and blow it out whenever the grid is opened. The grid in this location is subjected to continued tramping and soon gets bent and stuck in one position. Hot-air ducts are often laid on top of the floor from front to rear of the cabin between the legs of the chair. Individual shutters in the sides of these ducts are subjected to enough abuse from the feet to get jammed in service and are undesirable. A more desirable location for the heater inlet is on the side wall of the cabin close to the seat and 5 to 8 in. off the floor and designed to blow the air out and slightly down.

The location of the air outlets from the cabin requires some experimental work to determine the direction of natural air flow in the cabin and to determine the region of lowest pressure or best air-stream velocity on the outside of the fuselage. On one experimental plane several locations for outlet ducts, which presumably should have produced suction, blew air into the cabin instead. Every airplane cabin will have a definite motion of air in it. In high-wing monoplanes this is usually from the rear forward because of the openings about the wing spars and fittings which connect the cabin with the suction at the top of the wing and because the pressure on the sides of the fuselage forces air in through door and window casings, which are usually at the rear of the cabin.

If the direction of air flow is well defined, the suction vents should be located at the opposite end of the cabin from the neutral air exit. This will even up the air flow and distribution of heat in the cabin. It will be difficult to keep the other end of the cabin warm in very cold weather, if the vents are placed at the end of the natural air exit.

Recirculation of cabin air has been often proposed and practiced in a few ventilator systems. Recirculation has two advantages—namely, that the weight of the heating unit can be reduced and that rapid changes in outside air temperatures do not affect cabin temperatures as quickly. It has the very great disadvantage that odors from airsickness are recirculated through the cabin and remain in the air longer. This one disagreeable feature makes recirculating of cabin air undesirable.

LOCATION OF AIR INTAKE

The success of a good ventilating and heating system depends largely upon the location of the air-intake openings on the outside of the plane. Because of the many types of airplanes, with widely differing conditions of structure and air flow, no specific rules can be laid down for the locations of these intake openings. The following general rules may be applied to any plane:

1 Locate the intake at a point entirely free from possible contamination with exhaust gases. This rule should be self-evident; yet an inspection of heaters and ventilators now in service on many planes will prove that its importance is not appreciated.

2 Locate the intake at a point where no oil or gas fumes can enter it.

3 Locate the intake high enough to be free from dust picked up by the propeller while running on the ground for warm-up or taxying.

4 Locate the intake in the propeller slipstream if possible to get the advantage of increased air blast.

SAFETY REQUIREMENTS

Freedom from carbon monoxide and fire hazards weigh about

equally in importance. Fig. 1 gives a series of curves showing the toxic effect of carbon monoxide as determined by the Bureau of Standards. A concentration corresponding to the line of "Perceptible effects" is a danger line for the pilot. These percentages of CO are so small for the longer periods of time that every precaution should be taken to locate heater and ventilator inlets where there is no possibility of taking in exhaust gases.

The air close to the fuselage of a plane having an engine mounted in the nose of the fuselage is contaminated with fumes from the engine compartment. This is especially true where N.A.C.A. cowls and Townend rings are used, as the air from such installations flows in a stratum close to the fuselage. Because of this, air inlets should not be located close to the skin of the fuselage. One type of ventilator often used consists of a scoop on the outside of the fuselage attached to a tube passing through the fuselage wall to the inside of the cabin. The cabin end of the tube is fitted with a nozzle at an angle relative to the outside scoop, so that when the scoop is facing the airstream the nozzle blows air on the passengers. The assembly is rotatable, so that the scoop can be turned at any angle to the air stream, and when turned to the rear it produces a suction. One of these ventilators is installed for each passenger.

This type of ventilator is inferior in every respect, except weight, when compared to a ventilator fed by a duct. It is very noisy, admitting engine and propeller noises and air-rush noises, and even producing a sharp whistle if turned at an angle to the airstream. If this type of ventilator is used, it is futile to attempt to soundproof the cabin walls, as any benefit from soundproofing is canceled by the noise admitted by this type of ventilator. The cabin of a modern nine-place single-engined transport was well soundproofed during the development of the model. It was considered one of the quietest cabin planes ever developed. A customer's specifications called for the scoop-type ventilators under consideration. After these had been installed, the cabin was as noisy as any in which no soundproofing has been installed. The ventilation system was changed to a type having individual ventilators opening from ducts bringing the air in from the wing butts. With this system of ventilation the cabin was as quiet as at first, and it was found that the incoming air was 3 to 5 F cooler than that admitted by the rotating-scoop type which took air from a layer 1 in. from the surface of the fuselage.

Two very good locations are available for ventilator inlets of the duct system. High-wing monoplanes offer a very good location in the leading edge of the wing butt sufficiently far out from the fuselage to be free of contaminated air, but in the area of the propeller blast. This location has the advantage of receiving a strong blast of air at all times, even when the ship is on the ground with the engine idling. The top of the nose of the plane also offers a good location, especially if there is no engine in the nose of the fuselage. The inlet scoops may be located directly inside the ring cowl of a radial engine and the ducts carried back through the fire wall. That portion of the duct passing through the engine compartment must be insulated by asbestos, as the heat of the compartment will raise the temperature of the air several degrees if the aluminum duct is left bare.

Weight and Volume Requirements

A duct-ventilator system taking fresh air in from the leading edge of the wing at a point 7 in. out from the fuselage of a 10-place plane has been constructed for a total weight of 13.3 lb. (See Fig. 2.) This includes all parts such as the wire screen at the intake, brackets and clamps to hold the ducts, ventilator nozzles which may be turned in any direction and which are equipped with valves to regulate the amount of air blast, marker plates for each ventilator to instruct the passenger, and the ducts completely concealed between the outer and inner skin of the cabin. The weight of the ducts represents 0.29 per cent of the weight empty of this particular plane with a structure of welded-steel-tube fuselage and wooden fairing covered by fabric. A metal monocoque fuselage could incorporate these ducts as part of the structure, with the skin of the fuselage as the outer wall, thereby saving about 20 per cent of the weight of the ventilator system.

Airplane cabin volumes vary between 25 to above 60 cu ft per person for planes between 4 and 12 passengers. From tables on heating and ventilating it appears that adult persons at rest should be supplied with about 1400 cu ft of air per person per hour if the CO_2 content is to be kept below 0.08 per cent, or a total of 14,000 cu ft of air per hour for a 10-place plane.

Such a plane with a cabin volume of 440 cu ft would have about 32 complete changes of air an hour, or a complete change in 1.87 min. Two 4-in. ducts, one from the leading edge of each wing, would deliver this volume with an air velocity of only 1340 ft per min in the ducts, and there would be sufficient pressure at the end of a 20-ft duct to deliver a good airstream from the last ventilator. Changing the air in the cabin 32 times an hour without producing violent drafts requires inlets and outlets well placed along the cabin.

Hot-Air and Cold-Air Ducts

Heater and ventilator ducts properly designed can be entirely hidden between the cabin interior and outer skin of the ship. Fig. 2 shows the combination heating and ventilating system developed for the Pilgrim 100-A. The ventilator inlet openings are located on the lower side of the leading edge of each wing stub just outside the hot-air film from the engine. The ducts are $3^1/_4$ in. in diameter and are made of 0.025 in. thick aluminum. The opening is covered with No. 8 mesh wire screen. Letters A indicate the individual ventilator outlets. A_1 is a duct to a central ventilator blowing down on to the front bench and intended

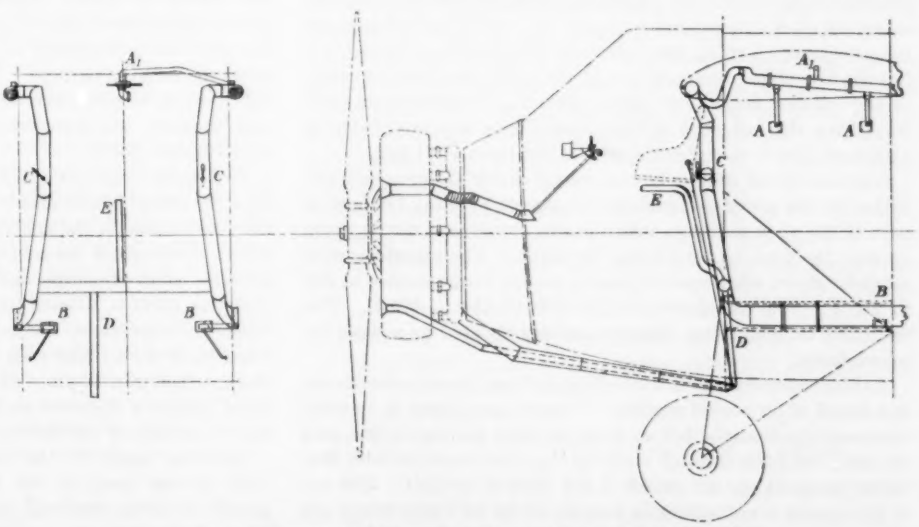


Fig. 2 Combined Heater and Ventilator Duct System

for the middle passenger. Letters *B* indicate the individual heater outlets. The last outlet blows to the rear to heat the person in the last seat.

The reverse curve in the upper ventilator ducts, as they leave the vertical ducts, is necessitated by control cables and structural members. At *C* in each vertical duct is a butterfly valve with a lever marked "Off" and "On." In cold weather these valves are shut off, cutting the cold air off from the lower ducts and allowing only hot air from the heater to enter the lower shutters. In very hot weather, the heater is shut and these valves are opened, which permits cold air to enter at top and bottom.

The duct *D* comes from the hot-air exhaust heater and enters a horizontal tube connecting the two vertical ducts. *E* is a duct leading to the pilot's cab and blows hot or cold air directly on to the pilot's feet. The location of the individual outlets, and especially that for the pilot's cab, requires a thorough knowledge through actual observation and study of the particular cabin and cockpit under consideration. The pilot's cab is subjected to very strong air currents. Any holes in the fire wall or seams and openings about the walls of the cab are undesirable because the temperature of the air leaking through cannot be controlled. Correctly located heater and ventilator outlets can make the pilot's cab very comfortable in any kind of weather.

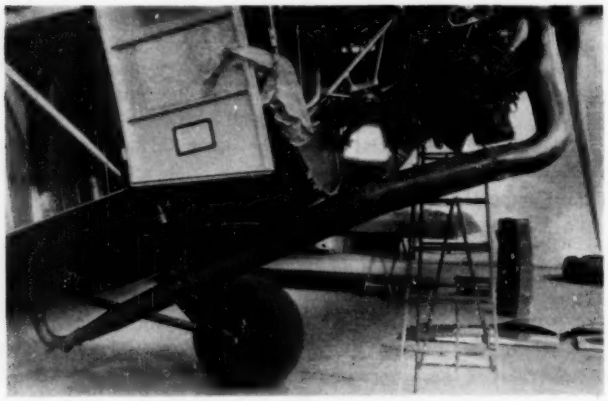


FIG. 3 HOT-AIR-EXHAUST HEATER ON PILGRIM 100-B

Heat Sources. The high temperatures of the engine exhaust permit heater units to be cut to a minimum size and weight when utilized for cabin heating. Since operating conditions vary considerably, it is impractical to figure too closely on exhaust-gas temperatures. During one test on a 575-hp radial engine in a full-throttle climb, temperatures at the junction of the two arms of the exhaust stack were above 1600 F. Temperatures halfway down the tail stack of this same engine averaged between 1100 and 1300 F at cruising speeds of 1650 and 1750 rpm.

Solution-cooled engines have a second source of heat available. Either by the use of an auxiliary radiator or by using the hottest part of the main radiators with suitable scoops and ducts, warm air may be obtained for heating the cabin. The temperature of ethylene-glycol when used for engine cooling is maintained at 240 to 250 F. The commercial liquid boils at 310 to 330 F. The operating temperatures, therefore, would be used as a basis for calculations.

A third source of heat is the cooling air from the cylinder barrels and heads of air-cooled engines. Funnels are placed at suitable locations to collect the hot air from the head and barrel fins, and the air is led back through ducts to the cabin, with suitable flap valves to direct the air outside if not wanted for heat. The use of this source is not advisable because of the oil fumes which are always collected from the engine.

HEATING SYSTEMS

It is proposed now to outline the various types of heating systems and discuss their relative merits as applied to aircraft use and to discuss a few of the important details of their design.

1 Hot-Air-Exhaust Heater. This type of heater is the oldest and most often used in the aircraft industry. Its improper construction has caused several deaths from monoxide poisoning, and many flights terminated short of their destination for the same reason. The main causes of danger from carbon monoxide poisoning are:

(*a*) Openings such as joints and splices in the exhaust pipe ahead of the air intake which permit a leakage of exhaust gases to get into the hot air going to the cabin.

(*b*) Welded seams, either annular or along the exhaust pipe, which crack open from repeated stresses set up by expansion and contraction in service, allowing exhaust gases to leak into the hot air going to the cabin.

(*c*) Poor design, which throws expansion stresses on to some part of the sheet metal, which eventually fails, allowing gases to leak into the heater.

All three of these faults may usually be found in every incorrectly designed exhaust-type heater.

The hot pipe is entirely enclosed and no inspection is possible, so that cracks developing are not caught by mechanics or by inspectors before they get big enough to become dangerous. Partial carbon-monoxide poisoning has occurred so often from hot-air-exhaust heaters that the Army will not accept this type of heater.

The hot-air type of heater, if correctly designed, can be made safe and reliable when the plane is given regular periodic inspection. At the present stage of heater development it is the cheapest to build, the lightest in weight, the simplest to operate, and the easiest to maintain. Fig. 3 shows a correctly built exhaust heater. It consists of a seamless stainless-steel tube 0.049 in. thick passing through the exhaust tail stack. There are absolutely no seams or welds in the tube inside the exhaust pipe. The tube is clamped at the entrance and exit of the exhaust pipe by bolted split collars welded to the exhaust pipe or by simple asbestos-packed expansion joints so that it is quickly removable for inspection. The air intake is located under the anti-drag ring just ahead of the engine, and flexible tubing is used between the air scoop and the heater tube. The bypass valve which controls the entrance of air into the cabin should allow a flow of air through the tube at all times, spilling it to the outside air when not wanted for heat. This keeps the heater tube from getting hot enough to deteriorate quickly. A slight increase in exhaust manifold size is necessary to accommodate this tube and not restrict the exhaust passage. This entire heater assembly, including the extra weight of the enlarged exhaust stack, heater controls, and brackets, has been built to weight 18.6 lb for installation on a 10-place plane.

Preliminary calculations for hot-air-exhaust heaters generally miss the actual heat-transfer and inlet-air temperatures by wide margins because of the uncertainty of predicting operating conditions. Location of the cold-air intake, resistance of the ducts to air flow, cabin pressures, and heat conditions in the exhaust are variables peculiar to each particular plane which must be determined by experiment. More than sufficient heat will usually be obtained if the air tube runs the entire length of the exhaust tail stack, which in a 10-place plane is about 5 ft. The tube should be of sufficient diameter to deliver about three-quarters the required amount of ventilating air.

Referring again to the combination heater and ventilator-duct system used in the Pilgrim 100-A 10-place transport plane, the cross-sectional areas of the various air ducts are as follows:

Wing-butt air intake = 8.3 sq in. each
Upper ventilation duct = 2.5 sq in. each
Heater pipe = 4.8 sq in.
Lower heater ducts = 8.0 sq in. each.

The inlets from the wing butts connect with both the upper ventilator and lower heater ducts. Butterfly valves in the connecting pipes allow cold air to be fed to the heater ducts which supply ventilation in hot weather and modulation of hot-air temperatures in winter. Undiluted hot air from this exhaust heater enters the ducts at about 140 F with outside air at +10 F. This temperature is reduced by cold air from the ventilator-duct valves. This combined heating and ventilating system can be applied to hot-liquid and vapor heaters with equal success.

The hot-air-exhaust heater is limited in application to planes having an engine in the nose of the fuselage. Planes having only outboard engines are more efficiently equipped with hot-liquid or vapor heaters.

2 Hot-Liquid Heater. The principle of operation of this type is the same as that of hot-water heating systems for buildings. Fig. 4 is a diagrammatic layout of such a system applicable to airplane use. The liquid used in this system may be glyco-ethylene (commercial Prestone), quenching oils of high boiling point, or any other liquid which has a boiling point of over 300 F, which will not freeze or congeal at —20 F, which is non-corrosive, and which will have a flash point high enough to be free from fire hazard should the system develop a leak.

It is not economical from a weight standpoint to use liquids of lower boiling point than 300 F, because the weight of the radiator goes up as the temperature difference between liquid and air is reduced. Since airplanes often stand outside, the liquid used in the heating system must flow freely enough to be easily pumped through the pipes when the engine is started in the coldest weather. Some liquids that could be used because of their physical characteristics must be eliminated because of their corrosive effect on steel, copper, bronze, aluminum, duralumin, or any other metal used in aircraft construction. The solution of the liquid in water especially must not be corrosive, since it is bound to be spilled over the structure and will absorb water from rain or the natural humidity of the atmosphere. The odors from the hot liquid must not be disagreeable or strong, since in spite of normal care they will get into the cabin from whatever small amount may be spilled in the engine compartment.

The liquid is force circulated through the system by an engine-driven centrifugal or vane pump driven either through a flexible shaft or directly from a suitable attachment on the engine, usually the gun-synchronizer drive shaft. The pump should take the cool liquid from the radiator and force it through the heater. It will handle liquid at 200 F or above, and clearances in the pump must be relieved to prevent binding at the boiling point of the liquid used.

The heater has been successfully constructed as a series of coils in the enlarged exhaust tail stack. Another type has a tube running through the manifold for a sufficient distance. A third type, shown in Fig. 4, has an independent heater through which some of the exhaust gases are bypassed. A butterfly valve manually and thermostatically controlled governs the flow of exhaust gases through the heater. The independent heater, although heavier and more expensive to build than the coils in the exhaust pipe, has the advantage that it is not continually in the heat of the exhaust and the heating surface may be made as large as desired to supply heat for any extreme condition of cold weather. If so constructed, it must be thermostatically controlled, the thermostat cutting off the exhaust gases from the heater.

No thermostats are on the market which can be applied to such use on an airplane without redesign to reduce weight and size. One type of thermostat has the sylphon bellows encased in the tube which is immersed in the liquid to be controlled. This type has given trouble from failure of soldered and compression joints and from deterioration of the bellows while operating at the high temperatures encountered. The bellows should be outside the heater and actuated by the expansion of liquid in a separate bulb or tube in the heater. The manual control to the valve cutting off the exhaust gases must be designed to permit the thermostat to function at any position of the control. The heater may be built of 0.035-in. stainless steel, with exhaust inlet and outlet ducts of 0.049-in. material. Particular attention must be paid to eliminating all possible gas pockets in the heater, as these will cause surging and hammering in the heater lines and are a cause of failure of this type of heater. The thermostat must be located at the exit point of the heater where it will be surrounded by the hottest liquid, otherwise there is possibility of the liquid reaching the boiling point, causing it to hammer in the lines.

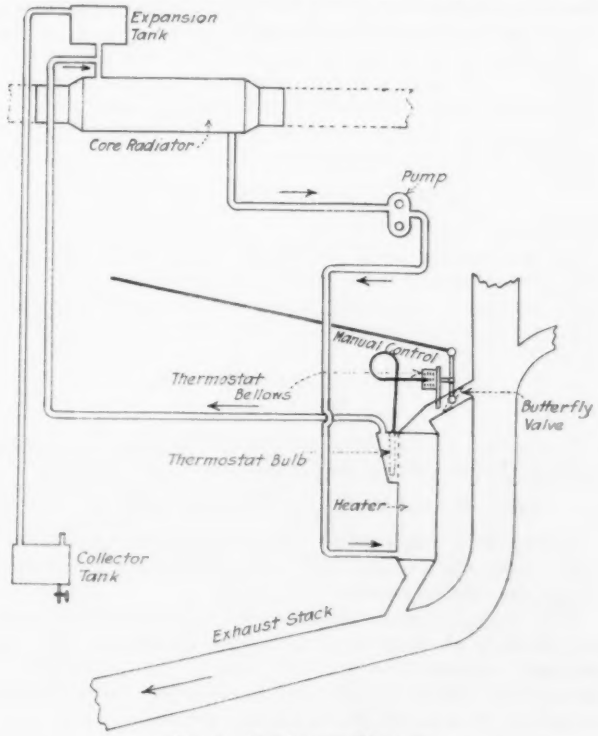


FIG. 4 HOT-LIQUID HEATER

The heating surface of coils or tubes located directly in the exhaust stack must be proportioned in relation to the radiating surface of the radiator so that the heat added to the liquid will be completely taken out on each complete cycle of the liquid and a temperature equilibrium be reached before the liquid reaches the boiling point.

Calculations for heat required must be made on the maximum operating condition of —20 F atmospheric temperature. The expression for heat required to maintain the cabin at 70 F is:

$$H = H_A + H_C - H_P$$

Continuing with the 10-place transport plane:

H = heat required to raise 14,000 cu ft of air per hour from —20 F to +70 F

$H_A = WC_P (t_2 - t_1)$.

The weight of 14,000 cu ft of air at —20 F = 1265 lb.

$H_A = 1265 \times 0.2375 (70 + 20)$
= 27,000 Btu per hr

H_C = heat lost by radiation from the cabin.

This is a very difficult quantity to calculate because there are no accurate data on heat radiation from glass, doped fabric or cabin interior surfaces, and plywood surfaces at the air velocities obtained in flight. Even data on heat radiation from surfaces such as a metal monocoque fuselage flying at speeds of 140 to 180 mph are not available.

Taking the best data available from heating and ventilating text books, the heat radiation from the fabric-covered 10-place transport was estimated to be 7100 Btu per hr.

H_P is the heat generated by the passengers in the cabin. This quantity is not large, and since the cabin temperature must be the same for one person as for ten, it will be neglected.

The total $H = 27{,}000 + 7100 = 34{,}100$ Btu per hr. The temperature of the entering air would be:

$$34{,}100 = 1265 \times 0.2375\ (t_2 + 20)$$
$$t_2 = 94\ \text{F.}$$

Using glyco-ethylene as the liquid to be circulated, the next step is to determine the size of the pump required to give the right circulation. Assume a 50-deg drop through the radiator. The specific heat of commercial Prestone is about 0.7 at 300 F. Then:

$$34{,}100 = W_z\ 0.7\ (300 - 250)$$
$$W = 975\ \text{lb per hr.}$$

The weight of the liquid at 300 deg is about 9.3 lb per gallon. The capacity of the pump would be:

$$975/9.3 \times 60 = 1.75\ \text{gal per min.}$$

A very small centrifugal or vane pump driven from the gun-synchronizer shaft, either direct or by a flexible shaft, will easily deliver the required amount.

The estimate of the heating surface in the exhaust tail stacks will be only an approximation unless experimental data are procured from which to arrive at a conclusion. A preliminary estimate can be made by taking the mean coefficient of heat transmission as 14 or 16 Btu per hr per sq ft per deg mean temp difference. Then:

$$SUd = 34{,}100$$

S = heating surface in sq ft
U = 14 Btu per hr per sq ft per deg mean temp difference
d = mean temp difference between the liquid and exhaust gas.

$$d = \frac{t_1' - t_f}{\log_e \frac{t_1'}{t_f}}$$

t'_1 and t_f = initial and final temperature difference between the liquid and gases

$$d = \frac{850 - 700}{\log_e \frac{850}{700}} = 770$$

$$S = \frac{34{,}100}{14 \times 770} = 3.2\ \text{sq ft.}$$

The liquid is circulated to the radiator from the heater. This radiator is placed as close to the heater as the layout of the plane will allow, so that unnecessary weight may be eliminated. Preferably it is placed ahead of the fire wall or at some point interposed in the air duct which is remote and shielded from passengers and pilot to eliminate the danger of scalds in the event of a wreck or failure of the radiator. Technology paper No. 211 of the Bureau of Standards gives the most complete data on radiators suitable for use with this type of heater. Especial attention should be given to radiators listed as E_6, A_7, C_{12}, and C_{14} in this report.

None of these radiators are suitable for use in the hot-liquid heater in their standard construction because they are fabricated with solder that is weak at the operating temperatures of the circulating fluid. Silver solder or a solder of equivalent strength should be used in constructing the radiator. A leak in the radiator, while not actually dangerous, may have a serious effect on the passengers because certain hot liquids when suddenly coming in contact with cold air form dense fumes which immediately fill the cabin, throwing the passengers into a frenzy.

At the highest point in the line between heater and radiators a line passes up to an expansion tank. This tank should be as high as possible above the radiator and should hold the entire liquid supply. A vent line from the top of the tank runs down to a collector tank located in the bottom of the plane to catch whatever vapor may condense in the vent line. This tank should have an accessible drain that is not difficult to get at, as this tank will be drained periodically. The liquid used in this system is usually expensive, and it pays to collect the small amount that might be wasted. The system should be filled through a funnel in the expansion tank.

Attempts have been made to eliminate the pump and operate the system on the thermo-syphon principle. These have met with failure or unsatisfactory performance. The difference in density between the hot and cold columns of liquid is too small to produce a rapid flow of liquid. The heat transfer is too slow to properly heat the cabin, and the heat is slow in starting after the engine is started. One such installation required ten minutes from the take-off of the plane before heat reached the cabin in sufficient quantities. The weight of this system was more than double that of a force-circulated system because the pipes had to be 1 in. in diameter on the hot side and $^3/_4$ in. on the cool side to prevent surging and hammering from boiling in the heater, and the vertical distance between heater and radiator had to be as great as the fuselage would permit. The force circulation system was successful with $^3/_8$-in. lines and no head between heater and radiator. The weight of the liquid in the thermo-syphon system was 14 lb as against 4.5 lb for the force-circulation system, using a separate heater unit.

Several installations have been made which substitute for the radiator, tubes carrying the hot liquid around the cabin inside air ducts in which are individual openings. This practice is not advisable, since the weight is very much greater than a unit radiator system, and it subjects the passengers to the danger of being scalded in the event of failure of a pipe or fitting either during service or in a crash.

The main advantages of the liquid heating system are that it entirely eliminates the danger of carbon-monoxide poisoning, which is a possibility in the hot-air heater. Provided that the correct installation is made, it eliminates odors such as scorching and frying oil smells, which often enter from hot-air-exhaust heaters. It is more easily adaptable to planes having outboard engines, as the hot liquid is easily circulated through insulated pipes to any desired location for the radiation. Its disadvantages are its weight and maintenance expense. It has been estimated that each pound of weight saved in airplane structures and equipment is worth $90. The weight analysis estimated from two experimental installations is as follows.

	Hot-liquid heater (independent heater), lb	Exhaust pipe coils, lb
Heater	7.0	3.0
Connecting tubes and shut-off valve	3.0	0.0
Lines	1.5	1.5
Radiators and brackets and fittings	6.0	6.0
Liquid (Prestone)	9.0	3.0
Pump and brackets	2.5	2.5
Pump drive mechanism	2.5	2.5
Thermostat and brackets	2.0	0.0
Expansion tank and brackets	2.0	2.0
Total	35.5	20.5

On the basis of $90 a pound the exhaust-pipe-coil type is worth $1350 more than the independent type. From a maintenance standpoint it is also very much more economical. In Alaska and Canada the independent-heater type might have its advantages because of its flexibility in operation.

Vapor Heaters

Several types of vapor heaters have been experimented with in aeronautical work, but have not been used to any extent in commercial planes. Few data are available from experimental work on any of the various types of vapor heaters.

Hermetically Sealed Heater System

This system consists of a boiler and radiator connected together with suitable tubing and evacuated of air. The boiler is located in the exhaust tail stack and consists of either coils of tubing or a single streamline container functioning as a flash boiler. The radiator may be any of the conventional types capable of taking pressures in excess of 400 lb per sq in. A small quantity of a liquid of low freezing point is introduced into the system before the air is evacuated. The internal volume of the system is small, and the quantity of liquid is carefully measured, since it is nearly all in the vapor state and under pressure at all times when the engine is running. Vapor from the boiler passes up to the radiator, is condensed on the walls of the radiator, and trickles back to the heater. The pressures in this system are high, usually between 300 and 600 lb per sq in.

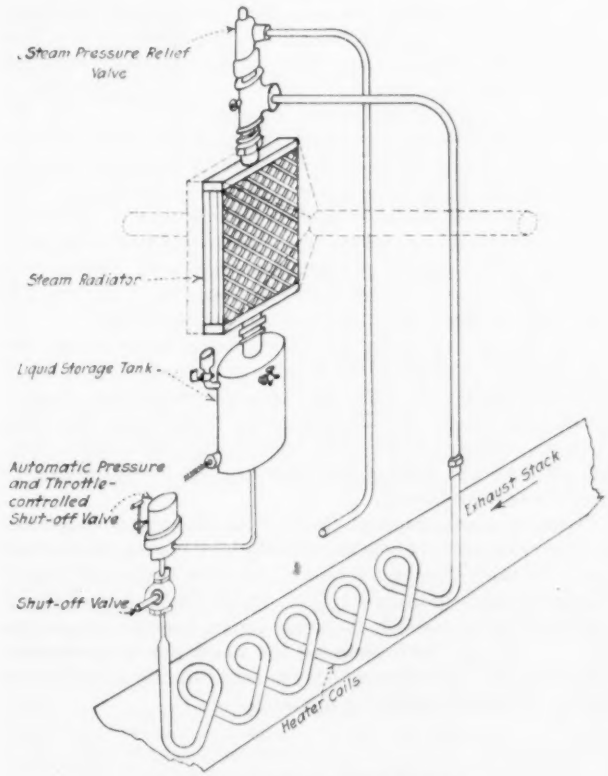


Fig. 5 Steam-Pressure Heater

The success of this system lies in keeping the radiator as close to the boiler as possible and eliminating all pockets that can catch and retain liquid. The volume of the liquid is necessarily very small, and a few small pockets in the radiator and lines will keep enough liquid out of circulation to impair or entirely spoil the functioning of the system.

No data as to weight are available on this system. It is compact, but the radiator and lines are heavy to withstand the high pressures. Attempts have been made to apply this principle to a system which piped the vapor about the cabin to individual radiators, but the system failed because all the liquid was caught in pockets in the lines and kept out of circulation. This system is best applied on planes having an engine mounted in the nose of the fuselage so that air ducts run directly from radiator to cabin. The high pressures make it undesirable because of danger from explosions in the event of crash.

Steam Pressure System

This system is the same in principle as an ordinary house-heating unit, but with added control features. Fig. 5 shows a system which has worked successfully as an experimental unit. A flash boiler consisting of coils in the exhaust stack vaporizes and superheats the heat-transfer medium, which in an experimental unit was a mixture of alcohol and water in proportions to withstand

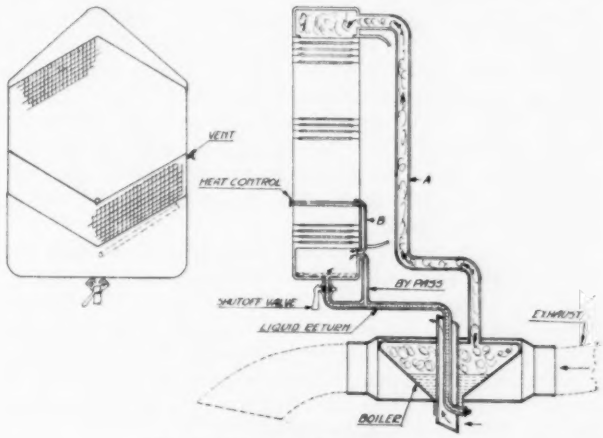


Fig. 6 Huffman Vapor Heater

zero weather without freezing. The superheated system passed to a radiator in the heater duct, from which the condensate dropped by gravity to a storage tank of 1 qt capacity. A control and regulator valve in the line between the collector tank and boiler regulates the flow of liquid. This valve is designed to automatically shut off the flow when the pressure exceeds 20 lb, and also when the engine is throttled below one-quarter speed. This prevents the pressure from rising and blowing off the safety valve, which would quickly use up the liquid in the system, and it prevents the liquid from filling the coils when there is insufficient heat in the exhaust stack to vaporize it. A second valve is closed when the ship is idle, to prevent water from accumulating in the coils if the throttle should be opened.

The boiler coils are constructed of $^3/_4$ by 0.049 stainless-steel tubing. The delivery lines are $^3/_4$ by 0.035 stainless-steel tubing. The radiator should withstand 150 lb pressure with safety, although the safety valve is set to blow off at 50 lb gage. All joints and seams must be strong at high temperatures, because the coils are designed with sufficient surface to superheat the steam to a high temperature. The collector tank is made from a $2^1/_2$ by 0.049 steel tube and for an installation in a 10-place plane, with a capacity of only 1 qt. The filler and vent cocks are in this tank.

The automatic control valve is of special design, using syphon bellows to prevent leakage. The shutdown valve is also a bellows valve so that no leakage will occur. The return lines are $^3/_8$-in. copper tubing.

A small orifice is placed in the line between the collector tank and boiler coils to limit the feed of liquid to the boiler so that the

pressure will not build up above the blow-off point of the safety valve should the automatic control valve get out of order.

By using specially designed fittings and valves, this system can be made to weigh about 12 to 18 lb for a 10-place plane. Its initial cost is greater than for the hot-air heater and about the same as the hot-liquid heater. It is easily adaptable to planes having an engine mounted in the nose of the fuselage and to multi-engined planes if the radiator can be placed above the boiler coils and a good slope maintained in the return lines from radiator to boiler. This being impossible, an engine-driven pump with a metering valve can be used to feed the liquid to the flash boiler from the collection tank. This system is capable of rapidly adjusting itself to varying conditions of air temperature, and if thermostatically controlled, with the thermostat located in the cabin, will keep the cabin at a very even heat. By increasing the heating surface of the radiator, the amount of heat delivered to the cabin may be proportionately increased without changing any other part of the system. This system also has the advantage of delivering steam to the radiator at the same temperature at all altitudes normally obtained in transport work. Some objection is raised to this system on the ground that there is danger from steam burns in event of a crash. If this system is located ahead of the fire wall, there is little danger of any injury from such cause, as the volume of the vapor is small and very little liquid is in contact with hot surfaces. The liquid in the collector tank is far enough below the boiling point so that it will not flash into steam when the pressure is relieved. The pressures are too low to cause damage from explosion.

Cabin heaters are in operation utilizing the hot liquid from solution-cooled engines. A part of the liquid from the engine is circulated through a radiator in the hot-air ducts. The air which passes through this radiator is led to the cabin or spilled outside by a bypass valve in the air ducts according to the need for heat in the cabin. The radiator is a part of the cooling system for the engine, and if made sufficiently large, it may be possible to reduce the area of the main radiator slightly.

Because of the low temperatures of the circulating liquid, the radiator tubes must be long to raise the air to a sufficiently high temperature. This system may cause complications in the engine-cooling system unless very well laid out. It seems to offer little advantage from a weight standpoint over the exhaust-heater liquid cooler or the steam coolers.

Huffman Vapor Heaters

The system consists of a boiler and radiator with the necessary steam and liquid return line connecting the two and a means of driving the heat from the radiator. The boiler is mounted in an enlarged section of the exhaust line and the radiator at some convenient point within the cabin. The radiator is divided into two sections by means of a partition, the upper section being about four times the size of the lower section. A reservoir having the approximate capacity of the boiler forms the lower part of the radiator.

The upper and the lower sections of the radiator are connected together by means of a tube extending from the lowest point of the upper section to the lowest point of the bottom section. A hand valve is installed in the line connecting the two sections. The lower section at its uppermost point is vented to the atmosphere. The steam line from the boiler enters the upper section of the radiator at its top.

The liquid return line is attached to the bottom of the radiator reservoir with a hand shut-off valve directly below the point of attachment. A bypass line from the liquid return line enters the top of the reservoir.

Referring to Fig. 5, liquid in the boiler is heated to the boiling point by the exhaust. Steam thus generated rises and fills the radiator through pipe *A*. Heat is driven from the surface of the radiator by means of a fan electrically or otherwise operated, or by means of air brought in from the outside through a suitable tube so arranged that the air will flow through it due to motion of the vehicle or airplane.

The size of the boiler with respect to the radiator and the quantity of heat supplied to the boiler are such that under normal operating conditions more steam is generated than can be condensed in the radiator. Thus it will be seen that steam will flow from the upper radiator section through the heat-control valve and pipe *B* to the bottom of the lower radiator section and rise toward the top, which is vented. If more steam than could be condensed in the lower radiator sections were permitted to pass through the heat-control valve, it would escape through the vent, with the result that the liquid would in a short time boil off.

To prevent this loss the heat-control valve is turned "in" until less steam than can be condensed in the lower radiator section is permitted to pass through it. A stop provided on the valve is then adjusted to prevent the valve from being opened beyond the point where a loss of steam through the vent would occur. After the initial adjustment of the "stop" on the valve, no further adjustments are necessary.

When the heat control valve is turned "in," restricting the flow of steam through the radiator, pressure (varying from a few ounces to 2 lb, depending upon the height of the reservoir above the boiler) is built up in the radiator, pipe *A*, and the boiler. This pressure results in the lowering of the liquid level in the boiler and a rise in the level in the return line. When the liquid in the boiler is reduced, the quantity of steam is also reduced; therefore the system automatically permits only sufficient liquid to remain in the boiler to generate the amount of steam required.

The design of the boiler and its installation in the exhaust line are such that a sump in the bottom and a small portion of the bottom surface of the boiler are below the temperature required to boil the liquid. This feature assists in stabilizing the liquid level and prevents surging and "percolating" when, due to pressure in the boiler, most of the liquid has been forced through the return into the reservoir.

When no heat is required, the liquid in the boiler is transferred to the reservoir. This is accomplished by closing the shut-off valve on the bottom of the reservoir and the heat-control valve. When the heat-control valve is closed, all liquid in the boiler is instantly forced through the liquid return line and bypass into the reservoir due to pressure on the steam side of the system. With the shut-off valve closed, the liquid cannot return to the boiler; therefore generation of steam stops.

AER-55-8

Artificial Pay Load for Coming Commercial Airships

By T. L. BLAKEMORE,[1] LAKEHURST, N. J.

The most prominent field for the lighter-than-air vessel is across long stretches of water. The "Graf Zeppelin" confirms this by selecting as its first commercial venture the route between Germany and Brazil. There is a question whether the seagoing airship will merely be a high-speed, extra-fare auxiliary to the steamship or whether by radical improvement over present limited pay-load capacity it can be developed as a direct competitor to passenger-carrying ocean vessels at somewhat near to existing rates. The author suggests a number of means for artificially increasing pay load.



IT HAS BEEN generally recognized for some time that the most promising field for the lighter-than-air vessel commercially is across long stretches of water. In fact, we find the *Graf Zeppelin* confirming this choice of field by electing a direct ocean haul from Germany to Brazil as its first commercial venture. It seems conservative, therefore, to accept the limitation that in the future the only existing form of transportation service which the coming airship will parallel is that offered by the steamship over long distances at sea. The commercial airship should today be considered definitely and solely in the steamship's field. It seems to be no question of the airship versus the airplane or the airship versus the railway.

There is a question, however, as to whether the seagoing airship will take its place in the coming transportation scheme of things merely as a high-speed, extra-fare auxiliary to the steamship schedule or whether by radical improvement over present limited pay-load capacity the lighter-than-air craft can be developed to the point where it will offer direct competition to passenger-carrying ocean vessels at somewhat near to existing rates. In other words, is there any immediate promise of so improving the percentage of pay load of the airship or (its corollary) of decreasing the operating costs to such an extent that, while retaining the great advantage of speed over the steamship, the lighter-than-air craft can be operated at a rate more in keeping with established ocean fares?

The majority school of thought to date believes that the airship will be operated for some years as an extra-fare auxiliary to surface craft, using high speed, novelty, and added comfort as justification for the higher passenger rates charged. The author concurs in the general opinion that these factors, in themselves, will attract patronage and are sufficient reason for the excess passenger rates over the steamship, initially; but if we are to judge from the history of all other forms of human transportation, there will shortly come a time when it will be necessary to earn more dividends at lesser rates to the passenger; in other words, to carry more pay load or to cut operating costs, or both. As a natural means toward this end, the major effort of airship designers has been to increase pay load "statically" by recourse to larger and larger volumes. For it is common knowledge, and easily shown in theory, that any increase in airship volume has the advantage of automatic increase in per cent of useful load, of which pay load is a proportion; and the theoretical limit of size seems to be well beyond the volume of any airship flying or projected today. Practically, however, this increment in pay load due to increased size is not being realized in full proportion to theory. Without going into the many reasons for this falling off of the "static" pay load, it is believed sufficient for the present to point out that this natural method of increasing pay load in future airships has perhaps been overstressed; and if so, it may be found not only advantageous but economically necessary to develop artificial means for increasing pay load, in order that the airship may become more quickly practicable, commercially.

The purpose of this paper, then, will be to present the cumulative advantage of minor means at hand or which can be developed for artificially increasing pay load without recourse to excessively large volumes.

Fortunately, we have already begun to realize increased pay load artificially. The swiveling propellers of the *Akron* are, in a way, the first applied effort in this direction. For it has already become operating practice to lift this ship off the ground, statically "heavy" by about 10,000 lb, using the vertical dynamic thrust of its propellers. After the ship has reached a safe altitude, this extra load is gradually transferred, as the ship gets under way, to a component of its forward motion, and is directly creditable as artificial pay load.

On reference to Tables 1 and 2, it will be noticed that the *Akron*, of 6,500,000 cu ft capacity and 60 knots cruising speed, has a useful load of about 40 per cent, or 160,000 lb. Of this useful load, it is estimated that 16,000 lb would represent the proportion of actual static pay load on a given voyage, say from Lakehurst to Friedrichshafen, Germany. This amounts to around 4 per cent of the total lift of the ship. It should be realized, further, that the swiveling propellers, as designed for this ship, are pitched for down-thrust rather than up, and have developed over 15,000 lb in the haul-down position. It is conceivable, then, that in commercial practice, in which the purpose is to get the greatest possible pay load off of the ground, these propellers could be designed for maximum up-thrust, even though it might lead into the development of adjustable-pitch propellers for this

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 SOME COMPARATIVE FACTORS OF STRENGTH AND PERFORMANCE IN U. S. S. "SHENANDOAH," "LOS ANGELES," AND "AKRON"

	"Shenandoah"	"Los Angeles"	"Akron"
Gross lift, lb	124,500	153,000	403,000
Useful lift, lb	44,100	62,600	170,000
Useful/gross	0.354	0.410	0.422
Fuel load, lb	26,000	40,000	127,000
Fuel per hour at 50 knots, lb	580	510	705
Endurance at 50 knots, hr	45	78.5	180
Propulsive coefficient at 50 knots	40	44	60
Specific fuel consumption, lb per hp-hr	0.53	0.47	0.47
Horsepower at 50 knots	1,050	1,090	1,500
Air volume, cu ft	2,300,000	2,800,000	7,250,000
Designed speed, knots	50	68	72.8
Designed speed, ft per sec	85	115	123
Maximum velocity of sudden gust, ft per sec	17.0	20.6	56.0
Bending moment due to gust, ft per lb	650,000	1,300,000	9,750,000
Maximum stress due to gust, lb per sq in.	3,800	5,000	10,800
Static and secondary stresses, lb per sq in	9,700	9,900	5,300
Maximum total stress, lb per sq in	13,500	14,900	16,100
Ultimate strength of longitudinals, lb per sq in	26,000	33,000	37,000
Factor of safety, lb per sq in	1.9	2.2	2.3

TABLE 2 COMPARATIVE WEIGHTS OF U. S. S. "SHENANDOAH," "LOS ANGELES," AND "AKRON"

	"Shenandoah,"		"Los Angeles,"		"Akron"	
	Weight, lb	Lift, %	Weight, lb	Lift, %	Weight, lb	Lift, %
Hull structure	30,363	24.39	31,310	20.46	91,622	22.73
Fins, rudders, etc.	4,086	3.28	3,190	2.08	12,680	3.15
Cover, gas cells, valves, etc.	16,039	12.88	17,650	11.54	36,688	9.10
Power plant	14,849	11.93	20,800	13.59	50,412	12.51
Water recovery	2,055	1.65	2,332	1.52	10,400	2.58
Fuel system	3,204	2.57	2,610	1.71	7,100	1.76
Ballast system	906	0.73	627	0.41	2,950	0.73
Electrical, radio, etc.	1,119	0.90	2,555	1.67	3,760	0.93
Quarters, compartments	672	0.54	2,270	1.48	9,687	2.41
Mooring and handling	2,130	1.71	1,410	0.92	3,150	0.78
Control car, miscellaneous	4,980	4.00	5,675	3.71	4,451	1.10
Total weight empty	80,403	64.58	90,429	59.10	232,900	57.79
Gross lift at 95% full of gas, lifting 0.062 lb per cu ft	124,500	.....	153,000	.....	403,000	.....

NOTE: Taken from article on "Progress in Airship Design from U. S. S. *Shenandoah* to U. S. S. *Akron*," by C. P. Burgess, in Journal of the American Society of Naval Engineers, August, 1931.

primary purpose. Should this be done, the addition of 15,000 lb to the pay load of the ship at the start of a flight is practicable and well within the present bounds of operation safety, due to an equal amount of reserve water on hand at this time for emergency purposes. The credit, therefore, from this first means of taking on artificial pay load would be 15,000 lb, or 3¾ per cent more pay load.

Again, should the ship be brought up to 70 knots initial speed, shortly after take-off, its dynamic lift (as an airplane) at a 6-deg angle of attack and an elevator angle of 0 deg is around 15 short tons, or 30,000 lb. The ship is already 15,000 lb heavy from the swiveling-propeller take-off, so that there is still lift and room for a matter of 15,000 lb more of pay load that could be carried if it could be gotten aboard while the ship is under way. The plan immediately suggests itself to employ a number of shuttle planes, either at Lakehurst or stationed at points along the land end of the line of flight, to bring aboard mail and express aloft—thereby increasing the artificial pay load by this amount, or as much of it as could be safely handled with our present technique of aerial transfer and airship practice. It should be realized that, as a guarantee of the safety of such an operation, there is aboard 100,000 lb of fuel, in addition to the 16,000 lb of water on hand after the dynamic take-off. The ultimate net gain, then, from this second operation could be 15,000 lb, or 3¾ per cent more pay load; and the penalty would be nothing more than a slight slowing down of the ship during its first 15 hours and the possible wastage of a proportion of reserve fuel in case of unforeseen difficulties. During these first 15 hours of flight, the water-recovery apparatus has presumably been rejecting water at the rate of 1 ton per hour, thus bringing the ship back to safe equilibrium and normal cruising flight.

As a third method of getting more pay load off the ground, for any rigid airship of given volume, the suggestion has been made to artificially superheat the helium-gas cells, thereby creating more initial static lift by reason of (*a*) a decrease in the density of the contained gas and (*b*) an increase in the normal volume. Assuming that it will continue to be the practice of airship commanders to conserve helium gas by maintaining a hangar or ground volume at take-off around 95 per cent of full cell inflation (this in order to permit operating the ship at a safe "pressure height" of around 2000 ft without valving helium), the plan would be to artificially heat the airship gas for some hours prior to take-off, so that it would leave on a voyage with an actual static increase in lift—in other words, more pay load at take-off.

After getting under way, the flow of air over the outside cover would rapidly dissipate the excess heat contained, causing a gradually increasing heaviness due to contraction of gas volume. Just how rapidly this "superheat" would dissipate or how gradually the heaviness would increase is a matter of conjecture at this time owing to lack of complete test data; but there is very little question, based on tests made by Lieut. Calvin M. Bolster on the *Los Angeles* during 1929, that sufficient cheap heat can be supplied to an airship prior to flight, in the hangar or at the mast, to give it an initial superheat over the surrounding air of 10 F. And on this basis, an additional "artificial" pay load of 9000 lb can be lifted off the ground, under normal conditions. In order to guarantee the safety of this third means of gaining pay load, it is necessary to have ready at hand an equivalent amount of disposable weight. The author suggests that it may be found to be good economic practice to have standing by 9000 lb of non-insured express matter or freight, equipped with quick releases and large parachutes, for immediate release in case of temperature inversion or other operation difficulties. Also, in order to retard a possible rapid dissipation of superheat, it may be found practicable to keep the air space hot between gas calls and outer cover (a fortunate advantage of rigid airships) by utilizing the rejected heat from the water-recovery apparatus. It should be kept in mind, in reference to the last, that immediately after take-off the engines will be at full power and the recovery of water will not be required for some hours thereafter, thereby allowing hot air from the shielded condensers to be blown directly into the insulating air space. The object in this is not to obtain further heat, but to retain what we have until the normal fuel consumption of the ship will reduce the loss of lift due to natural cooling of the gas. As a further guarantee of safety, in this respect, it is pointed out that for a great percentage of normal flights weather reports could be relied upon to give the superheated airship the advantage of the morning sun-up effect to sustain this artificial superheat. It is believed conservative, therefore, to credit the impressed superheat plan with at least 9000 lb of artificial pay load, or an increase over the straight static pay load of 56 per cent.

In the foregoing, only the direct effects of impressed superheat at take-off have been considered. In reality, if it should prove out that the application of direct heat from the condensers in flight to the insulation air space of the hull is found practical, the indirect effect of further safety to the airship against ice and wet snow formation on the top side and the control surfaces is hard to estimate. An approximation could be based on the excess amount of normal reserve fuel (or ballast) which a given commercial airship will have to carry for, say, a winter flight across the Atlantic, in anticipation of the formation of ice or sleet. Dynamically, the *Akron* has already flown 16,000 lb heavy from this cause in the early stages of a protracted cruise to the tropics. Fortunately, the ship gradually lost the load due to increasing temperatures, rather than accumulating further ice load; but had it been a commercial ship operating on definite schedule

across the Atlantic, it is probable that for that particular time it would have had to steer a course well south of freezing temperatures. And such a roundabout course, leaving favorable winds out of consideration, would have meant as much as 10 per cent more fuel to negotiate the passage. This extra fuel would have cut into the pay load by an equal amount, and from this line of reasoning, the author believes it conservative to say that any commercial airship possessing a positive means of snow and ice prevention, such as indicated, has a positive advantage in useful load of at least 5 per cent of the total fuel, or 31 per cent over the pay load of any airship not so equipped. The difference between 5 per cent and 10 per cent in fuel allows for added static weight of the necessary heating blowers, ducts, and equipment. Incidentally, the airship which can operate safely at northern temperatures has the sustained advantage of the increased lift of the colder atmosphere, and this would mean added pay load for the same fuel reserve.

For the fifth means for artificially increasing the static pay load of coming airships, it will be necessary to consider the effect of excessive parisitic drag on the present useful load and indicate possible increments by cutting down the drag. The *Akron*, for instance, has 880 sq ft of drag area, including the parasitic, and in spite of the excellence of this design and low drag area compared to other airships, the author believes that the commercial airship will find it necessary to better this drag in a number of ways. For instance, the present practice is to install sufficient water-recovery condensers in the form of surface-type aluminum panels to give 80 per cent water recovery at full power; and the drag area of this full power installation amounts to around 23 sq ft, or 2.6 per cent of the entire drag of the airship. While this full-power installation is no doubt necessary for a military airship, it should be realized that the commercial airship will operate mostly at cruising speeds on, say, four engines at two-thirds power; and on this basis, the obvious thing to do to further cut down parasitic drag on the airship is to install the eight full-power units in such a way that the four not in use are fully retracted within the hull and the four in use are slightly retracted. By so doing the water-recovery drag could be reduced to possibly 1.0 per cent of the total, and this expressed in terms of the full 100,000 lb of fuel carried on a given flight, would mean a saving of at least 1000 lb of fuel.

In a similar way, some practical means may soon be found for installing the water radiators on these eight engines as retractable units, so that when any group of power units is not in use, the parasitic drag on the ship can be automatically reduced by withdrawing those units within the hull line of the ship. This idea, of course, is not new, and in fact has already received the attention of the Bureau of Aeronautics for application to military airships. But with the coming commercial airship the importance of increased pay load may make such a step very necessary. It is estimated that an additional 1000 lb of pay load instead of fuel could be carried by the airship using retractable radiators.

One of the advantages peculiar to airships as compared to airplanes is that the full engine power installed for emergencies is rarely used. The *Akron*, for instance, operates most of the time on alternate four engines out of the eight, at two-thirds power (equivalent to 45 knots speed). From the point of view on maximum pay load of coming commercial airships, therefore, there is a possibility of cutting down the drag of the four idle engines by making all eight engines in turn retractable. At this date such a step no doubt will seem quite visionary and possibly hazardous. But when it is considered that airplanes are being "retracted" today into the hold of the *Akron* and that only a few years ago this very practical operating means of saving airship drag was just as radical, in a similar way it is believed to be a mere matter of time before commercial airships adopt this or some equivalent means for reducing the present excessive resistance and weight of power-plant installations. The concentration of power plants into four rather than eight units, as adopted for the Zeppelin *LZ-130* now nearing completion, may be considered a stop in this direction, in that the total drag area of power plants is being reduced, and, in theory at least, the pay load should be proportionately bettered.

The advent of the commodious auxiliary control station in the lower fin suggested the possibility of eliminating the present control car, thus incorporating the ship captain's bridge, navigating quarters, etc. in this aftermost station. This would automatically reduce the parasite resistance still further, say 1 per cent, thus saving another 1000 lb of fuel as well as eliminating the major control system with its cable and sheave gear now running the full length of the ship. The weight reduction thereby would be at least 1000 lb more, so that this change would, alone, increase the pay load $12^1/_2$ per cent.

It is difficult to see any further reduction in the parasitic drag of an airship of the type of the *Akron*. The ventilators and exhaust hoods are necessary and well streamlined, and fins and control surfaces are beautiful examples of fine aerodynamic form. The buffers or landing pontoons, possibly, are open to question as to whether or not these emergency ground-handling accessories are required on commercial airship routes operating between established airports equipped with mechanical handling gear. But for the purpose of this article it is assumed that any weight (or drag) saved by eliminating the buffers would be balanced by weight spent in additional local strength to meet the requirements of mechanical handling at the terminals.

From the foregoing discussion, it is believed conservative, therefore, to summarize the additional pay load accruing from possible savings in drag at a total of 4000 lb, or 25 per cent increase over the present static pay load.

Improvements in aerographic service and in terminal handling methods have also a direct bearing on the pay load of coming commercial airships. The first is that the average yearly time to cross the Atlantic can be lessened by proper weather advice, and in direct proportion to the dependability of such service, the fuel reserve for any given flight can be lessened. The second, as pointed out in another paper, "Ground Handling of Rigid Airships," by Lieut. C. M. Bolster, presented to this Society, is by lessening the elapsed time in taking off and landing. Aerograph service and mechanical-handling methods have already gone far toward guaranteeing quick and safe passages for airships of today, and for this reason are not properly to be included as further means for increasing pay load of coming commercial airships. The author's object, however, in mentioning these two factors in this connection is to point out that natural improvement in service, whether aerographic, terminal, or in flight, can be relied upon in the future to cut down the length of passage and by so doing reduce the fuel required. And this, in turn, adds to the pay load. If, for instance, instead of carrying 100,000 lb of fuel, the commercial airship can safely reduce to 90,000 lb for any given run, solely because of the definiteness of auxiliary service, then 10,000 lb of extra pay load can be carried—again an increase of over $62^1/_2$ per cent over the original.

Summarizing the cumulative advantage of the six major or minor means for artificially increasing pay load so far discussed, we find the figures as given in Table 3.

This somewhat startling total is based, of course, on the assumption that all six methods would prove fully 100 per cent practical, as estimated, and that no sacrifice in the safety of operations would be involved. It should be noted, also, that the total reduction in reserve fuel is the sum of the last three items only, i.e., 19,000 lb, or 19 per cent of the total of 100,000 lb, assumed to be necessary for a given flight of a ship of the *Akron*

TABLE 3

1	Dynamic take-off, lb	15,000
2	Shuttle planes, aerial landing, lb	15,000
3	Artificial superheat at take-off, lb	9,000
4	Ice and sleet prevention, reduction in reserve fuel, lb	5,000
5	Accumulated reduction in parasitic drag, lb	4,000
6	Lessening of average flight time, lb	10,000
		58,000
	Original "static" pay load, lb	16,000
	Combined static and artificial pay load, lb	74,000
	Increase, per cent	360

class. The other three items, totaling 39,000 lb, are purely "artificial," and are limited solely by the dynamic power of the ship immediately after take-off.

Returning to the main object of this paper, which is to point out certain artificial means by which present rates as estimated for airship travel can be reduced without resort to larger volumes, it means that, if fully applicable, there is promise of scaling down present estimates in the direct ratio of 16,000 to 74,000, or down to about one-fourth of the rate now necessary to pay dividends.

This estimate of reduction in passenger rates in coming commercial airships has been based solely on increased poundage of pay load. We should, from a commercial point of view, consider also the economic advantage of service improvement incident to certain features of the artificial pay-load plan. For instance, with positive ice and sleet prevention, the direct northern course across the Atlantic will (under favorable wind conditions) save the passenger time, and reduce proportionately the flight operation cost of the airship. Again, the added safety of this feature should attract patronage and increase the load factor. It is beyond the scope of this paper to estimate the rate advantage of such features, nor is it possible to anticipate and list all of the artificial means by which coming airships can legitimately carry more pay load. The author feels that the purpose will have been served if he succeeds in pointing out here a few of the prominent fields of airship research which may individually or collectively be the ultimate means for bringing the airship passenger rate down to that of its surface competitor, the steamship. The basic object is of course to stimulate thought along these lines.

AER-55-9

Combustion Processes in Aircraft Engines[1]

By H. K. CUMMINGS,[2] WASHINGTON, D. C.

This paper presents three general types of combustion which occur in homogeneous mixtures of explosive gases (preflame oxidation, normal burning, and the explosive wave) and reviews available experimental evidence as to the combustion processes which actually occur in spark-ignition engines. Oxidation is detected optically by changes in absorption spectra and chemically by analysis of samples removed from the combustion chamber. The progress of flame away from the spark plug has been followed by visual, photographic, and electrical means, with simultaneous measurements of pressure development. Spectra of visible radiation and measurements of infra-red radiation, obtained through windows in the cylinder head, assist in analyzing combustion phenomena. The characteristics of preignition and knock are understood, but the exact mechanism of knock remains obscure.



THE types of combustion which occur in conventional aircraft engines are those common to all internal-combustion engines with spark ignition. However, large bores, relatively high compression ratios, and the general reliance on air cooling in aircraft engines favor those undesired combustion phenomena (preignition and detonation) which may give local cylinder pressures many times the normal explosion pressure and which are accompanied by excessive heat dissipation to the cylinder walls. The requirement of low weight per horsepower and the fact that they are operated most of the time at over 70 per cent of full load make aircraft engines more likely than automobile engines to suffer mechanical damage from these phenomena.

In the paper attention will be directed mainly to combustion as studied in actual engines, but some consideration must first be given to the much greater volume of work which has been done under other conditions on gaseous mixtures such as constitute the working substance in the internal-combustion engine. Three distinct phases of the gaseous oxidation reaction (slow combustion without flame, normal burning, and the explosive wave) are possible, and all may occur in the engine. Work on the oxidation of hydrocarbons with special reference to slow combustion has been reviewed by Egloff, Schaad and Lowry (1),[3] and by Mardles (2). The usual methods are (*a*) measurement of pressure change as the gas mixture is heated slowly in a closed bulb and (*b*) analysis of gas mixtures after passage through a heated tube. Closely related to this work on oxidation rates and oxidation products, which will be considered further in a later section, is the measurement of ignition temperature which has been reviewed by Bridgeman and Marvin (3). They pointed out the wide variation in values reported, depending on the specific method and apparatus used. A "true ignition temperature," considered a property of the fuel, was defined, and a modification of the Tizard and Pye adiabatic compression method (4) was suggested for its experimental determination.

When a homogeneous mixture of explosive gases fed to a bunsen burner is ignited, the sharply defined inner cone is the zone within which the sudden transformation of the gases takes place. The study of such flames by Gouy (5) led to the important conclusion that the movement of the reaction zone with respect to the active gases is constant at constant pressure and independent of the mass movement of the gases in which it is propagated. The work of Stevens (6), using a soap bubble as a transparent spherical bomb of constant pressure with central ignition, established (*a*) that the uniform rate of propagation of the reaction zone when measured relative to the active gases is proportional to the product of their concentrations, for complex as well as for simple fuels, and (*b*) that, over a wide range of pressures, this uniform rate of propagation is independent of the pressure. These results show that in all cases of the "normal burning reaction process" the rate of transformation within the reaction zone is proportional to pressure and indicate that the ultimate processes are binary impacts as would be called for by the kinetic theory of chain reactions. In the constant-pressure bomb, the reaction zone is a spherical shell inclosing the reaction products and expanding radially at a constant rate, both in space and with respect to the active gases. In the constant-volume bomb, however, the rate of displacement of the reaction zone is not constant in space, and its rate with respect to the active gases is indeterminate, due chiefly to the mass movement of the latter. Only in case of a spherical bomb with central ignition does the reaction zone remain spherical.

In 1881, the explosive wave was discovered independently by Berthelot and Vieille (7) and by Mallard and LeChatelier (8). This mode of transformation propagates with a constant velocity of at least 1000 meters per second, the value depending on the composition of the gas mixture, and may result from normal burning in a closed container, but any means of producing in the mixture a compression wave intense enough to ignite the gases will start the explosive wave in those mixtures able to support it. The high speed of the explosive wave may be accounted for on hydrodynamic grounds according to the Chapman-Jouguet theory (9) or by means of the theory of chain reactions as proposed by Lewis (10). Velocities computed on the Chapman-Jouguet theory are in good agreement with actual velocity measurements by Dixon and others. An investigation of the limits of detonation in gaseous mixtures by Wendlandt (11) is concerned with the borderland between normal burning and the explosive wave.

The type of apparatus used by Dixon (12) to obtain photographic records of flame propagation in a glass cylinder where ignition was produced by adiabatic compression has been adapted by Duchene (13) for studying normal burning in a glass explosion chamber into which the charge is compressed by means of a piston. The charge is fired by a spark just before the end of compression, and the piston remains at top dead center. These experiments reproduce certain of the conditions which affect combustion in the engine cylinder more closely than is usual in experiments with explosions in constant-volume bombs. Four

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[1] Publication approved by the Director of the Bureau of Standards of the U. S. Department of Commerce.

[2] Chief, Automotive Power Plants Section, Bureau of Standards.

[3] Numbers in parentheses refer to bibliography at the end of the paper. (See also A. Egerton, *Jl. Inst. Petrol. Tech.*, 14, 656, 1928, for 75 references on detonation and related subjects.)

Presented at the Sixth National Aeronautic Meeting, Buffalo, N. Y., June 6 to 8, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

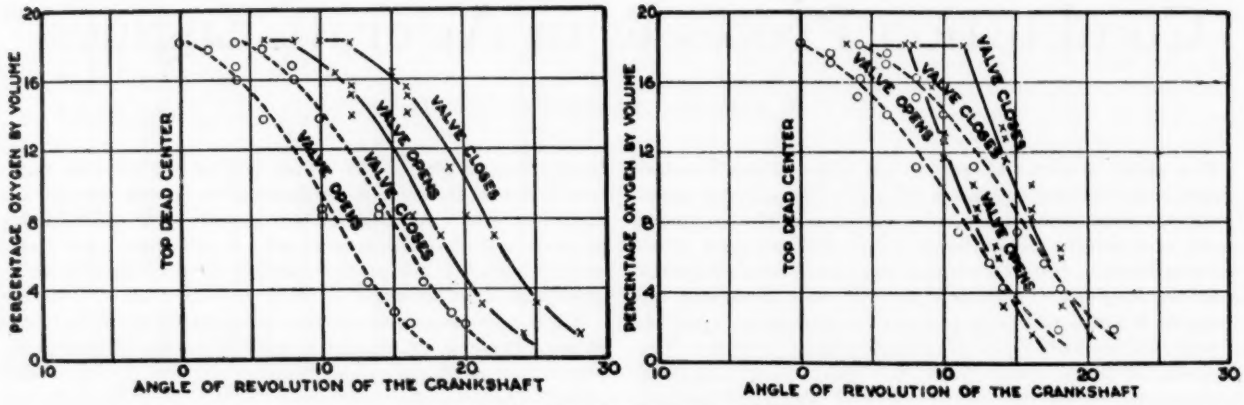


FIG. 1 OXYGEN CONCENTRATION CURVES

(At left, engine not knocking; at right, engine knocking violently. Engine speed, 1000 rpm. Spark advance, 25 deg. Mixture ratio, 80 per cent of theoretical air. Throttle, fully open. Fuel, gasoline. Compression ratio, 5 to 1. Broken line, sampling valve located 3 in., or 7.6 cm, from spark plug. Continuous line, sampling valve located 4¾ in., or 12.1 cm, from spark plug.)

fuels were used, and temperature, mixture ratio, and compression ratio were varied in the work reported. It is understood that this work, done under the direction of Jouguet, is still in progress, but no recent publication is available.

At the request of the Bureau of Aeronautics, Navy Department, the Bureau of Standards has undertaken a study of the effects of temperature, pressure, and radiation on hydrocarbon vapors, optical instead of chemical means being employed to detect molecular changes. In this work ultra-violet absorption spectra are obtained for each fuel vapor, both with and without the addition of air, at various temperatures and pressures. Spectra of aldehydes, alcohols, and other possible oxidation or decomposition products are compared with the hydrocarbon spectra in order to account for permanent changes in absorption. By inserting a pair of quartz windows in the cylinder head of a single-cylinder engine and passing a beam of light through the combustion chamber, it is possible to study changes in the absorption spectrum of the charge while the engine is being operated on any desired fuel. No test results have yet been released for publication. Measurement of pressure changes with time during the heating of various fuels with air will give data on oxidation rates.

In Callendar's second report on Dopes and Detonation (14), tests were mentioned in which a Ricardo E-35 engine was run under power and then motored over with the ignition switched off, in order to obtain from the exhaust samples of charge which had been compressed but not burned. The samples were reported to contain both aldehydes and peroxides. Subsequently, E-35 engines at Old Shoreham and at Cambridge University were equipped with water-cooled sampling valves for the purpose of obtaining gas samples at different points in the cycle, with the engine running under power. Thornycroft reported (15) that analysis of the gases withdrawn through a sampling valve, which was open for a very short period during the compression stroke, showed the presence in small quantities of alcohols and aldehydes. At Cambridge, Steele (16) found no evidence that samples taken during the intake, compression, or exhaust stroke contained peroxides, although reaction with potassium-iodide solution indicated the presence of an oxidizing agent. Work on oxidation of fuels in an engine prior to ignition was also undertaken at the California Institute of Technology in 1927, but no results appear to have been published.

The use of exhaust-gas analysis to determine completeness of combustion is a method of long standing, but the first attempt to follow the progress of combustion in an engine by chemical means appears to have been made about ten years ago. Preliminary studies of combustion in a low-compression oil engine with hot-bulb ignition were made by Brooks (17), who used a water-cooled auxiliary valve and copper cooling coil to obtain and rapidly chill gas samples at five points in the explosion stroke. A run under detonating conditions showed characteristic differences from normal-burning runs, such as the presence of hydrogen, carbon monoxide, and free carbon, and the application of this method to the study of combustion in Otto-cycle engines was proposed. A similar sampling valve was developed independently by Lovell, Coleman, and Boyd (18) and used to study the rate of burning of the charge in an air-cooled Delco engine. When the engine was operated at 1200 rpm on a maximum-power gasoline-air mixture appreciable burning appeared to start about 10 deg after ignition, one-half of the available oxygen was consumed 35 deg after ignition, and combustion was approxi-

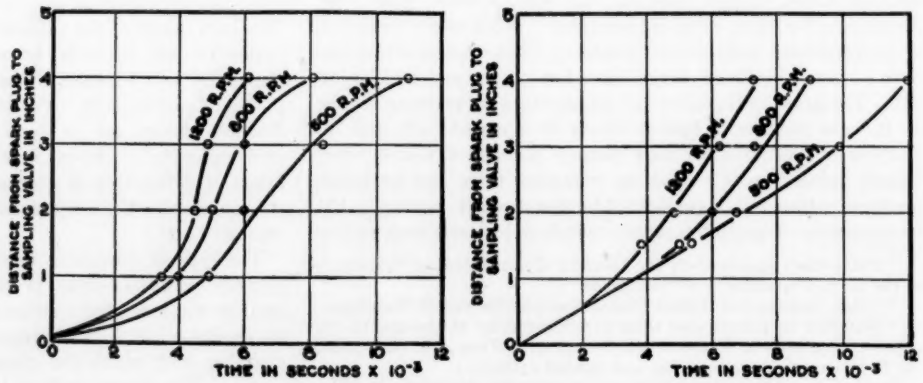


FIG. 2 EFFECT OF ENGINE SPEED ON PROGRESS OF COMBUSTION ZONE

(At left, through middle of combustion chamber; at right, along combustion chamber wall. Spark advance, 16 deg at 500 rpm, 22 deg at 800 rpm, 26 deg at 1200 rpm. Mixture ratio, 80 per cent of theoretical air. Throttle, fully open. Fuel, gasoline. Compression ratio, 5 to 1. Each experimental point, or circle, shows, for a given location of the sampling valve and a given engine speed, the time, in seconds, after ignition when the oxygen concentration began to decrease rapidly.)

mately completed at 50 deg after ignition. The location of the sampling valve with respect to the spark plug was found to have little effect on the results, but the use of a knocking fuel appeared to cause a uniform increase in the rate of burning. The conclusions drawn from these observations were subject to revision when it was found that the sampling valve opened nearly 10 deg late and that it remained open more than 20 deg instead of 2 deg. Experiments made a few years later, on a water-cooled ell-head engine equipped with a new and improved sampling valve, were reported by Withrow, Lovell, and Boyd (19). The times at which the valve opened and closed were known definitely, and samples were taken at different places, either down the middle of the combustion chamber or along one side. Values of the oxygen concentration in samples taken at two distances from the spark plug and at various points in the cycle are shown graphically in Fig. 1, which is reproduced from the original paper. The fact that the time interval, between the instant of ignition and the instant that oxygen concentration began to decrease rapidly at the sampling valve, increased as the distance between the spark plug and the valve was increased, indicates that a definite combustion zone passes through the gasoline-air mixture at a finite velocity. The steepness of the oxygen curves is affected by

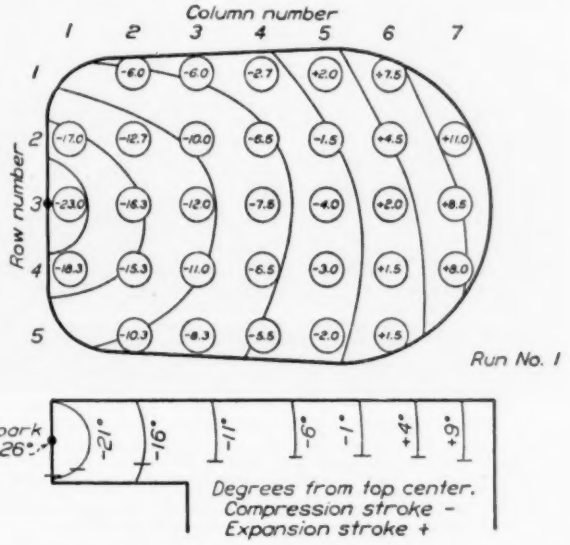


FIG. 3 TYPICAL FLAME DIAGRAM

(The circles on the plan view of the combustion chamber show the locations of the windows. The stroboscope reading within each circle indicates the position of the engine crank, in degrees from upper dead center, when the flame first becomes visible under the corresponding window. A vertical section through the middle of the combustion chamber is also drawn. In the absence of any information as to the curvature of the flame front in this view, it has been represented by circular arcs drawn about the point of ignition. Horizontal dashes show the level of the top of the piston for successive positions of the flame and indicate the extent of piston movement during a normal explosion.)

various factors, such as the duration of the valve opening and the fact that each sample averages numerous cycles, but the curves indicate that the combustion zone propagated through the mixture is comparatively narrow. These particular curves, obtained under identical conditions except that in one set tetraethyl lead was added to the gasoline to suppress knock, indicate definitely that detonation occurs in the

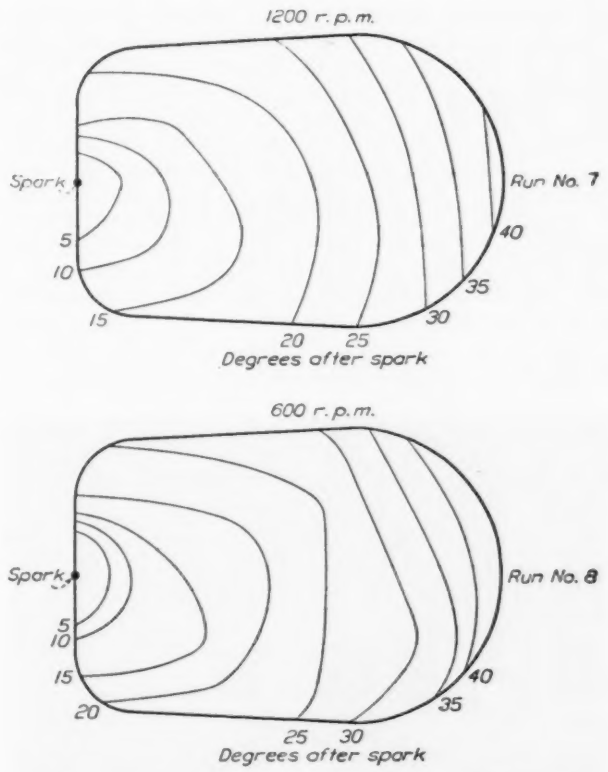


FIG. 4 EFFECT OF ENGINE SPEED ON FLAME DIAGRAMS

(The time required by the flame to cross the head is 12 milliseconds at an engine speed of 600 rpm and less than 6 at 1200 rpm. In a bomb of the same dimensions as this combustion chamber, the time required is 60 milliseconds.)

part of the charge which burns last and are not inconsistent with the view that the combustion zone travels more rapidly through the last one-fourth of the combustion space in detonating explosions than in non-detonating explosions. The curves reproduced in Fig. 2 indicate that, in normal combustion, the combustion zone travels at an average speed which varies, over the range studied, directly with engine speed. At any engine speed, combustion proceeded through the middle of the combustion chamber at an accelerated rate until it was more than one-half of the way across the chamber, and then traveled the remaining distance at a decreasing rate of speed. The combustion zone proceeding along the wall of the combustion chamber was found to travel the entire distance at an accelerating rate.

Visual or photographic observations of the appearance of flame at each of several windows in the combustion chamber of an engine have been made with the aid of a stroboscope by Ricardo

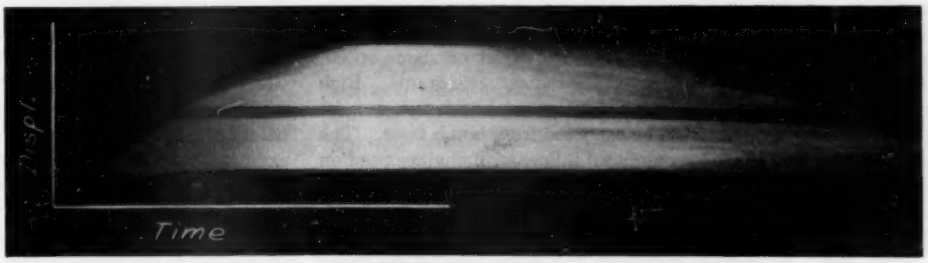


FIG. 5 PHOTOGRAPHIC RECORD OF EXPLOSION IN BOMB

(This time-displacement record of the progress of combustion down the middle of a closed bomb, similar in shape to the combustion chamber of the engine, shows the slow start of the flame, its acceleration to a maximum speed near the center of the combustion chamber, and its decreasing speed during the later portion of its travel.)

(20), Glyde (21), and Horning (22). Ricardo observed that detonation appeared only after the flame had nearly completed its travel. In work done at the Bureau of Standards for the National Advisory Committee for Aeronautics, Marvin and Best (23) provided an ell-head engine with a special water-cooled head in which were inserted 31 quartz windows symmetrically distributed over the combustion space. The work covered by the report cited was done with non-knocking gaseous fuels and with a symmetrical combustion chamber of uniform thickness, but it has been extended since to include liquid fuels, four different designs of combustion chamber, and the effect of spark position and of dual ignition. A typical flame diagram and the data used in plotting it are shown in Fig. 3. The stroboscope setting for the first appearance of flame at each window is determined and a simultaneous measurement of cylinder pressure is

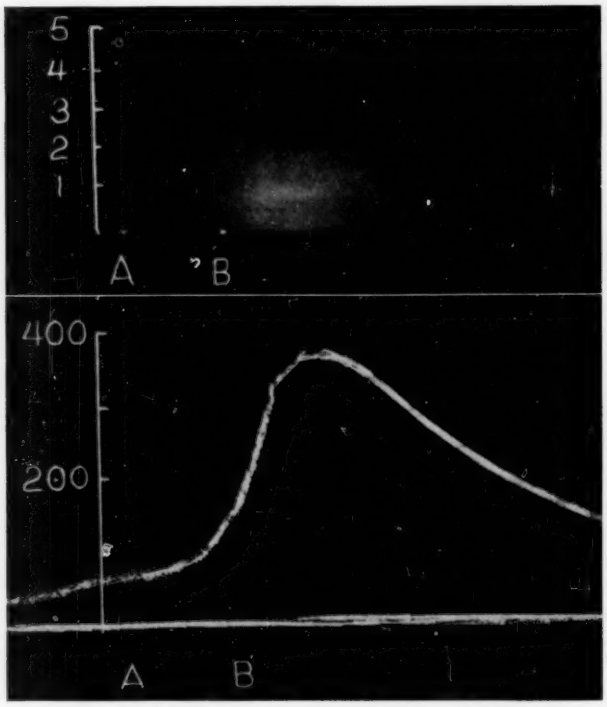


FIG. 6 RECORD OF A NORMAL EXPLOSION

(These records of flame movement and pressure development are for a single non-knocking explosion in an engine operating on benzene. The vertical scales are, respectively, distance from the spark plug in inches and pressure in pounds per square inch. The horizontal scale is degrees of crankshaft revolution, the distance *A*-*B* being 20 deg.)

made with the balanced diaphragm pressure indicator (24). Both flame and indicator diagrams thus obtained, for a given set of engine operating conditions, are reproducible, but they necessarily are averages and give no information regarding variations from cycle to cycle. Compression ratio, charge density, and spark advance were found to have little effect on flame speed, although having marked effects on the indicator diagrams. Rich or lean mixtures and increased dilution with exhaust gas decrease flame speed, as would be expected from bomb tests. The outstanding effect is the pronounced influence of engine speed on flame speed. As shown in Fig. 4, the flame crosses the head in about the same number of degrees of crank angle, or in about one-half the time, at 1200 rpm as at 600 rpm. Possibly the increase in engine speed, by greatly increasing the raggedness of the flame front, enables it to ignite a given volume of charge in a much shorter time. In a bomb of the same dimensions as the combustion chamber used, the average flame speed is less than one-tenth of that in the engine at 1200 rpm. All of the time-displacement diagrams for the engine exhibit the same characteristics as the left-hand boundary of the photographic bomb-explosion record shown in Fig. 5. The flame starts slowly, accelerates to a maximum speed near the center of the combustion chamber, and decreases in speed during the later portion of its travel.

A large number of photographs of combustion in a gasoline engine, as it appeared through a narrow quartz window that

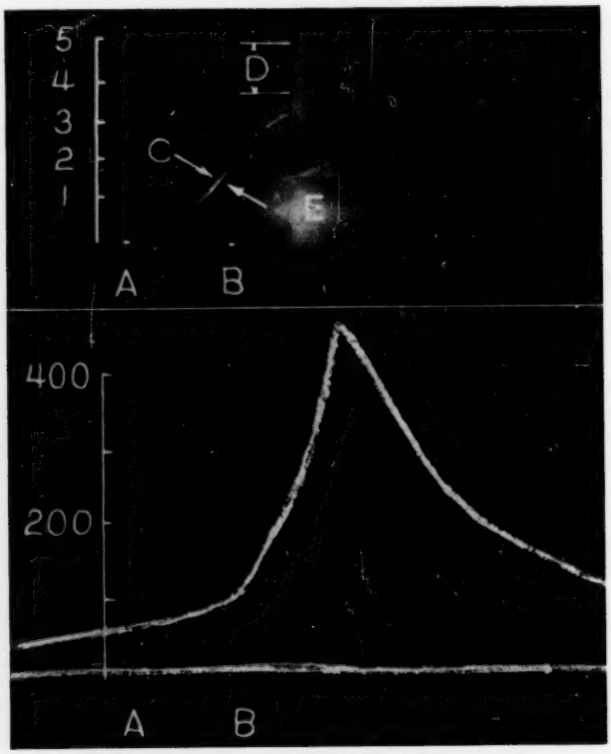


FIG. 7 RECORD OF A DETONATING EXPLOSION

(These records of flame movement and pressure development are for a single knocking explosion in an engine operating on gasoline. As in Fig. 6, the vertical scales are in inches and pounds per square inch and the horizontal scale is in crankshaft degrees. In the upper record, *A* is the ignition spark and *B* is a second spark 20 deg. later, *C* indicates the combustion zone prior to knock, *D* the detonating zone, and *E* the afterglow.)

extended over the entire length of the combustion chamber, with simultaneous records of pressure development have been obtained by Withrow and Boyd (25). These remarkably complete records of individual explosions were secured by using the electrical pressure indicator of Martin and Caris (26) in synchronism with a new combustion camera. The special water-cooled cylinder head, with its removable quartz window, carried the camera, while the pressure element was incorporated into the spark plug. The combustion chamber was a tapered wedge with the broad end over the valves and the thin narrow end at the far side of the piston. The window included a view of the spark-plug gap which was over the intake valve. Fig. 6 shows a record of a normal explosion in which a benzene-air mixture was ignited 20 deg before top dead center, and Fig. 7 shows a record of a detonating explosion with gasoline as the fuel. On the upper record in each case *A* is the spark which fired the charge and *B* is a second spark 20 deg later to establish the time scale. These two reference points are marked automatically on the pressure record also. Distances on the upper record are given in inches, and the pressure scale on the lower record is in pounds per square inch. Each flame photograph shows the projection of the combustion zone and a more or less extensive region of "afterglow," indicated,

respectively, by *C* and *E* in Fig. 7. *D* in this figure represents the detonating zone. Visual observations of explosions, by the aid of a rotating mirror, show the combustion zones more clearly because they are greenish blue, while the afterglow is a bluish white or a reddish yellow, depending upon the mixture ratio.

The records presented by Withrow and Boyd, covering the entire range from normal combustion to severe knock, indicate that a relatively narrow combustion zone starts at the spark plug and moves progressively through the charge. Oxidation of the

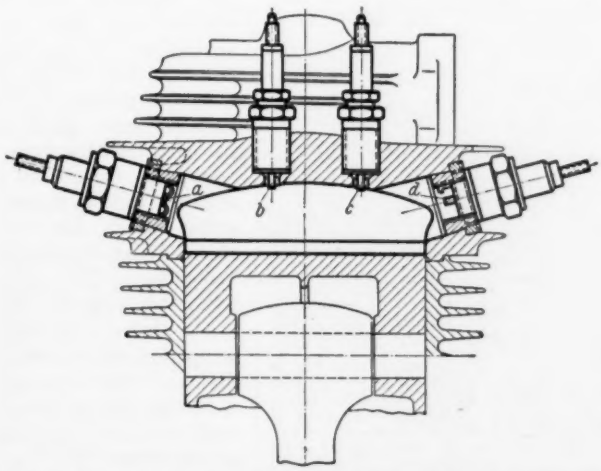


FIG. 8 COMBUSTION CHAMBER, SH-13 AIRCRAFT ENGINE CYLINDER
(In this sectional view of the Sh-13 engine cylinder, *a* is the ignition gap, and *b*, *c*, and *d* are the ionization gaps. Two standard spark plugs and two midget plugs were used.)

fuel is apparently complete within this zone, but the products of combustion to the rear of the flame front continue to emit light for some time. A knocking explosion differs from a normal explosion only in the way the last portion of the charge burns. Withrow and Boyd state this difference substantially as follows: Whereas in a non-knocking explosion the flame continues to move at a comparatively constant rate to the end of the combustion space, in a knocking explosion the latter portion of the charge inflames at a much higher rate than normal. According to their records, this rate was often so high that at the instant of knock the flame appeared simultaneously throughout that portion of the charge still remaining to be burned. The extremely high rate of inflammation in that portion of the charge which burns at the instant of knock is attributed to auto-ignition occurring within it. The authors' remaining conclusions are summarized in the next three sentences. The violence of the knock is determined by the proportion of the total charge involved in the spontaneous inflammation, or the amount remaining to be burned at the instant knock occurs. The very rapid inflammation of this last portion of charge is accompanied by a very rapid rise in pressure, and the magnitude of this pressure rise increases along with the intensity of the knock. The sole effect upon combustion of the presence of tetraethyl lead in the gasoline is to prevent the extremely rapid inflammation of the latter portion of the charge and the accompanying abrupt pressure rise.

An electrical method of measuring the rate of flame propagation in engine cylinders was developed at the Bureau of Standards in 1919, and preliminary data were published by MacKenzie and Honaman (27). A Liberty aircraft engine cylinder was provided with three spark plugs, one of which was used to fire the charge, while the arrival of the flame at each of the observing plugs was detected by ionization of the gap and recorded on an oscillograph. Turbulence effects with velocities ranging from 1 to 4 meters per second were observed, and it was found that the time intervals recorded were abnormally short when the engine was knocking violently. The somewhat erratic results obtained were due partly to inadequate control of engine-operating conditions. Recently further work on flame propagation by the ionization method has been conducted at the German Aeronautical Research Establishment (D.V.L.) by Schnauffer (28, 29). He used a Siemens and Halske Sh-13 aircraft engine cylinder, equipped with one firing and three measuring plugs, as shown in Fig. 8. The influence of mixture ratio, engine speed, turbulence, compression ratio, and fuel composition on rate of flame propagation were studied, and the results accord with those obtained by Marvin and Best. Measurements of flame propagation and pressure development under normal burning and knocking conditions obtained by Schnauffer confirm the photographic records of Withrow and Boyd. One oscillograph record, indicating that ionization of gaps *C* and *D* was nearly coincident with the development of maximum cylinder pressure, is reproduced in Fig. 9.

The spectroscopic study of visible radiation from a quartz window in the combustion chamber of an engine was begun at the Massachusetts Institute of Technology in 1926 by Clark and Thee (30) and was continued by these investigators and their co-

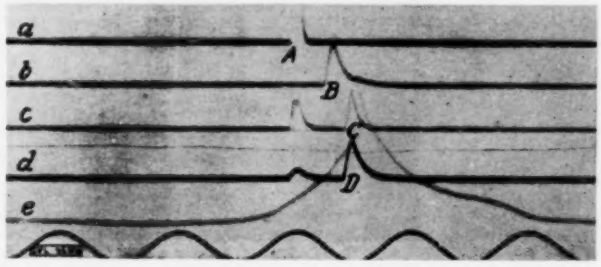


FIG. 9 OSCILLOGRAPH RECORD OF A DETONATING EXPLOSION
(*a*, ignition current; *b*, *c*, and *d*, ionization currents; *e*, pressure line; *A*, ignition instant, shown by induction on *c* and *d* also; *B*, *C*, and *D* show arrival of flame at corresponding gaps. The pressure-line peak is directly over the letter *C*. The time scale is given by the sine curve at the bottom of the record, the distance between peaks being 20 milliseconds.)

workers for several years. However, they were not in a position to distinguish between radiation from the flame and radiation from the products of combustion. Recently Withrow and Rassweiler (31), using with a stroboscope disk the special cylinder head developed for photographing the progress of combustion, have made separate spectroscopic studies of the flame fronts and afterglows of explosions produced with several fuels in a gasoline engine.[4] Referring to Fig. 10, spectrum 12, which represents the flame front of gasoline burning in the engine under non-knocking conditions (spark advance 10 deg) shows the CH and Swan C_2 bands characteristic of the inner cone when gasoline is burned in a blast burner. Spectrum 11, obtained from the flame front of gasoline burning under knocking conditions (spark advance 25 deg), shows the CH and C_2 bands very faintly and a heavy continuous spectrum at the red end. Spectrum 13, obtained under the same test conditions as spectrum 11 except that the knock was suppressed by adding Ethyl fluid to the gasoline, is practically identical with spectrum 12 if the lead lines and the lead-oxide bands are neglected. Spectrum 17 is the flame spectrum of lead, given for comparison with spectra 13–16. Spectra 25–28 in Fig. 11 represent the afterglows (*E* in Fig. 7) in the combustion of gasoline and benzene. It will be seen that these spectra re-

[4] In an article on the "Emission Spectra of Engine Flames," which appeared after this paper was written, Rassweiler and Withrow report the extension of their spectrographic studies into the ultraviolet. They find bands characteristic of free OH radicals present in both flame-front and afterglow spectra. See *Ind. & Eng. Chem.*, 24, 528 (1932).

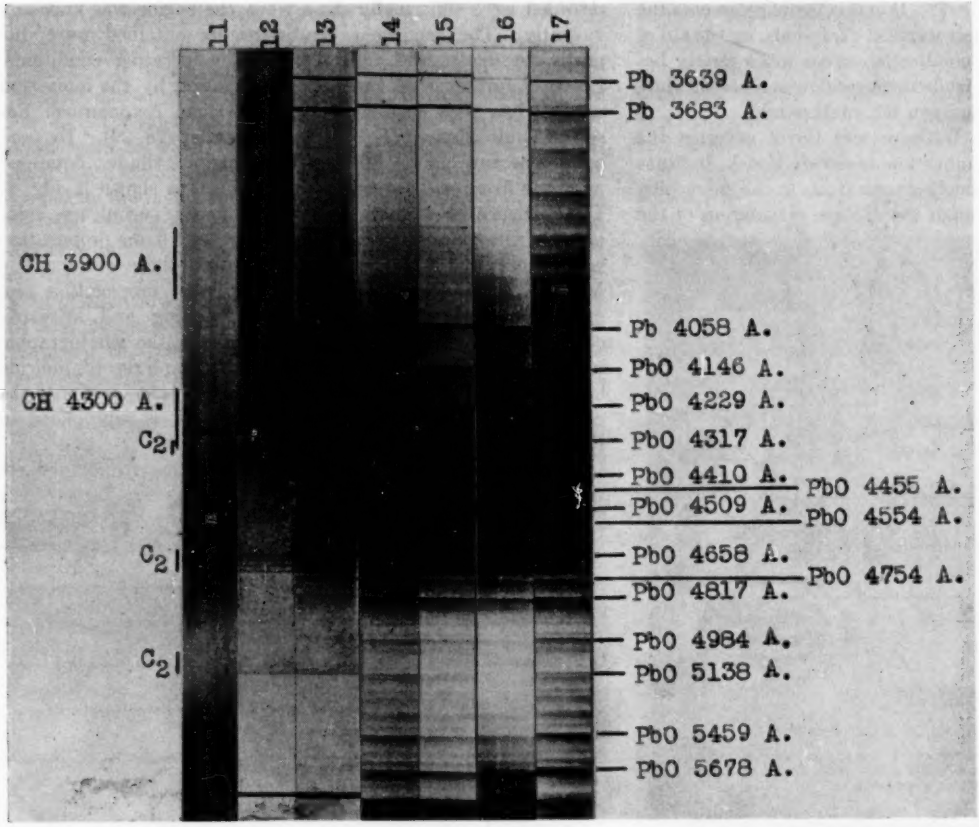


FIG. 10 ENGINE FLAME-FRONT SPECTRA

(Spectra of light emitted from detonating zone, mainly under non-knocking conditions—enlarged 3 to 1. Engine speed, 1000 rpm.)

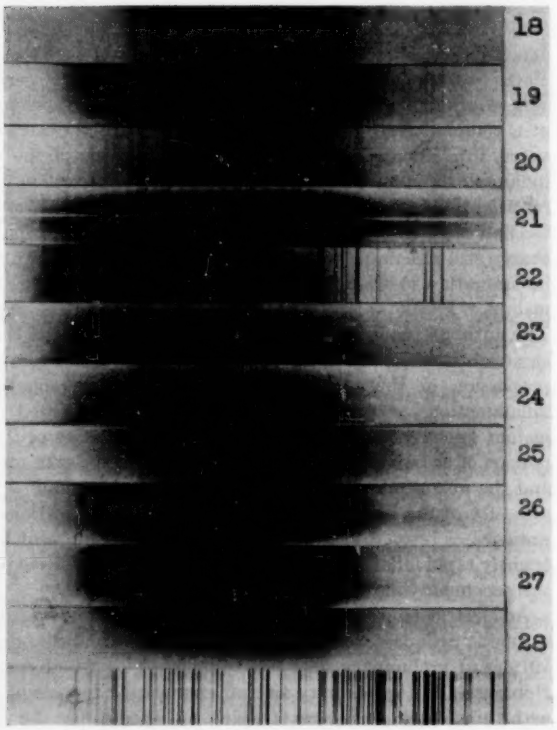
Spectrum	Fuel	Knock	Spark advance	Stroboscope
11.....	Gasoline	Yes	25°	Used
12.....	Gasoline	No	10°	Used
13.....	Gasoline and Ethyl fluid	No	25°	Used
14.....	Gasoline and Ethyl fluid	No	20°	Not used
15.....	Gasoline and tetraethyl lead	No	20°	Not used
16.....	Methanol and Ethyl fluid	No	10°	Not used

NOTE: For comparison, spectrum 17 gives the flame spectrum of lead, obtained with excess oxygen in the flame.

semble the spectrum of the carbon-monoxide flame front much more closely than that of the gasoline flame front. Hence it appears likely that spectra 25–28 are due to CO, CO_2, or O_2. The authors state that "the afterglow spectrum is emitted by the same molecules as those which give off light during the reaction between carbon monoxide and oxygen. At present it appears probable that carbon dioxide molecules are the emitters." The luminescence of the products of combustion in a gas engine on successive compressions noted by David and Davies (32) may be significant in connection with the increase in intensity of the afterglow as cylinder pressure approaches a maximum.

A study of infra-red radiation from the engine cylinder during combustion was undertaken at the Bureau of Standards last year by Steele (33). The apparatus used by Marvin and Best was modified so that the total radiation transmitted by a fluorite window in the cylinder head could be passed through the stroboscope and focused on a sensitive vacuum thermocouple which was connected with a suitable galvanometer. The variation in total radiation (up to 11μ) through the explosion period was determined first, and then the energy in certain regions was investigated with the aid of suitable filters.

Preignition, defined as ignition from any source prior to the passage of the ignition spark, has been discussed very fully by



Spectrum 18 Gasoline flame front in detonation zone, engine knocking
Spectrum 19 Carbon-monoxide flame front in detonation zone, engine not knocking
Spectrum 20 Carbon-monoxide–air flame in burner
Spectrum 21 Carbon-monoxide–oxygen flame in burner
Spectrum 22 Carbon-monoxide afterglow in engine; taken about 1 in. from spark plug, engine not knocking
Spectrum 23 City-gas afterglow in engine; taken near spark plug, engine not knocking
Spectrum 24 Ethylene afterglow in engine; taken near spark plug, engine not knocking
Spectrum 25 Gasoline afterglow in engine; taken near spark plug, engine knocking
Spectrum 26 Gasoline afterglow in engine; taken near spark plug, engine knocking
Spectrum 27 Benzene afterglow in engine; taken near spark plug, engine not knocking
Spectrum 28 Benzene afterglow in engine; taken near spark plug, engine not knocking

NOTE: An iron arc spectrum is given at the bottom for comparison. In the case of all engine spectra, except 26 and 27, excess oxygen was added to the intake air.

FIG. 11 COMPARISON OF AFTERGLOW AND FLAME-FRONT SPECTRA

(Flame spectra showing similarity between afterglows and carbon-monoxide oxygen flame—enlarged 3 to 1. Engine speed, 1000 rpm. See explanatory note at bottom of the left-hand column.)

Sparrow (34). With some fuels, especially hydrogen, preignition may be due to spontaneous ignition, but it is usually caused by the presence of hot spots in the cylinder, such as an improperly cooled exhaust valve or an overheated spark-plug electrode. Slight preignition may cause only a small reduction in engine power such as would result from an excessively advanced spark. However, mild preignition is likely to grow worse rapidly with progressive loss in power. Fig. 12 shows an indicator card taken in the altitude laboratory at the Bureau of Standards with an

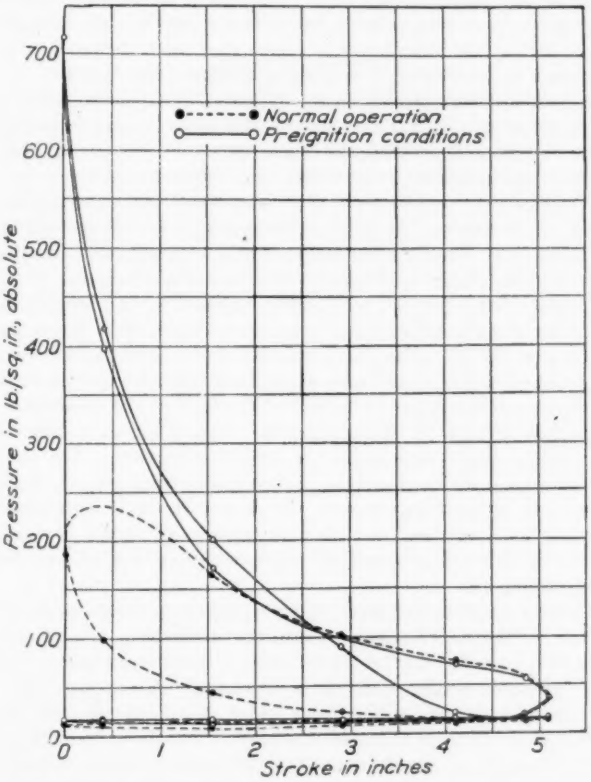


FIG. 12 EFFECT OF PREIGNITION ON INDICATOR CARD

[Two indicator cards are shown, one under preignition conditions (full line) and one under normal conditions (broken line). Tests were made on individual cylinders of a multi-cylinder engine in the altitude laboratory at the Bureau of Standards. Engine speed, 1600 rpm. Compression ratio, 8.3 to 1. Barometer, 57.8 cm. Hg (corresponding to an altitude of 7000 ft). The power developed in the preigniting cylinder was negative.]

engine preigniting. The power developed in the preigniting cylinder was negative, and a portion of the power developed by the other cylinders was employed in overcoming the resistance which it offered. Severe detonation may result in preignition, but these are separate and distinct combustion phenomena. Preignition occurs prior to the passage of the spark, and detonation seldom earlier than 30 deg after the spark.

In regard to detonation in aircraft engines, the report made to the Society of Automotive Engineers five years ago by a Research Sub-Committee on the Causes and Effects of Detonation (given in the appendix to this paper) still remains substantially true today. The present review shows that our knowledge regarding combustion processes in engines has been made more definite and that promising new lines of attack on the problem have been developed, but it still appears that we do not know exactly what detonation is nor precisely how anti-knock dopes act. It has not even been established beyond controversy that the Berthelot detonation wave ever is set up in the engine cylinder.

The courtesy of T. A. Boyd, of General Motors Corp., in supplying the photos for Figs. 6, 7, 10, and 11 is gratefully acknowledged.

Appendix

RESEARCH SUB-COMMITTEE REPORT ON THE CAUSES AND EFFECTS OF DETONATION

Fundamentally, detonation is a type of combustion which is abnormally rapid as compared with what may be called normal or non-detonating combustion. The behavior seems somewhat analogous to the difference between the slow burning and detonation of many common explosives. Just what mechanism is responsible for the speeding up of the reaction or some part thereof has not, in our opinion, been at all definitely established, although many ingenious theories have been set forth. It is known, however, that detonation is increased by increasing the temperature or the pressure of the charge during the time of the explosion, which may be brought about in many different ways. The presence of localized hot spots or an increase in the length of flame travel also tend to increase detonation. Detonation varies greatly with mixture ratio, generally passing through the maximum between the point of maximum power and that of maximum economy.

Probably the greatest single factor which influences detonation is the chemical composition of the fuel, but even here there are apparently few generalizations which can safely be made. Apparently those paraffin hydrocarbons which occur in many straight-run gasolines tend to knock badly, while certain other paraffin hydrocarbons are good antiknock fuels. As a class, the naphthene, aromatic, and olefin hydrocarbons which occur in many straight-run or cracked gasolines tend to knock distinctly less than the normal paraffin hydrocarbons.

A great many elements and compounds have the quality of affecting detonation profoundly. While no hard and fast statements can be made, it appears in general that the anti-knock property is fundamentally a property of the individual element, although it is modified to a certain extent by the form of the compound in which the element occurs. Thus many, though not all, compounds of nitrogen have a moderate anti-knock effect, and practically all volatile compounds of iodine, selenium, tellurium, nickel, cobalt, tin, and lead have marked anti-knock effects, some of them being effective even when present in extremely small quantities.

The effects of manifestations of detonation are, first, a disagreeable noise, varying from a slight "ping" to a very loud clatter, a loss of power if knocking is at all severe, and a tendency to overheat the engine and the cooling water. If marked detonation continues, this overheating frequently results in preignition, which is, however, an entirely separate phenomenon. In extreme cases, mechanical effects such as breaking of spark-plug porcelains, the actual rupture of piston heads, and damage to bearings or other structural parts of the engine may result.

Probably the most important effect of detonation is an indirect one, in that it limits the compression pressure which it is practicable to embody in an engine to give satisfactory performance with ordinary commercial fuels, and this, in turn, seriously limits the fuel economy which could be obtained if this limitation could be removed.

One of the most serious problems in connection with detonation is that of accurately comparing the detonation characteristics of different fuels. This is complicated by the fact that so many factors in engine design and operation have their several effects on detonation; but many laboratories are now working on this problem, and while we do not believe that any entirely satisfactory standard method has yet been developed, this should be worked out to a satisfactory point within the next year.

Future work directed toward the solution of this problem should largely proceed along four lines:

(1) The development of a standard and reproducible method of testing and rating anti-knock fuels
(2) Modification of engine design to make it possible to get higher compression pressures with a given fuel without detonation
(3) Modification of the fuel in the direction of making it stand higher compression pressures in a given engine without detonation
(4) Fundamental research to determine the mechanism of detonation and its prevention.

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Discussion

EDWARD ADAMS RICHARDSON.[5] In a paper presented before the National Oil and Gas Power Meeting of June 8, 1932 ("Heating Fuels for Injection Engines"), the writer discussed basic theories of combustion. Reference should be made to this paper by those interested in combustion processes.

In discussing detonation waves, it is most important to note their high velocities, from over 3000 fps to as high as 9000 fps. The writer believes that Withrow, Boyd, and others have been misled into believing that spontaneous self-ignition occurs ahead of the combustion wave when detonation sets in, owing to failure to realize how high these velocities may be. In the Withrow-Boyd photographs of flame progress, for instance, the apparently vertical combustion line at detonation would make an angle of 0.001 to 0.003 radian with the vertical if we deal with a wave phenomena. It would require careful measurement to detect such a small angle. Furthermore, these same photographs uniformly showed a fillet between the normal combustion region and the "vertical" line. This fillet points very plainly toward a very high rate of flame-front acceleration, not to self-ignition of the mass. Furthermore, a study of the flame-front reaction as a chemical process affected by temperature, density, and wave motions in the gases shows that high rates of flame acceleration are possible. This being true, it must be realized that a flame front is a form of discontinuity in a gaseous mixture which tends to increase. It is highly improbable that even a homogeneous mixture at nearly uniform temperature as high as the ignition temperature could ignite uniformly and instantaneously throughout the mass. Any slight temperature difference tends to increase due to chemical action rates which differ, and such a tendency leads naturally to the development of a flame front. The only stable phenomena in a homogeneous or heterogeneous mixture is the tendency to develop a flame front based on the element of volume most favorably situated. Several flame fronts may start from several elements, but the element most favorably situated grows faster than the others until it dominates the scene. Other reasons will be found in the paper referred to.

The increase in the rate of combustion with engine speed deserves comment. It must be obvious that turbulence will be substantially directly proportional to the engine speed. Hence, for a given engine configuration, the space displacement of an element of the gas will be roughly constant for a given crank position irrespective of speed. As normal combustion flames are very slow affairs, while the circulation of the contents of a combustion chamber is much more rapid, it should not be particularly surprising if the flame is distributed throughout the combustion chamber largely by the circulation. Fig. 4 shows a lack of symmetry easily explainable by normal circulation. We need more data on air movement in engines during combustion. Such data are of basic importance. If a slow flame is distributed largely by circulation, and the circulation is a function of crank position rather than speed, the observed effect is accounted for approximately.

With respect to the effect of dopes, reference should again be made to the paper noted. There the mechanism of chemical reaction from the kinetic theory point of view is discussed, and the probable action of dopes is developed. A vaporized substance of high self-ignition point which can dissociate with the absorption of energy at a proper temperature is shown to have the ability, even in small quantities, of almost instantaneously absorbing energy which otherwise would be utilized in the work of accelerating the flame velocity to that corresponding to detonation. A very slight delay in energy liberation is sufficient to permit the gas to burn normally, whereas otherwise it would detonate. This suggestion deserves further consideration, for it is based squarely on the most basic principles of physics and chemistry and is in line with the mathematical requirements of the kinetic theory. The effect is similar to that which occurs at the critical point on the heating of steel. At this point the temperature rise is halted while energy is absorbed for molecular changes.

With regard to the suggestion that we do not know whether Berthelot detonation waves are ever set up, the writer is convinced that the Boyd pictures require it. Also, the pressure effects noted may be higher than any theoretical mean pressure obtainable. This is only possible when a flame front is involved having high velocity which is brought to a halt at the walls, and calculation shows that the flame velocity must be of the order of the detonation-wave velocity. The kinetic theory practically demands such an effect. It requires work and time to accelerate a flame front.

The writer showed, in the paper referred to, that any fuel could be burned without detonation provided it were introduced into the cylinder by injection while at a temperature above the self-ignition temperature corresponding to the conditions existing in the combustion chamber. Hence he recommended such heating,

[5] Bethlehem, Pa. Jun. A.S.M.E.

particularly for injection engines, as an alternative preferable to fuel choice, modification, or the use of dopes.

In the same paper it is shown that this degree of heating and mode of injection would eliminate dependence on spark plugs or the temperature of the compressed air for ignition and so would secure that wide range of choice of compression ratios considered desirable by the designer and operator of engines.

Obviously, the writer considers the Research Sub-Committee Report on Causes and Effects of Detonation very much out of date in many respects. A reexamination of available data appears to be in order, and a new report is desired.

Author's Closure

The lack of symmetry in Fig. 4 which Mr. Richardson mentions merely indicates a normal circulation or swirl of the charge conditioned by the geometry of the combustion chamber, but the primary effect of turbulence on flame speed may be a local mixing which materially increases the effective depth of the reaction zone at the higher engine speed.

Mr. Richardson is quite justified in pointing out that such records as Fig. 7 and Fig. 9 do not require the assumption of spontaneous inflammation of the charge remaining at the instant of knock. If a detonation wave were set up, this would give similar records, but we cannot agree that the records which were obtained by Withrow and Boyd necessarily involve the Berthelot wave.

In addition to the references contained in the paper, it may be of interest to note that the National Advisory Committee for Aeronautics has recently issued as Technical Memorandum No. 694 a translation of a further report by Duchene under the title, Combustion of Gaseous Mixtures. American Chemical Society Monograph No. 61 ("The Catalytic Oxidation of Organic Compounds in the Vapor Phase," by Marek and Hahn) contains a chapter on The Cause and Suppression of Knocking in Internal-Combustion Engines which considers several proposed mechanisms of hydro-carbon oxidation and discusses various theories which have been advanced to account for detonation and for the action of knock inducers and knock suppressors.

AER-55-10

Developments in Aluminum Alloys in Relation to Economies in Aircraft Construction

By C. F. NAGEL, Jr.,[1] and G. O. HOGLUND,[2] PITTSBURGH, PA.

This paper reviews the mechanical properties and corrosion resistance of available aluminum-alloy structural materials, and as to the fabricating situation, discusses the status of technical information, equipment, and practices (forming, jointing, and corrosion prevention). A summary coordinates these matters to show what has been accomplished and the general direction of present developments, with the thought that the present lull in production provides an opportunity to plan for the future and conduct the indicated experimental work. The discussion is limited to the structural parts of aircraft, such as fuselage and wing and control surfaces, omitting power plant, landing gear, and accessories.

PRIOR to 1923 there was available only one strong aluminum alloy, commonly known as duralumin (17ST). The quality of this material has gradually improved. The present specifications give a minimum yield point of 30,000 lb per sq in. for the common gages of sheet, but the 1920 specifications guaranteed only 25,000 lb. The tensile strength of all 17ST sheet produced during 1923 averaged 56,600 lb per sq in., but for 1931 the figure was 60,000 lb. By improvement in internal soundness and elimination of surface imperfections, workability has been bettered, and hence uniformity in fabrication.

Then came Alclad 17ST sheet, introduced in 1927. This provided a material of greatly enhanced corrosion resistance, at a sacrifice of about 10 per cent in strength. In 1928 17SRT sheet was made available. This possesses a guaranteed minimum yield point 40 per cent greater than that of 17ST. The structure of the U.S.S. *Akron* was fabricated from this material.

Now a new alloy, 24S, in development for the past two years, will be placed on the market. This is also of the duralumin type, but has much higher strengths than 17ST. It will be available in four grades, 24ST, 24SRT, Alclad 24ST, and Alclad 24SRT. There is also now available a new aluminum-alloy tube (17SRT) having a guaranteed minimum yield point of 52,000 lb per sq in. Table 1 gives the typical or average mechanical properties of 24S in comparison with 17S.

Corrosion Resistance

So gradually has the change come about, it probably is not realized that the corrosion resistance of 17ST has been improved greatly during the past few years, particularly that type described as "intercrystalline embrittlement." Only rarely do we now find, when the product heat treated by the manufacturer[3] is subjected to severe corroding influences, that the resulting attack is of this nature. The principal reason for this lies in improved heat-treating equipment and technique and a more complete knowledge of the subject. The same trend has been true of the product heat treated by the various fabricators,[3] though naturally this has trailed behind to some extent.

With the production of Alclad 17ST (1927) an improvement of an entirely new magnitude was effected. Fig. 1 shows the appearance of 17ST and Alclad 17ST sheet samples that have been constantly in salt spray for $3^1/_2$ years for one group and 5 years for another set. The superiority of the Alclad 17ST is evident.

Fig. 1 shows four samples. In all four cases the test samples were exposed in the bare state, without protecting coating of any kind. Beginning at the left side these are described as follows:

1 Alclad 17ST channel made from 20-gage sheet. This sample was exposed to 20 per cent salt spray for 5 years. Metal is stained, but still entirely sound.

2 This channel was made from 20-gage 17ST sheet and shows the effect of corrosion after the same exposure as sample 1.

3 This is a riveted joint made from two pieces of Alclad 17ST sheet and a 17ST rivet $^3/_8$ in. in diameter. This sample has been exposed to 20 per cent salt spray for about $3^1/_2$ years. The absence of corrosion attack on the surface and the cut edges of the sheet, as well as on the 17ST rivet head, should be noted.

4 This joint is similar to that of sample 3, except that it was made from $^1/_{\text{c}}$-in. 17ST sheet. The exposure was the same as that for sample 3. The severe corrosion attack on the surface and edges of the sheet, as well as on the rivet head, should be noted.

Just as 17ST equals 17SRT and Alclad 17ST equals Alclad 17SRT in corrosion resistance, so 24ST equals 24SRT and Alclad 24ST equals Alclad 24SRT. It is necessary to indicate the relationship of the 24S group (bare and Alclad) to the 17S group. To be technically correct, it should be said that each grade in the 24S group is a trifle inferior in corrosion resistance to its counterpart grade in the 17S group. However, test data indicate that

TABLE 1 AVERAGE MECHANICAL PROPERTIES OF 0.064-IN. SHEET

	Yield point, lb per sq in.	Ult. ten. str., lb per sq in.	Elongation, % in 2 in.
17ST	36,000	60,000	20
Alclad 17ST	33,000	54,000	18
17SRT	45,000	61,000	14
24ST	43,000	65,000	20
24SRT	53,000	68,000	14
Alclad 24ST	39,000	60,000	17
Alclad 24SRT	48,000	62,000	13

[3] By "manufacturer" is meant the materials (sheet, tube, etc.) producer, while "fabricator" refers to the aircraft plant.

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[1] Technical Director, United States Aluminum Company. C. F. Nagel, Jr., was graduated from Cornell University in 1915 with the degrees of A.B. and B.Chem. He has been with the Aluminum Company of America since that date, except for two years of service with the United States military forces. His activities in the Aluminum company embraced development, manufacture, and exploitation of aluminum and aluminum-alloy products, with especial attention to use of these products in the aircraft industry.

[2] Associate Engineer, Aircraft Development Aluminum Company. G. O. Hoglund, after graduating from the University of Michigan in 1925, took a position in the Bureau of Aeronautics, Navy Department, working on the design and development of propellers for naval aircraft. In 1928, he was transferred to the engineering department at the Naval Air Station, Lakehurst. This led to an appointment as assistant professor of aeronautical engineering at the University of Minnesota in 1928–1929. Since that time he has been with the Aluminum Company of America, working first on the application of the aluminum alloys in the aircraft field and since January, 1932, in charge of the welding development division.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

this difference is so small as not to warrant any difference in fabricating procedure, the difference being especially small when comparing the Alclad grades. With 24S or Alclad 24S the heat-treating temperature is somewhat lower, namely 920 F, than in the 17S group.

Forms, Dimensions, and Uniformity

Prior to 1923 duralumin (17ST) was available in all forms such as sheet, tube, extruded shapes, and rivets, but the available

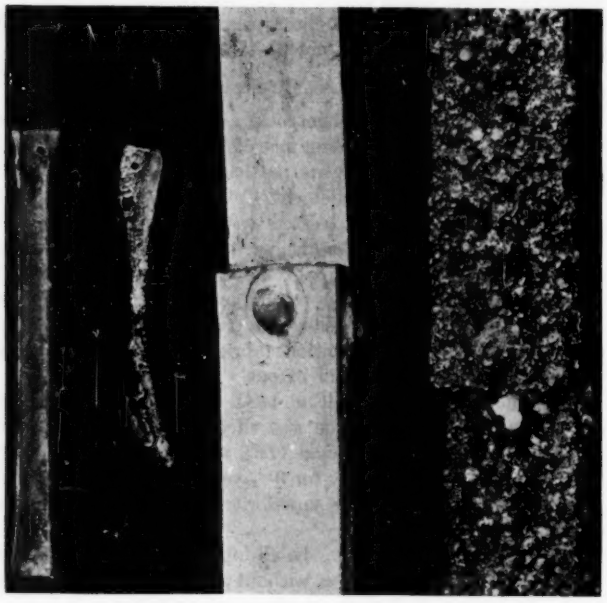


Fig. 1 Sample Sheets in Corrosion Test

sizes were limited. These dimensions have since been extended, and sheet is now available much wider and longer and in lighter gages, permitting reduction in joints and weight and improvement in appearance, especially as improved flatness has been accomplished. There is a greatly increased range of streamline tubing and other shapes, and data regarding their characteristics have been developed. In extruded shapes much thinner sections are now possible, permitting of more efficient metal distribution.

Technical Knowledge in Fabricating

Prior to 1923 the aircraft industry naturally was unfamiliar with the aluminum alloys, and in fact the available knowledge was quite limited. With few exceptions aircraft plants possessed little or no metal-fabricating equipment. During the following period an enormous amount of experimental work and practical experience has been accumulated and disseminated. The various government bureaus, the many units in the aircraft industry, the raw-product manufacturers, and scientists all over the world have concentrated upon solving the problems involved.

The past ten years has seen most aircraft plants equipped with aluminum-fabricating equipment. A new class of experienced workmen has been created, versed in the technique of manipulating the light alloys. Where ten years ago the mere task of handling aluminum alloys was in itself a formidable problem, these changes, wrought by increased facilities and experience, have been so complete that today it is generally the easiest metal to employ.

Forming

In the earlier stages, the aircraft industry was largely forced to base design on those available materials which needed no further fabrication in their plants except assembly. Hence, sheet forming was carried out only to a minor extent, most of such materials being purchased in the preformed condition.

Gradually, as newer designs called for increased amounts of the light alloys, and as production warranted, the fabricator installed his own forming equipment. Next, based on the new facilities accumulated meanwhile, designs were altered to permit more efficient use of metal, the parts being formed in the aircraft plant.

While details differ from plant to plant, the sequence of fabricating operations on sheet now generally consists of heat treating, flattening, shearing, and shaping.

As duralumin is more workable promptly after quenching, the four operations are so coordinated that a piece of sheet will go through them in not over one hour. Certain intricate parts may require shaping while in the annealed state and heat treating thereafter, and also some fabricators make various details from material as received from the manufacturers in the heat-treated state.

It also has been the general practice to heat treat the rivets just prior to use; though where strengths and freedom from danger of corrosion permit, a modified or dilute form of duralumin rivet is employed which may be driven in the heat-treated and fully aged condition.

The practice of eliminating heat treatment in the fabricator's plant and forming heat-treated sheet as received from the manufacturer has been gaining adherents, and rightly so. The reason this has not come into more widespread use is a combination of lack of appreciation of the degree of forming that can be so accomplished and the occasional necessity of allowing for a little larger radius in the design, with its attendant tool change. An illustration of what may be accomplished is the forming in the girders used in the construction of the U.S.S. *Akron*. The material is 17SRT sheet, which possesses a workability not quite so high as that of 17ST. Fabrication consisted simply of shearing the sheet to required width and length, and then putting it through the forming operations. This material was not heat treated in the fabricator's plant. Compared to the previously mentioned procedure, this reduces fabricating costs and insures maximum corrosion resistance.

Jointing

The standard assembly procedure to date has been by means of rivets and bolts. Considerable progress, however, has been made in the direction of improving heat-treating technique, so that corrosion resistance has been improved, and in developing more efficient riveting technique and equipment, which have lowered costs. A unique advance along this line is the patented automatic riveting machine developed by the Aircraft Development Corporation and employed in the assembly of the 0.0095-in.-thick Alclad 17ST skin of the Metalclad *ZMC-2*. This apparatus took three strands of 0.035-in.-diameter A-17ST wire and sheared off the rivet lengths, punched the necessary holes through two thicknesses of sheet by means of these tiny wire lengths, inserted the wires, and then headed them, all in continuous operation. The spacing also was automatic. This equipment was capable of driving 140 rivets per minute, or about 40,000 per 8-hour shift, and required two operators. Jointing is one of the expensive items of fabrication, and much promise can be held out for reduction in costs by improving upon design and equipment so as to facilitate this process.

Recently, electric spot welding has definitely come into the picture. Considerable experimental work has already been done. At least two planes have been built—one by Nicholas-Beazley Airplane and Motor Company several years ago, and this past

year a plane by the Fokker Aircraft Corporation. Both these planes substituted spot welding to a great extent for riveting. Several aircraft companies, as well as other agencies such as the Aluminum Company of America, are now actively experimenting in developing design and technique. The art has been advanced to a status warranting practical application in the industry. The logical sequence of application would be, first, to secondary joints and then, later, as experience may justify, to more important items. Application of spot welding should greatly reduce fabricating costs.

Merely to indicate the efficiency obtainable from a spot-welded joint, figures are given in Table 2 comparing the tensile strength of spot welds and rivets in Alclad 17ST sheet.

TABLE 2 TENSILE STRENGTH OF SPOT WELDS IN ALCLAD 17ST[a]

Gage	Spots or rivets per in.	Tensile strength, lb[b]	Rivet strength, lb[c]
0.064	1	675	368[d]
0.064	2	1335	736
0.064	3	1940	1079
0.051	1	490	368
0.051	2	914	736
0.051	3	1360	1077
0.032	1	320	300
0.032	2	690	600
0.032	3	880	900
0.020	1	177	140[e]
0.020	2	356	280
0.020	3	528	420
0.016	1	147	112
0.016	2	258	224
0.016	3	406	336

[a] These values are not to be used in design. They only represent actual values obtained and have been included to show that a spot-welded joint in tension possesses about the same strength as a riveted one. [b] Average of four specimens. [c] Calculated. [d] These nine tests are with 1/8-in.-diameter rivets. [e] These six tests are with 3/32-in. rivets.

One may naturally question the effect of spot welding upon corrosion resistance. Spot-welded 16-gage Alclad 17ST sheet placed in a 20 per cent salt spray continuously for one year showed a loss in tensile strength of only 0.86 per cent. While the pure aluminum layer on Alclad 17ST is somewhat pierced during spot welding, the potential difference between the exposed 17S core and pure aluminum protects the spot. This phenomenon naturally indicates the desirability of an Alclad material when spot welding.

Corrosion Prevention

Prior to 1923 this subject was hardly recognized as a problem. In fact, during the early years the principal task confronting fabricators was to use metal at all. While the materials manufacturer was inviting attention to the need of protection, actual troubles had to be encountered before this subject received much consideration; there were enough problems on hand without thinking of protection against corrosion. It is not surprising, therefore, that some corrosion troubles were encountered. However, in keeping with the progressive attitude of the industry, the subject was actively attacked, and in due course of time methods were developed that have proved successful. The peak of corrosion troubles has been definitely passed. This is especially evident upon visiting the larger repair stations and comparing the volume of reconditioning work necessary today with that of previous years.

Seaplanes operating in the tropics represent the most severe type of service to which aircraft are subjected. Constructed largely from normal duralumin, the standard practice today for this service is to anodically treat and shop-prime all details prior to assembly, followed by a second coat of paint after assembly.

To this must be added suitable heat-treating equipment and supervision to insure proper operations in order to obtain maximum corrosion resistance. This system is giving the desired protection. It is recognized, however, that this degree of protection is not necessary for all types of service nor for all parts of planes.

The next developments will be in the direction of reducing the expense of protection. It is not necessary to build landplanes with that same degree of protection required by seaplanes operating in the tropics. Even in seaplanes the superstructure is not subjected to as severe conditions as the boat hulls or pontoons. Experimentation will gradually indicate safe but less costly procedure. Undoubtedly bare Alclad, without any paint except that desired for decorative reasons, will prove entirely satisfactory for most landplanes, and probably also for superstructures in seaplanes. Two coats of aluminum paint applied to both surfaces of a sheet 0.064 in. in thickness will increase the weight by about 4.3 per cent. Also the operation of treating details prior to assembly is costly. However, bearing in mind former materials available and maintenance costs when adequate protection was not provided, these practices have been justified. With the Alclad materials, however, as for example Alclad 24ST and Alclad 24SRT, it will be possible to build lighter and less costly structures.

This should not be interpreted as suggesting any broad-scale let-down in corrosion-prevention practices. The present practices for boat hulls and pontoons should be retained in their entirety. There is warrant, however, to investigate unprotected Alclad materials in landplane structures and superstructures in seaplanes. It is not advisable to use normal duralumin in an unprotected state.

Summary

The past decade has been concerned with introducing aluminum into aircraft. The coming decade will be concerned with applying it more efficiently and at lower costs. It is believed that the direction these efforts will take will be the employment of higher-strength materials possessing greater corrosion resistance with refinement of fabricating technique to reduce cost. The immediate indications are that this will be accomplished by the wider use of Alclad in the form of Alclad 24SRT and Alclad 24ST. Heat treatment in the fabricator's plant will be largely eliminated. Proper use of bare Alclad will greatly reduce expense of protection. Along with this will come an increased application of spot welding, further contributing to reduction in costs. By virtue of the higher strengths of the materials, there will be lighter structures, increasing the pay load.

An illustration follows of approximately what might be expected in a hypothetical plane when fabricated from either 17ST or Alclad 24SRT. With 17ST material, minimum yield point of 30,000 lb per sq in.: Gross weight, 13500 lb. Weight, empty (everything except fuel, oil, crew, and pay load), 7600 lb. Useful load (crew, passengers, oil, fuel, and cargo), 5900 lb. Pay load, 3643 lb. With Alclad 24SRT material, minimum yield point of 46,000 lb per sq in.: Increase in yield point over 17ST, 53 per cent. Decrease in structural weight, 906 lb (0.53 × 3420 lb × 1/2 = 906 lb). Increase in pay load, 906 lb. Pay load, 4549 lb. Percentage of increase in pay load, 24.8.

Calculations have been based on guaranteed minimum yield point. It is realized that there are structural parts in which form failure may occur before the full yield strength of the material will develop, and that the application of a material with a higher yield point in members subjected to compressive loading would not, in general, enable a reduction in weight in a direct ratio to the increase in mechanical properties. Tensile members, however, are not subjected to this limitation. In a composite structure the assumption is made that one-half of the decrease in structural weight that is obtained if a material of higher yield strength is applied can actually be realized in an airplane.

Discussion

R. S. Barnaby.[4] One of the most interesting developments in naval aircraft structures, from the material engineer's point of view, has been that connected with the use of the strong aluminum alloys. It has been an uphill road, at best. Prior to some time around 1920 or 1921 the use of aluminum was avoided because of its susceptibility to corrosion when exposed to sea service. When the American equivalent of duralumin, the German alloy made famous by its use in the Zeppelin airships, was put on the market, it was hailed as the "corrosion-resistant" strong aluminum alloy, and the Navy forthwith incorporated a lot of gussets of this material in some wooden floats for seaplanes. No finish of any kind was used. Results were speedy, conclusive, and disappointing.

Shortly after this sad experience the phenomenon now known as "intercrystalline corrosion" was recognized, and it became apparent that while this high-strength alloy might be more corrosion resistant than some of the other aluminum products, it was not practicable to use it without protection. This brought up the problem of finish—a difficult one because it is notably hard to get paints to stick to aluminum.

One by one these problems have been attacked and solved. Proper heat treatment has been found to practically eliminate intercrystalline attack. The anodic oxide film has not only increased the corrosion resistance measurably, but has improved paint adherence so that it is now possible to get paint coatings to stick satisfactorily. The introduction of Alclad has further strengthened the defenses against corrosion, so that now aluminum alloy is a standard structural material for naval aircraft.

A point, however, that cannot be too strongly emphasized is that extreme care in design, construction, and maintenance is the price which must be paid to obtain satisfactory results. No paint coats are perfect. Most of them are not entirely impervious to water, especially if the water can stand on the coating long enough. Consequently, designs must be such that there are no places where water can be trapped and stand in contact with the material for long periods of time. Undrained ledges, pockets, corners, etc., must be avoided. Seams and lap joints must be completely filled with some protective compound. The use of leather, fabric, or wood in contact with aluminum should be avoided. Where such a combination is necessary, the leather, fabric, or wood used must be impregnated with some water-excluding material so that it will not become water soaked and foster corrosion of the metal.

Probably the most important and at the same time most difficult thing to eliminate is in connection with the use of dissimilar metals in contact. Where such joints occur in the presence of moisture, corrosion is greatly accelerated due to the electrolytic reaction. While copper, brasses, and bronzes are the worse offenders, stainless steel is almost equally bad, and recent serious corrosion in Navy flying boats has occurred where cadmium-plated chrome-molybdenum steel and aluminum alloy have been used in contact with one another.

These various difficulties are mentioned, not in a discouraging vein, but to indicate the practical precautions which years of service have shown to be necessary if the advantages of the aluminum alloys are to be enjoyed in seaplane and seaboard aircraft construction.

The Navy has not found that one coat of primer and one coat of paint are sufficient. At least two coats of paint over the primer are necessary to secure adequate protection. While it appears possible that bare Alclad may be used for many parts of land planes, it should be borne in mind that the aluminum shell does corrode, and that while its corrosion has little effect on the physical properties of the core metal, its appearance, when used in a place where it cannot be kept wiped off and polished, is likely to be disconcerting.

The writer would like to refer to Table 1 and call attention to the fact that the properties listed are the average mechanical properties, and are therefore not figures which the designer may use in the design of aircraft. He must use lower figures—those which the Aluminum Company will guarantee. These will be found to be some 8 per cent lower in the case of the ultimate strengths, 10 to 15 per cent lower in the case of the yield points, and 10 to 15 per cent lower for the elongations. The design or guarantee figures should be added to Table 1 to make it complete for the designer's use.

[4] Lieutenant-Commander (CC), U. S. Navy; Bureau of Aeronautics, Navy Department, Washington, D. C.

Authors' Closure

By C. F. Nagel, Jr. Commander Barnaby states, "When the American equivalent of duralumin, the German alloy made famous by its use in the Zeppelin airships, was put on the market, it was hailed as the 'corrosion-resistant' strong aluminum alloy, and the Navy forthwith incorporated a lot of gussets of this material in some wooden floats for seaplanes. No finish of any kind was used. Results were speedy, conclusive, and disappointing."

I have heard this statement made previously and in some cases, I know, with the belief that the American manufacturer desired to create the impression that duralumin was resistant to corrosion to the extent that it could be used successfully in the bare unprotected state. It so happens that I have been connected with the development and production of duralumin, or 17S, as put out by the Aluminum company, since 1915, which marks the date when this was first done in this country, and hence I am in a position to know the attitude the Aluminum company has taken. Of course, we know far more about this material today than we did in that early period. However, we have at no time advocated the use of this material bare and unprotected. I agree with Commander Barnaby, on the other hand, that in many quarters the impression did exist that duralumin was so resistant to corrosion that it could be successfully used bare and unprotected under a great variety of service conditions. In my opinion, this unwarranted impression arose from rather too optimistic statements in publications, principally those from abroad.

With reference to the statement that "it is notably hard to get paints to stick to aluminum," in some measure this is true, but it might be interpreted wrongly. Normally, aluminum and aluminum-alloy sheets are produced with a much smoother or glossier surface than some other common metals, as for example steel. The adherence of paint to a surface is enhanced by a slight roughness due to a mechanical bond and is conversely made more difficult with increasing smoothness. To this extent, I subscribe to Commander Barnaby's statement. On the other hand, one might interpret this statement as implying that the characteristics of aluminum and aluminum alloys are such as to make it more difficult for paint to adhere to it than to steel. I am quite positive that the opposite is true. The ability of paint to adhere to a metal surface is also influenced by the inherent corrosion resistance of the underlying metal. All corrosion is attended with the formation of corrosion products, and I believe that in most cases these corrosion products, in addition to oxide, consist of gas. The formation of these products, therefore, tends to destroy the bond and to strip off the paint. All the aluminum alloys employed in aircraft possess an inherently greater corrosion resistance than the normal variety of steels, excluding the stainless steels, and I am sure Commander Barnaby will subscribe to the statement that, if the glossy surface will be removed from aluminum, the paint will adhere more readily and retain its integrity better on aluminum and aluminum alloys than on steel.

AER-55-11

Relation of Quality to Economics of Manufacture of Aviation Gasoline

By G. G. OBERFELL,[1] BARTLESVILLE, OKLA.

Quality of aviation gasoline is the problem of the fuel manufacturer and the designer of aircraft engines. Present aviation gasoline is the result of demands for an increase of power output per unit of weight and displacement of aircraft engines. The improved engines represent designs intended to take full advantage of the best grades of gasoline available in sufficient quantities. Any future improvement in gasoline quality should bring about commensurate improvements in gasoline engines. The quality of aviation gasoline is thus setting the pace for improvement in aircraft engines of the spark-ignition type in use at present.



THE quality of aviation gasoline is a mutual problem of the fuel manufacturer and the designer of aircraft engines. The aviation gasoline of today represents the results of improvements in fuel-manufacturing processes as effected by the insistent demand for an increase of power output per unit of weight and displacement of aircraft engines. The improved engines represent designs which are intended to take full advantage of the best grades of gasoline available in sufficient quantities. It is therefore to be expected that any future improvements in the quality of aviation gasoline, if the fuel is available in sufficient quantities, will bring about commensurate improvements of aircraft engines. To this extent, then, the quality of aviation gasoline is setting the pace for future improvements in aircraft engines of the spark-ignition type in present use.

The properties of gasoline essential to good operation of aircraft engines may be listed as follows: 1 A.S.T.M. distillation range, volatility. 2 Reid vapor pressure, vapor lock. 3 Octane number, anti-knock characteristics. 4 Gum content. There are other properties which, while probably not less essential, are common to practically all gasolines, such as absence of entrained water, low sulphur content, freedom from acids, alkalis and non-volatile residues, color, odor, etc. This paper is confined to the four properties first mentioned, since the differences in quality of aviation gasolines are due primarily to differences in those four major properties.

In general the properties of fuel necessary for satisfactory performance of automobile engines are the same as those required for aviation gasoline. It is mainly a question of the magnitude or the degree to which emphasis for the particular property is given. For the purpose of explaining and emphasizing the important characteristics of aviation gasoline, the properties and specifications of motor fuels as well as of aviation gasolines are included in this discussion. The term "motor fuel" refers to gasolines intended for automobile engines. In reviewing properties and testing methods, the author has purposely omitted those tests which have been found by experience to be less significant of the characteristics of the fuel which they were intended to valuate.

A.S.T.M. Distillation Range, Volatility

The A.S.T.M. distillation test (D 86–30) is useful only because it has been correlated with engine performance. The same statement holds true for any satisfactory testing method, because the value of the testing method is limited by the degree to which it has been correlated with actual practice. While the conditions under which the gasoline is distilled in the A.S.T.M. distillation test are vastly different from the conditions under which the gasoline is distilled or vaporized in the induction system of an internal-combustion engine, relations have been established between test results and performance of engines in actual service. The A.S.T.M. distillation test therefore affords a satisfactory measure of the volatility characteristics of gasolines.

It has been found that the 10 per cent, 50 per cent, and 90 per cent evaporated temperatures of the A.S.T.M. distillation test will give the required information regarding the volatility of gasoline. These points are therefore the ones to be considered for specification purposes. A low 10 per cent temperature denotes good starting and contributes to the accelerating characteristics of motor fuels. It is the volatility of that portion of the gasoline up to the 10 per cent point that assures quick starting rather than the 10 per cent temperature alone. In the case of an exceptional fuel having a high vapor pressure, but otherwise little front-end volatility, the vapor-pressure test discussed farther on is a better criterion of the starting qualities than is the 10 per cent point. For this reason, the vapor-pressure test and the 10 per cent temperature are both useful from the viewpoint of ease of starting. The 50 per cent evaporated temperature is a criterion of the warming-up and accelerating characteristics of gasolines. Since these factors depend upon the proportion of fuel-air mixture and its distribution to the engine cylinders, it is more a question of the volatility or actual boiling-point range of a gasoline up to the 50 per cent point than the 50 per cent temperature alone. Therefore, a consideration of vapor pressure and of the 10 per cent and the 50 per cent temperatures is useful in this connection. The 90 per cent temperature has been correlated with the temperature

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

at which condensation takes place from a mixture of gasoline, vapor, and air. This temperature, which is the dew point of the mixture, is of course a measure of the tendency toward dilution of crankcase oil due to the presence of high boiling constituents of the gasoline. Due to the relatively high overall volatility of aviation gasoline and to the high operating temperatures of air-cooled aviation engines, crankcase dilution is not as important to the aviation industry as it is to the automobile industry.

The relations which have been established by many investigators prove definitely that the 90 per cent evaporated temperature is a much better criterion of oil-dilution tendencies than is the A.S.T.M. end point that was formerly used. Specifications which limit the percentage of residue left in the flask

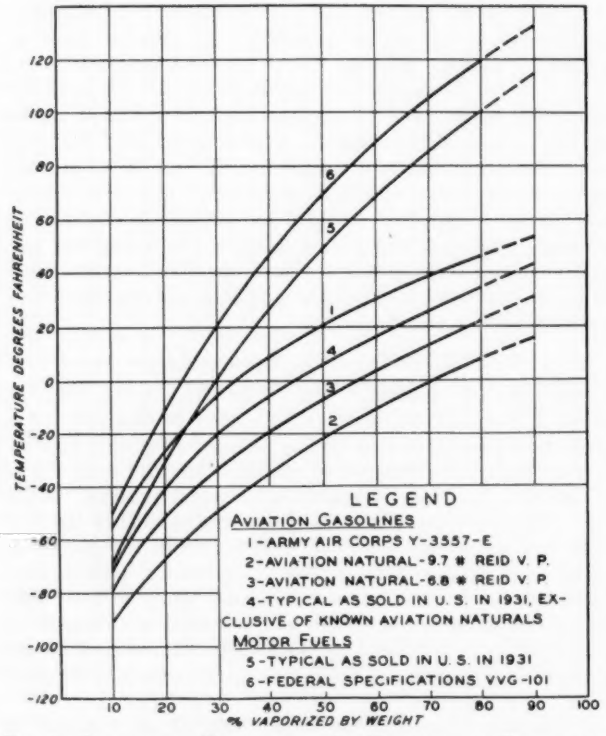


FIG. 1 CALCULATED EQUILIBRIUM AIR-DISTILLATION CURVES FOR AVIATION GASOLINES AND MOTOR FUELS

at the end of the A.S.T.M. distillation test prevent the presence of appreciable quantities of non-volatile residues. However, the distillation processes used by the motor-fuel manufacturers preclude the presence of any non-volatile residue. For this reason no difficulty is experienced in manufacturing gasolines to pass this specification. It is only by accidental contamination with other products that the presence of a non-volatile residue would be encountered.

The A.S.T.M. distillation test is a batch process; as the distillation proceeds, the undistilled portions become progressively less volatile. In the induction system of the engine the gasoline is partially vaporized in an air stream. In order to simulate engine-operating conditions, various investigators have conducted laboratory tests in which there were determined the percentages of gasoline evaporated at different temperatures in a stream of air in which the air-vapor mixture was in equilibrium with the unvaporized portions. Due to the relations which have been established between the equilibrium air distillation and the A.S.T.M. distillation test, it is possible to calculate for various air-fuel ratios the approximate equilibrium air distillation or volatility characteristics of gasoline from the A.S.T.M. distillation test. Calculations of this nature for a 12-to-1 air-fuel mixture are shown for various types of gasolines in Fig. 1. The data show that there is a pronounced difference between the relative volatility characteristics of the various gasolines. Curve 1 is an aviation gasoline complying with specification Y-3557E of the U. S. Army Air Corps, curve 2 represents a natural gasoline product having a Reid vapor pressure of 9.7 lb, while curve 3 is the same natural gasoline product reduced to a 6.8 lb Reid vapor pressure by distilling off part of the pentanes, the components of which are almost entirely responsible for imparting the front-end volatility to aviation natural gasoline. A comparison of curves 2 and 3 shows that in going from 9.7 lb to 6.8 lb there is a noticeable reduction in the overall volatility of the gasoline, which lowers the quality in this respect due to the more rigid 7-lb vapor-pressure specification. Reputable pilots with years of experience claim that there is a very noticeable difference between these two fuels when flying a plane with the surrounding atmospheric temperature below —20 F. Curve 4 represents a typical aviation gasoline as computed from the A.S.T.M. distillation tests of 329 samples obtained as the result of our survey of aviation gasolines marketed in the United States by the major companies during the past year. Curve 5 represents the typical motor fuel sold in the United States during 1931. The gasoline for which curve 6 was plotted coincides with U. S. motor gasoline specifications VV-G-101, which superseded 622A. An inspection of the curves emphasizes the much higher overall volatility characteristics of aviation gasoline as compared with motor fuels.

In the manufacture of gasoline the volatility of the product is controlled by segregation of products, by blending, and by distillation processes. If the gasoline is manufactured from crude oil, the volatility problem is primarily one of eliminating the high-boiling-point constituents or converting those components into low-boiling compounds in order to bring the volatility of the resulting product up to specifications. For the

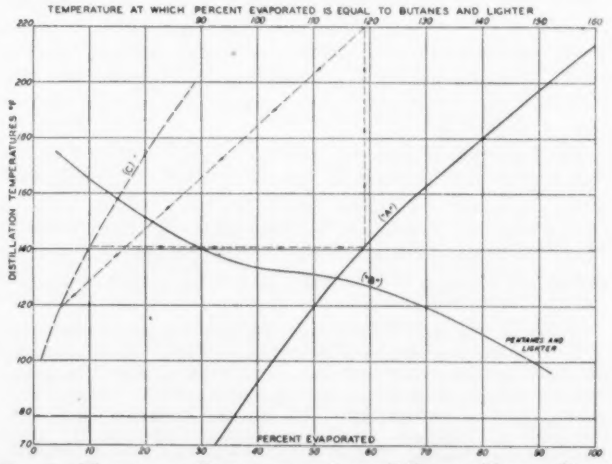


FIG. 2 METHOD FOR DETERMINING A AND B FACTOR VALUES FROM A.S.T.M. DISTILLATION CURVE

gasoline extracted from natural gas or from refinery vapors, the problem is just the opposite in that the low-boiling-point constituents are the ones removed in order to bring the vapor pressure of the gasoline down to the required specifications. In both of these cases a decrease in yield takes place in which the rejected products are wasted or are diverted to uses for which there exists a lower market value. Segregation of products always entails extra equipment and higher operating costs. Demands for higher quality for any commodity should always take into consideration all of these economic factors.

Reid Vapor Pressure, Vapor Lock

The vapor pressure of a liquid is a measure of its tendency to boil. A vapor-pressure test is in effect merely a more accurate and more satisfactory means of determining the initial boiling point of the gasoline than can be obtained by the A.S.T.M. distillation test. Vapor pressure is therefore dependent upon the true initial boiling point and is distinguished from volatility in that the latter is dependent upon the entire distillation range, although for all practical purposes volatility can be defined by the 10, 50, and 90 per cent evaporated temperatures.

Many testing methods have been proposed for measuring the vapor pressure of gasoline, and test results of the various methods have been compared with vapor-lock conditions that would be encountered in the operation of airplanes and automobiles. So far the Reid vapor-pressure method (D 323–31T) has given a better indication of vapor-lock conditions than any of the other methods which have been proposed.

The possibilities of establishing a relation between the A.S.T.M. distillation test and the Reid vapor-pressure test has been investigated by several laboratories. There is shown in Fig. 2 a relation which has been found as a result of work by our research department. This correlation was made possible by having in addition to the Reid vapor-pressure and A.S.T.M. distillation test results, a composition analysis covering the percentages of butane and pentane fractions for 270 samples of motor and aviation gasolines marketed throughout the

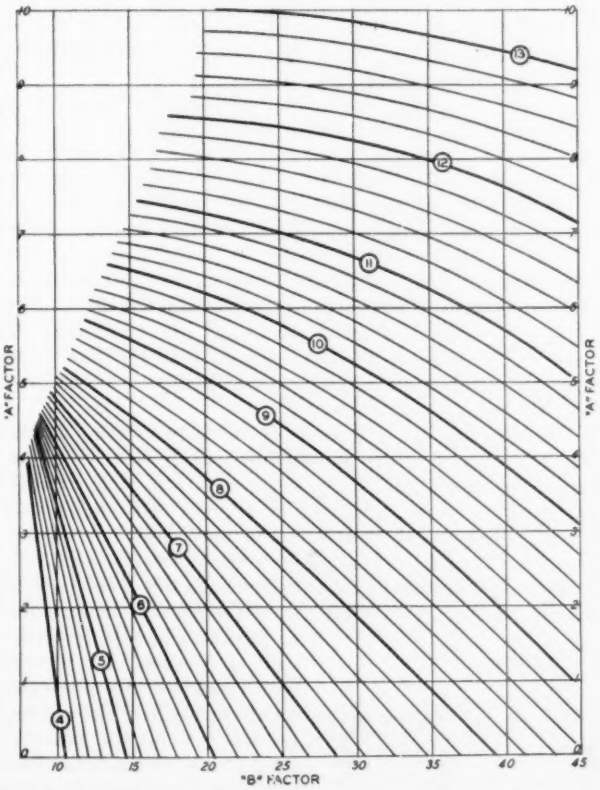


Fig. 3 Reid Vapor Pressure as Indicated by A and B Factor Relationship

United States. A relationship was established between the A.S.T.M. 10 per cent evaporated temperature and the temperature at which was distilled off that volume which was equal to the percentages of butanes and lighter found by actual composition analysis. A similar relation was found for the pentanes and lighter.

An illustration will serve to show how Fig. 2 is used. The heavy dashed curve C is plotted from the A.S.T.M. distillation record of the gasoline for which it is desired to estimate the Reid vapor pressure. The 10 per cent evaporated temperature of this particular fuel as read from the curve is 141 F. A line is drawn horizontally from this 10 per cent evaporated point to the curve A, and at the point of intersection with the curve A another line is drawn vertically to the upper axis which shows a temperature of 119 F as being the temperature in the A.S.T.M. distillation test at which is distilled off a percentage which should be equal to that of the butanes and lighter as found by composition analysis. The volume evaporated at 119 F in the A.S.T.M. test was 4.8 per cent as read from curve C.

The volume evaporated during the A.S.T.M. distillation test which was equal to the percentage of pentanes and lighter is obtained by noting the amount evaporated at the point of intersection of the curve C with the curve B. This point of intersection represents 14.9 per cent evaporated as read from the lower horizontal axis. This figure of 14.9 is a measure of the percentage of pentanes and lighter in the gasoline. To obtain the Reid vapor pressure of the sample refer to Fig. 3. The group of parallel curves represents Reid vapor pressure values. The A factor used as the ordinate denotes the percentage of butanes and lighter. The B factor used as the abscissa is the percentage of pentane and lighter. To estimate the Reid vapor pressure of the sample cited, find the intersection of the ordinate 4.8 with the abscissa 14.9. This point of intersection lies on the curve representing 8.2 lb Reid vapor pressure. The composition analysis and the Reid vapor pressure of the gasoline used in the example were as follows:

	Observed	Estimated
Propane and lighter, per cent	0.5	..
Butane fraction, per cent	4.7	..
Butanes and lighter, per cent	5.2	4.8
Pentane fraction, per cent	10.6	10.1
Pentanes and lighter, per cent	15.8	14.9
Hexane and heavier, per cent	84.2	85.1
Reid vapor pressure, lb	8.6	8.2

There is presented in Table 1 a list of samples selected for the

TABLE 1 COMPARISON OF OBSERVED AND ESTIMATED REID VAPOR PRESSURE

S. S. No.	Factors A	Factors B	Reid vapor pressure Observed	Reid vapor pressure Calculated	Deviation	Type of gasoline	Location
3832	7.1	21.5	11.3	11.0	−0.3	Motor natural	Minneapolis, Minn.
3855	3.0	33.7	9.1	8.95	−0.15	Aviation natural	Wichita, Kan.
3997	7.8	24.2	11.85	11.6	−0.25	Motor natural	Madison, Wis.
4047	9.0	18.5	12.4	12.25	−0.15	Refinery motor	Stow, Ohio
4119	4.9	21.7	9.9	9.05	−0.85	Refinery aviation	Cheyenne, Wyo.
4230	6.2	14.8	9.8	9.7	−0.1	Refinery motor	Amarillo, Tex.
4466	8.3	21.3	11.4	11.8	+0.4	Refinery motor	Tulsa, Okla.
4722	4.5	18.3	8.65	8.35	−0.3	Refinery aviation	Wichita, Kan.
4884	4.0	35.2	9.8	9.7	−0.1	Aviation natural	Bartlesville, Okla.
3432	4.3	12.4	6.6	7.0	+0.4	Refinery motor	Denver, Col.
3242	4.2	15.5	7.65	7.7	+0.05	Refinery motor	Lynn, Mass.
3273	3.7	13.8	6.35	6.75	+0.4	Refinery motor	Tulsa, Okla.
3243	3.1	21.0	7.35	7.65	+0.3	Refinery aviation	Hudson, Ohio
3241	1.8	22.0	6.85	7.0	+0.15	Refinery aviation	Boston, Mass.
3240	3.2	16.6	7.5	7.0	−0.5	Refinery aviation	Haskell, N. J.
3151	2.8	10.8	4.95	5.0	+0.05	Refinery motor	St. Louis, Mo.
3145	3.3	10.6	5.5	5.25	−0.25	Refinery motor	Amarillo, Tex.
3155	4.1	12.7	7.1	6.95	−0.15	Refinery motor	S. Sudbury, Mass.
2920	2.6	25.0	7.5	7.8	+0.3	Refinery aviation	Cheyenne, Wyo.
2994	3.0	12.4	5.4	5.75	+0.35	Refinery motor	Pontiac, Mich.
2983	2.7	13.3	5.8	5.8	0.0	Refinery motor	Amarillo, Tex.
3112	2.0	18.3	6.25	6.5	−0.25	Refinery aviation	Omaha, Neb.
3111	3.0	12.5	5.8	5.75	−0.05	Refinery aviation	Omaha, Neb.
3107	2.5	20.4	7.35	7.2	−0.15	Refinery aviation	Denver, Col.
3106	1.7	31.5	8.4	8.1	−0.3	Aviation natural	Des Moines, Iowa
3104	2.1	12.8	5.3	5.3	0.0	Refinery aviation	Boston, Mass.
3085	1.9	24.5	7.25	7.4	+0.15	Refinery aviation	Tulsa, Okla.
2991	3.0	11.2	5.5	5.25	−0.25	Refinery motor	Pontiac, Mich.
2965	5.5	15.1	8.85	8.9	+0.05	Refinery motor	Baltimore, Md.
3083	3.7	32.6	9.25	9.25	0.0	Aviation natural	Tulsa, Okla.
4977	6.2	22.7	10.2	10.2	0.0	Refinery motor	Kansas City, Mo.
6355	7.5	22.5	11.15	11.3	+0.15	Refinery motor	Okmulgee, Okla.
6428	6.7	18.8	10.2	10.4	+0.2	Refinery motor	Eldorado, Kan.
6524	7.1	24.2	10.4	11.05	+0.65	Motor natural	St. Louis, Mo.
6555	5.7	17.2	9.1	9.3	+0.2	Refinery motor	Denver, Col.
6546	5.9	15.5	9.0	9.35	+0.35	Refinery motor	Pontiac, Mich.
6611	6.0	17.5	9.4	9.65	+0.25	Refinery motor	Minneapolis, Minn.
6614	5.9	17.6	9.6	9.55	−0.05	Refinery motor	Houston, Tex.
6675	6.3	19.3	10.15	10.1	−0.05	Refinery motor	Tulsa, Okla.
6668	5.5	18.1	9.3	9.2	−0.1	Refinery motor	Indianapolis, Ind.

purpose of showing the wide range of products for which the described graphical method has been found applicable.

Experience with the graphical method of estimating Reid vapor pressures shows that the predicted results are in most cases within the limit of accuracy of the testing method, which is about 0.5 lb. The method is not proposed as a means of eliminating the Reid vapor pressure test, but it does provide an independent check which has been found to give reasonably accurate estimations. It is therefore useful in those cases in which only the results of the A.S.T.M. distillation test are available.

In actual practice vapor locking is occasioned by the rise in temperature of the gasoline in the fuel-feed system. This rise in temperature is caused by heat originating from the engine. If the temperature rise is high enough, the vapor pressure of the gasoline may increase enough to cause boiling and thereby produce vapor locking. One of the difficulties encountered in obtaining a direct measurement of the vapor-lock temperatures from the vapor pressure of the gasoline is explained by the fact that the vapor pressure of the gasoline is lowered by any evaporation loss that takes place from the time that the vapor-pressure test is made until the gasoline is used. These losses occur in transit, unloading, storage, and transfer of gasoline to the airplane. The magnitude of change in vapor pressure of gasoline due to evaporation loss differs for gasolines of widely different A.S.T.M. distillation characteristics. Thus two gasolines having the same original vapor pressure and undergoing the same percentage of evaporation loss might have decidedly different vapor pressures after this loss had been encountered. The gasoline with the flattest A.S.T.M. distillation curve would change less in vapor pressure for any given evaporation loss, and consequently would be susceptible to vapor lock at a lower temperature.

In addition to this ever-changing vapor-pressure condition of all gasolines prior to usage, another complication of determining the vapor-locking temperature from the vapor-pressure measurement is encountered due to the fact that the fuel-feed systems will accommodate a certain amount of vapors due to incipient boiling of gasoline which may take place before vapor lock occurs. Consequently, it is not the original vapor pressure of the gasoline but the vapor pressure of the gasoline after a certain amount of evaporation or boiling has taken place that determines the vapor-locking temperature of that gasoline. Moreover, the amount of evaporation that can be accommodated in the fuel-feed systems without vapor locking varies with the different fuel systems and for the same system under different operating conditions. Despite all of these variables, the Reid vapor pressure has proved to be a fairly good criterion of the vapor-locking characteristics of the gasoline.

Much has been said regarding the vapor pressure of the more volatile components of gasoline. Some of the more important properties of these highly volatile fractions are shown in Table 2. It is believed that the practical deleterious effects of the presence of propane in gasoline has been overemphasized. In any event, erroneous impressions have frequently been gained from the various published statements in this connection. For gasoline of the same original vapor pressure, propane is beneficial rather than detrimental when looked at from the viewpoint of vapor lock alone. For reasons given, it is evident that for gasolines of the same original vapor pressure the gasoline with propane would lose its vapor pressure more rapidly due to the same amount of evaporation loss than would the gasoline without propane. This means that if two gasolines leave the refinery with the same vapor pressure, one with propane and one without, and both gasolines are subjected to exactly the same conditions up to the time that vapor lock occurs, the one with propane will have the higher vapor-lock temperature. It also means that if two gasolines have the same vapor pressure when they enter the fuel-feed system, the one with propane will have the higher vapor-lock temperature in actual service if all conditions of service are exactly the same in both cases.

The elimination of propane is purely an economic problem. It concerns the fuel manufacturer primarily. The aviation and automotive industries need not give this problem any serious consideration. The progressive fuel manufacturers will of necessity solve this economic problem because elimination of propane means an increase in yield of gasoline from a given quantity of raw material. As a matter of fact, most gasolines contain only small quantities of propane. Of 24 aviation gasolines analyzed during the past year none contained more than a trace of propane.

In order to obtain some idea regarding the vapor-locking tendencies of gasoline of the same Reid vapor pressure, but having different volatility characteristics as evidenced by the A.S.T.M. distillation test, our laboratory conducted tests with a Waukesha knock-test engine to determine vapor-lock temperatures under carefully controlled conditions in such a laboratory unit. The relationship which we found cannot be considered more than an indication of trends. But certain tendencies were definitely established as far as vapor-lock temperatures under these particular conditions are concerned. For gasolines having the same Reid vapor pressure, but with widely varying distillation characteristics, the vapor-lock temperatures can vary to a measurable degree.

The vapor pressure of gasoline is controlled in plant practice by segregation of stocks, by blending, and by distillation processes. Frequently the vapor pressure is too high, and the more volatile components, such as propane and butane, must be removed without the loss of less volatile constituents. Otherwise, the reduction in vapor pressure will be accomplished at the sacrifice of plant yield and of front-end volatility of the resulting product. It is a real problem for the motor-fuel manufacturer to produce a fuel that combines ease of starting and freedom from vapor lock due to the limitations of many of the fuel-feed systems. Aviation gasoline should have the maximum volatility judiciously correlated with the required vapor-pressure specification.

It is important that any vapor-pressure specification for gasoline should take into consideration the point at which the sample for the vapor-pressure test is secured. A large number

TABLE 2 PROPERTIES OF VOLATILE FRACTIONS OF GASOLINE

Fraction	Sp. gr.	Theoretical vapor pressure at 100 F (lb per sq in. abs)	Increase in Reid vapor pressure due to presence of 1% by volume (lb per sq in. at 100 F)	Percentage present in average gasoline by volume	Lowest easy starting temperature, deg F	Octane number	Percentage that can be included in 12-lb R.V.P. gasoline if fractions more volatile are absent
Propane	0.509	187	2.46	0.2	−137	[a]	...
Iso-butane	0.564	73.9	..	..	−100	100	...
N-butane	0.584	52.4	..	..	− 82	92	...
Butanes	0.581	55.7	0.6	4.9	− 90	93	14.2
Iso-pentane	0.625	20.4	..	..	− 49	91	...
N-pentane	0.631	15.6	..	..	− 37	64	...
Pentanes	0.628	16.5	0.13	9.8	− 45	74	51
N-hexane	0.663	4.9	[b]	..	+ 7	35[c]	100
Hexanes	0.669	8.3	[b]	12.1	− 20	63	100
N-heptane	0.688	1.6	[b]	..	+ 44	0	100
Heptanes	0.724	2.9	[b]	13.0	+ 22	65	100

[a] Considerably better than iso-butane. [b] Negligible. [c] Estimated.

of tests have shown that normal evaporation losses incurred in transportation, unloading, storage, and filling the airplane tank cause a drop in Reid vapor pressure of about 1.0 lb between the tank car at point of shipment and the airplane gasoline tank. This loss in vapor pressure varies, as explained, depending upon the vapor pressure of the gasoline, upon its A.S.T.M. distillation characteristics, and upon the conditions of storing and handling the product. The high octane number of the light hydrocarbons coupled with their increasing availability should demonstrate that anything the aviation industry does which will permit of higher vapor pressures will improve the quality of the gasoline as evidenced by ease of starting, higher overall volatility, and better anti-knock characteristics. Improved fuel systems and rational vapor-pressure specifications are the only answers to the problem.

Octane Number, Antiknock Characteristics

It is with considerable trepidation that the author treads upon the ground belonging to octane numbers and anti-knock characteristics of motor fuels. Much work has been done by eminent authorities upon this subject. About one year ago one of them stated: "While anything like a complete knowledge of the relative behavior of different fuels in multi-cylinder aircraft engines and fuel-test engines is probably several years off, a test which provides partial control of anti-knock value is preferable to no test at all."

Most of the efforts thus far developed for determination of anti-knock characteristics have been devoted to motor fuels. The nearest approach to standards for aviation gasolines are the methods employed by the military services, particularly the ones in use by the Army Air Corps. Even in the case of motor fuels much doubt still exists as to the significance of the test results determined by methods now prevalent.

The Cooperative Fuel Research Steering Committee is to be highly commended for its work in developing the test procedure and apparatus for determining octane numbers of motor fuels. It is believed that the impelling motive to establish test conditions that will give duplicable results by different testing laboratories has to a certain extent overshadowed the necessity for having the test results afford a measure of the anti-knock characteristics of the gasoline under actual service conditions. When it appeared that the matter of knock testing of motor fuels was pretty well solved, further discrepancies appeared which have delayed the final standardization of the knock-testing method. Recognizing these deficiencies of the present testing methods, the U. S. Army Air Corps, the Cooperative Fuel Research Steering Committee, and the Ethyl Gasoline Corporation have adopted or have considered the adoption of test conditions that will insure more definite relations between test results and actual service conditions of aircraft engines and automotive engines. The general trend is toward an increase in jacket temperature and speed of the test engine.

It is well recognized that the magnitude of changes in anti-knock characteristics due to changes in engine-operating conditions depends upon the composition of the gasoline. Since no two gasolines are exactly alike in composition due to the large number and the complex nature of the compounds present in gasoline, it is imperative that any fuel used as a standard for knock reference should be capable of exact duplication. Normal heptane and iso-octane provide the desired requisites for such a standard. Normal heptane is a notorious knocker. Iso-octane has excellent anti-knock characteristics, and at the same time all of its other properties are similar to normal heptane. Since each of these compounds can be prepared in a high state of purity and for the reason that each substance is of definite chemical composition, they provide the basic requirements for the standards of knock reference. In addition, the wide range in anti-knock characteristic of heptane and octane makes it possible to determine the anti-knock characteristic of a sample of gasoline by matching it with a known mixture of the two pure compounds.

The octane number of a gasoline is defined as follows: "The octane number of any gasoline is the percentage of octane by volume in the mixture of octane and heptane that just matches the gasoline in anti-knock quality, as determined in the apparatus described and by the procedure specified." For example, if a gasoline is matched by a mixture of 65 parts of octane and 35 parts of heptane, its octane number would be 65.

Octane numbers above 100 are expressed numerically as 100 plus the percentage by volume of benzene in a mixture of benzene and octane. By definition, heptane would have an octane number of zero, iso-octane would have an octane number of 100, and benzene would have an octane number of 200.

From the title of this section it might appear that octane number and anti-knock value are synonymous. Such is not the case. The octane number of a motor fuel is a figure determined by a prescribed testing method and may not be an accurate indication of the anti-knock value of the gasoline. For example, investigations have been reported indicating that straight-run gasoline with an octane number of 64 evidences about the same tendencies to knock in automobiles as an average cracked gasoline with an octane number of about 69. In cases of gasoline made by very intensive cracking, this difference in octane numbers for equivalent knocking tendencies between straight-run and cracked gasoline may amount to 10 octane numbers or even more. Changes in the testing method in which a higher engine-jacket temperature and higher speed are employed may prove to give octane numbers having a more definite relation to anti-knock values encountered in actual service.

Two testing methods for determining the octane numbers of gasolines are in general use. One of these which was developed by the Cooperative Fuel Research Steering Committee thus far is used almost entirely in testing motor fuels. The more important conditions under which this test is conducted are as follows:

1 Engine, C.F.R. unit, manufactured by the Waukesha Motor Company, Waukesha, Wis.
2 Engine speed, 600 rpm
3 Engine-jacket temperature, 212 F
4 Spark advance, for maximum power, automatically controlled (basic setting 15 to 22.5 deg, depending upon compression ratio)
5 Carburetor adjustment, for maximum knock
6 Knock indicator, bouncing pin.

Aviation gasolines are tested by many laboratories with the series 30 engine of the Ethyl Gasoline Corporation. The essential requirements for tests of aviation gasoline for the U. S. Army Air Corps (Y-3557E) are as follows:

1 Engine, Ethyl Gasoline Corporation series 30 knock-test engine
2 Engine speed, 900 rpm
3 Engine jacket temperature, 375 F
4 Spark advance, 15 deg
5 Carburetor adjustment, for maximum knock
6 Knock indicator, temperature plug.

The paraffin hydrocarbons have greater anti-knock stability; that is, they retain their anti-knock characteristics over a wider range of engine-test conditions than do the unsaturated and

aromatic hydrocarbons. This is particularly true in the case of increased engine-jacket temperatures. The paraffin hydrocarbons are in general also more responsive to increase in octane number resulting from the addition of the first few cubic centimeters of tetraethyl lead. Straight-run refinery gasoline and gasoline from natural gas are composed primarily of the paraffin hydrocarbons. Generally speaking, it also appears that the more volatile paraffin hydrocarbons possess the two very desirable characteristics of anti-knock stability and lead responsiveness to a greater extent than do the less volatile paraffins.

It is undoubtedly true that progress in standardization and correlation of knock-testing methods with actual service conditions has been delayed, due in part to the costs of the primary standards, normal heptane and iso-octane, which sell for $25 per gallon. This price situation has necessitated the use of secondary standards. Due to the inherent anti-knock characteristics of the more volatile normal paraffins, the suggestion is made to use the strictly normal paraffin compounds of propane, normal butane, and normal pentane as primary standards. The gasoline industry can offer propane, normal butane, and normal pentane, having a purity of 99 per cent or higher, at a price which ought not to exceed more than about 5 to 10 per cent of the price of the present primary standards.

Many possible combinations of these normal paraffins could be worked out. It would be possible to eliminate both heptane and iso-octane and cover an anti-knock value range greater than that covered by all grades of commercial gasolines. If desired, normal heptane could be retained and thereby cover an even wider range of anti-knock difference. The retention of heptane would provide further possible combinations for devising a satisfactory scale of units for measuring anti-knock characteristics. The numerical ratings could be known as the "butane numbers" or "pentane numbers," whichever proved to be more generally acceptable. Since pentane would be a good base gasoline for comparison of anti-knock values of premium gasolines, the term "pentane number" might be more attractive.

It is realized that difficulties would be encountered in proportioning the fuel-air mixtures in some cases. In addition to the carburetor in present usage, it may be necessary to develop a "high pressure carburetor" or an auxiliary device by which the easily vaporized mixtures could be handled in the vapor state. If the fuel temperatures were maintained at temperatures below —20 F prior to carburetion, it would probably be possible to use a low-pressure type of carburetor in tests of all grades of commercial gasolines. If all products were handled and used under pressure at 60 F, no volumetric corrections would be needed.

Another interesting possibility in connection with the use of paraffin compounds in knock testing is the uniform increase in octane number of the heavier fractions of natural gasoline when lead is added. Due to the narrow boiling-point range of natural gasoline of low vapor pressure, it is possible to economically prepare from it suitable secondary reference standards possessing high knock stability and a lead responsiveness approaching that depicted by a straight-line relationship.

The anti-knock characteristics of gasoline are maintained and controlled in refinery practice by segregation of stocks from which the gasoline can be blended to the required octane number, by varying the volatility of the gasoline, by more extensive use of cracking processes, by blending with natural gasoline products, and by judicious use of lead. As stated, the tendency is toward higher engine-jacket temperature and higher speed during the testing of gasoline for anti-knock characteristics, and since cracked gasoline on account of the presence of unsaturated and aromatic compounds rapidly loses its octane number rating as the engine speed and jacket temperature are increased, it means that cracked gasolines do not have as high anti-knock values as the octane ratings have implied. Cracking processes will therefore have to be revaluated in so far as the economics of producing anti-knock gasoline is concerned.

The anti-knock characteristics of aviation natural gasoline are controlled chiefly by varying the volatility and vapor pressure of the product, to a less extent by segregation of stocks, and by the use of lead. Since aviation natural is composed almost entirely of paraffin hydrocarbons, its high knock stability and responsiveness to lead are in its favor, and as has been previously stated, the more volatile fractions of the natural gasoline show the best behavior in this respect From the standpoint of availability of aviation gasoline of minimum vapor pressure combined with maximum volatility it is imperative that butane be excluded from the aviation gasolines for the same reason that propane should be eliminated from motor fuels. There are enough pentanes available from the natural gasoline industry to supply at economic price levels the wants of the aviation industry for many years. The aviation industry in considering high-quality gasoline is therefore justified in demanding a product free from butane. In manufacturing aviation natural gasoline to meet present vapor-pressure specifications, it is necessary to remove all of the butane and part of the pentane. This procedure of eliminating part of the pentane which has high volatility and valuable anti-knock characteristics results in a fuel of an inferior quality than would otherwise be obtained. There seems to be no reason why all airplane-fuel systems in the future could not be adapted to satisfactorily handle aviation gasolines with a Reid vapor pressure of at least 10.5 lb. Such being the case, aviation natural gasoline would have a pentane content upward of 35 per cent. If a fuel system could be made to satisfactorily handle a gasoline with a Reid vapor pressure of 17 lb, it would be possible to have the fuel composed entirely of pentanes. Such a product would have the maximum possible overall volatility regardless of the source of the petroleum product.

Any limitation on volatility or vapor pressure not founded upon justifiable operating conditions encountered in actual service results in needless sacrifice of the more important qualities of the gasoline. To prevent the inclusion of pentanes in aviation gasoline or of butanes in motor gasoline by unduly rigid vapor-pressure specifications means not only a sacrifice of the valuable properties of the gasoline, but also necessitates an utter waste of natural resources accompanied by additional expenses for lead or a reduction in anti-knock value for the same amount of lead. If the additional volatility and vapor pressure can be safely used, more, better, and lower priced aviation gasoline will be available from the petroleum industry.

Gum Content

If gasoline is allowed to remain in storage for a long period of time, a resin-like deposit may result. Cracked gasolines and especially those from vapor-phase cracking processes are the ones most likely to evidence this phenomenon. If this gummy or resinous material is present in gasoline, it may deposit in the gasoline tank or fuel lines. It is more likely to make its appearance in the fuel-induction system, the intake valves, and the engine, where evaporation of the gasoline causes the gum to be deposited as a carbonaceous residue. Any method for the determination of the gum content of gasoline must take into consideration the fact that the amount of gum present in the gasoline increases with the period of time in which the gasoline is kept in storage. The rate of gum formation is slow at first, and after a certain period of time, depending upon the gasoline and conditions of storage, this rate increases very rapidly. The period of slow rate up to the time that the rapid rate of gum formation begins is frequently referred to as the "induction period."

For the reasons indicated, it has been necessary for the industry to consider two methods for determining the gum content of gasoline. If the gasoline is to be used very soon after the sample has been tested, the test should reveal the amount of gum present in the gasoline when the sample was taken. This determination is generally made by evaporating a measured quantity of gasoline and then determining the amount of residue left upon evaporation.

To determine the gum content of gasoline that may form due to long periods of storage, modification of an accelerated aging test has been used by different laboratories. U. S. Army specification Y-3557E requires that a 200-cc sample of the gasoline shall be subjected to an accelerated aging test at 212 F with oxygen at 100 lb pressure for four hours and that on evaporation of 100 cc of a mixture of the oxidized sample and gum solvent the amount of residue shall not exceed 10 mg.

It is probable that the foregoing test may be unduly severe in that it may exclude cracked gasoline that would give satisfactory performance with respect to gum behavior. It may also be equally true that any gasoline which passes the test will not give any gum-formation difficulties. Much remains to be learned regarding the amount and kind of gum that might be tolerated in gasoline without encountering noteworthy difficulties with engine operation. The low limit of 10 mg may be unduly severe, but it is the opinion of several investigators that an increase in this limit would not be justified until more definite information is obtained regarding the problem.

From a practical viewpoint, the problem of controlling the gum content of gasoline is one that mainly concerns the manufacturer of cracked gasolines. Manufacturing difficulties in this respect are solved by various refining methods, by segregation of stocks, and by the use of gum inhibitors. Natural gasolines are not subject to gum formation, but it occasionally happens that very small quantities of oil are carried over with the gasoline during its distillation from the absorption oil. This trace of oil shows up as gum in the method for determining the gum content. This problem is solved by using more efficient dephlegmators during the distillation process or by redistilling the gasoline.

Gasoline Specifications

A tremendous amount of valuable research work has been carried out by the various industrial organizations and Federal bureaus on the properties of gasoline. The data which were obtained have been used in formulating specifications for aviation gasolines and motor fuels.

The most widely known gasoline specifications are those of the Federal Specifications Board. Not infrequently these specifications are designed to meet special requirements of certain Government agencies and to that extent may be unduly restrictive in certain respects. Unfortunately the prestige carried by the technical bureaus of the Government has the undesirable effect in some instances of generally promulgating the Federal specifications. This is especially true in connection with the gasoline specifications adopted from time to time by the various states.

Many of the states require that the state specifications should be the same as those of the Federal Government and that the state specifications should automatically change with any revisions made in the Federal specifications.

There is presented in Table 3 a summarized review of recent Federal specifications for aviation gasoline. For the purpose of comparing with recent U. S. motor gasoline, specifications for the latter are also included. There has been a gradual tendency to increase the quality of all gasolines as evidenced by the provisions affecting the four major properties. The overall volatility of all gasolines has been increased as shown by the 10, 50, and 90 per cent evaporated temperatures. The specifications relating to vapor pressure have become more rigid, but such restrictions are in effect a backward step as far as quality is concerned. They may be justified from the viewpoint of the nature of the equipment in which the gasoline is being used, but this does not deny the fact that the quality of the gasoline is being sacrificed. The obvious remedy is better equipment as soon as replacements are in order.

Aviation Gasolines Marketed in the United States

For the past four years our company has been conducting a survey of aviation gasolines sold throughout the United States. The samples represent the products of the major refining companies and were in most instances obtained from the various airports in which the gasolines were sold. Extreme care was taken to see that the samples were truly representative of the products being sold. Upon arriving at the laboratory, the following tests upon most of the samples were made: A.S.T.M. distillation (method D 86–30), A.S.T.M. Reid vapor-pressure (method D 323–31T), composition analysis, and anti-knock determinations.

The results of the tests of the aviation gasoline survey are

TABLE 3 U. S. GOVERNMENT GASOLINE SPECIFICATIONS

	—U. S. Navy—		—U. S. Army Air Corps—		——Federal Spec. Board——		
Specification	Motor 7-G-2	Aviation 7-G-3	Domestic Y-3559A	Fighting Y-3557E	Aviation F.A.G.	323-B D.A.G.	U.S.M. VV-G-101
Effective date	2–1–30	2–15–31	5–13–31	1–8–32	1–21–28	1–21–28	10–21–31
Corrosion test	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Gum (copper dish), max.	....	....	3.0 Mg	....	....	....	....
Gum (aging test), max.	....	10.0[b]	15.0[a]	10 Mg[c]	....	....	....
5% recovered, max.	....	....	....	....	149	167	....
10% recovered, min.	140[e]	....	....	....	....	....	....
10% recovered, max.	176	....	....	....	....	....	....
10% evaporated, max.	....	167	167	167	....	....	167[c]
50% recovered, max.	284	....	221	....	203	221	....
50% evaporated, max.	....	212	....	212	....	....	284
90% recovered, max.	392	....	311	....	257	311	....
90% evaporated, max.	....	275	....	275	....	....	392
End point, max.	437	....	374	....	329	374	....
Recovery, %, min.	95	....	96	....	96.0	96.0	....
Loss, max.	....	....	2	....	2.0	2.0	....
Residue, max.	....	2.0	....	2.0	....	....	2.0
Reid vapor pressure, max.	....	6.5	6.5	6.5	....	....	12.0[d]
Color	....	25	....	Blue	25	25	....
Sulphur, max., per cent	0.10	0.10	0.10	0.10	0.10	0.10	0.10
Octane number	....	73	58	87	....	....	....
Type of engine	....	EGC-CFR	EGC	EGC	....	....	....
Compression, lb.	....	220–240	7 1/4–7 3/4R	200–220	....	....	....
Throttle opening	....	65°	....	65°	....	....	....
Mixture ratio for	....	M.K.	M.K.	M.K.	....	....	....
Spark advance	....	20°	14–16°	14–16°	....	....	....
Knock indicator	....	BP-TP	BP	TP	....	....	....
Engine speed, rpm	....	900	600	900	....	....	....
Engine jacket, deg F	....	300	300	375	....	....	....
Permissible $PbEt_4$ (cc)	....	No	....	6.0	....	....	....
Aromatics permissible	....	Yes	Yes	Yes	....	....	....
Acidity	....	....	....	....	Neut.	Neut.	....
Doctor	....	....	....	....	Neg.	Neg.	....
Freezing point, deg F	....	−76.0 F	−76.0 F	−76.0 F	....	....	....

[a] Gasoline in oxygen bomb two hours at 100 lb pressure and 212° F.
[b] Gasoline in oxygen bomb four hours at 100 lb pressure and 212° F.
[c] Maximum 10% evaporated temperature 149 F. where January mean minimum temperature is less than 27 F, applicable during December, January, February, and March (optional).
[d] Optional winter limit of 10 lb and summer limit of 8 lb where January mean minimum temperature exceeds 27 F.
[e] Reduced 5.4 F for each per cent of loss less than 4.0 per cent.
Abbreviations: EGC, Ethyl Gasoline Corporation Series 30; CFR, Cooperative Fuel Research; BP, bouncing pin; TP, temperature plug; M.K., maximum knock; R, ratio; Neut., neutral.

too voluminous to include results of all tests separately. They are presented in summarized form in Table 4.

Since our survey indicated that the aviation gasolines being sold in the United States were undergoing gradual changes in quality, we have tabulated separately characteristics of the aviation gasoline sold last year in order to facilitate comparison with those sold prior to 1931. The following trends are evidenced: Vapor pressures were slightly lower, octane numbers slightly higher, and volatility slightly higher for the 1931 aviation gasolines.

TABLE 4 PERCENTAGE DISTRIBUTION AND RANGES OF PROPERTIES OF AVIATION AND MOTOR GASOLINES

Per cent of samples at or below	Aviation gasolines in 1929 to 1931, inclusive — Evap. temp. 10%	50%	90%	R.V.P.	Octane number[b]	Aviation gasolines in 1931 — Evap. temp. 10%	50%	90%	R.V.P.	Octane number
<1[a]	110	140	196	3.9	50.0	110	140	195	3.90	50.0
5	117	150	210	5.15	56.3	118	148	205	5.15	56.3
10	123	156	221	5.55	58.5	127	156	216	5.50	58.5
20	130	164	229	6.00	60.0	133	161	225	5.90	60.3
30	135	173	234	6.40	62.0	136	168	229	6.30	62.6
40	138	179	242	6.75	63.8	138	177	233	6.56	64.0
50	140	184	249	7.05	65.6	140	181	238	6.82	66.3
60	143	191	254	7.40	69.0	144	186	247	7.10	73.5
70	147	198	264	7.85	73.3	147	193	253	7.45	75.4
80	150	206	275	8.25	75.8	150	198	261	7.85	76.7
90	155	215	287	9.10	79.0	155	205	271	8.80	79.0
95	161	221	305	10.30	81.9	160	215	267	10.00	81.5
100	174	254	349	11.80	90.0	178	254	350	11.90	89.0
	Motor fuels in 1931									
<1[a]	109	208	290	3.90	44.5					
5	120	221	324	5.65	48.4					
10	126	230	335	6.18	51.3					
20	132	240	344	6.84	55.9					
30	136	247	350	7.30	59.9					
40	138	252	354	7.65	63.6					
50	141	256	358	8.00	67.4					
60	143	259	362	8.36	70.7					
70	146	263	366	8.80	72.9					
80	150	267	372	9.30	74.4					
90	156	272	379	10.06	76.6					
95	160	276	385	10.70	78.2					
100	179	295	397	12.40	83.0					

[a] Less than 1 per cent. [b] At 212° F.

It is interesting in this connection to observe the trend of changes in U. S. motor gasoline, the data for which are available in various Bureau of Mines reports. The pertinent data pertaining to the volatility and vapor pressure of U. S. motor gasoline are shown in Fig. 4. The increase in vapor pressure and volatility of motor fuels during the past five years has been due to cracking processes and vapor-recovery systems at refineries, since the percentage of natural gasoline available has not increased since 1927. Moreover, the vapor pressure and volatility of the natural gasoline have been gradually lowered of recent years. Last year there was a material decrease in natural gasoline used by the refiners. This was due to some extent to the increased amounts of natural gasoline that is going directly to the consuming market, a situation which has been brought about by an increased knowledge of the value of the product and the low level of the prices for natural gasoline as compared with motor fuel.

Summary

The four major properties essential to good operation of aircraft engines and in which gasolines may differ greatly are volatility, vapor pressure, gum content, and anti-knock characteristics. Other properties not necessarily less essential but possessed generally by practically all gasolines pertain to absence of entrained water, low sulphur content, freedom from acids and alkalis and non-volatile residues, color, odor, etc.

The testing methods for determining volatility and vapor pressure of gasoline are well standardized, and relation of test results to service conditions are fairly satisfactory. The possibility of securing a fuel with better volatility characteristics which would at the same time improve the anti-knock quality of the gasoline is in the hands of the designers of aircraft and automotive engines. The restriction in volatility characteristics is due to the limitations of some of the fuel-feed systems of the equipment in which the fuel is used.

Much work has been done on determining the gum content of gasoline and an early solution to problems affecting duplicability of test results looks promising. However, much remains to be learned regarding the nature and deleterious effect of gums in gasolines. In the meantime the present low gum contents required by gasoline specifications will safeguard the industry.

A knock-testing method giving results susceptible of reasonably close duplication has been developed by the Cooperative Fuel Research Steering Committee. Trouble has been encountered by several investigators in that the test results are not always truly indicative of service conditions. The tendency is toward testing fuels at higher jacket temperatures and higher speeds, which procedure gives promise of affording a better measure of the real anti-knock value of the fuel.

The properties of aviation gasoline sold in the United States last year showed the following ranges:

Reid vapor pressure, lb	4.0 to 11.9
10 per cent evaporated temperature, °F	110 to 178
50 per cent evaporated temperature, °F	140 to 254
90 per cent evaporated temperature, °F	195 to 360
Octane No. (at 212°F and 600 rpm Waukesha engine)	50.0 to 89.0
Propane and lighter, per cent	Trace
Butanes, per cent	Trace to 4.8
Pentanes, per cent	8.6 to 38.9

By excluding some of the gasolines which may have possessed abnormal characteristics, it is estimated that the properties of

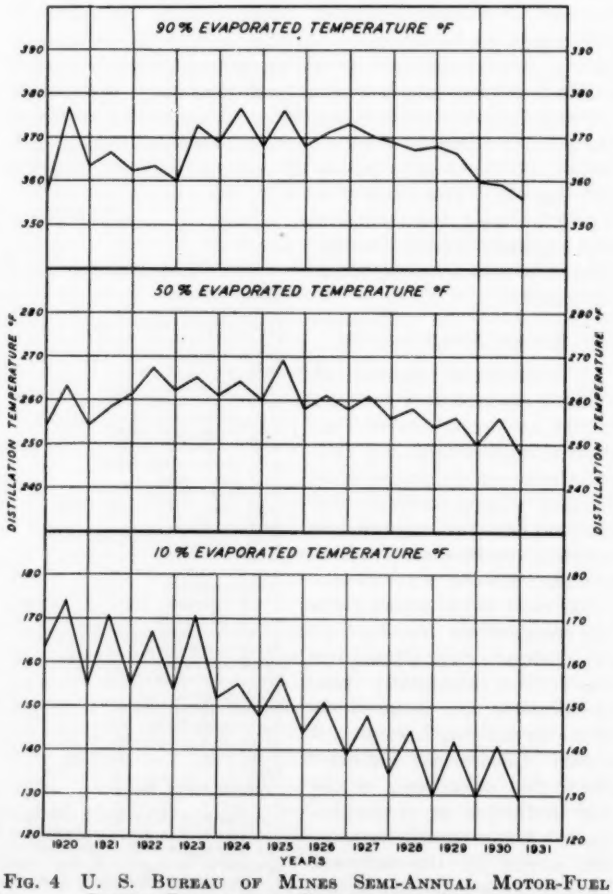


FIG. 4 U. S. BUREAU OF MINES SEMI-ANNUAL MOTOR-FUEL SURVEY DISTILLATION TEMPERATURE AVERAGES

(For 10, 50, and 90 per cent evaporated, January, 1920, to January, 1931, inclusive.)

motor gasoline sold in the United States last year varied through the following ranges:

Reid vapor pressure, lb	3.9 to 12.4
10 per cent evaporated temperature, °F	109 to 179
50 per cent evaporated temperature, °F	208 to 295
90 per cent evaporated temperature, °F	290 to 397
Octane No. (at 212 F and 600 rpm Waukesha engine)	44.5 to 83.0

	Range	Average
Propane and lighter, per cent	Trace to 2.0	0.2
Butanes, per cent	2.0 to 9.0	4.9
Pentanes, per cent	6.0 to 25.0	9.8

The quality of aviation gasoline is a mutual problem of the petroleum and aviation industries. The quality of the best aviation gasolines available in sufficient quantities has been used by the aviation industry as a basis for improvements pertaining to power output per unit of weight and displacement of aircraft engines. It is to be expected that this situation will continue. The aviation industry can render itself a valuable service in connection with the quality of aviation gasoline if future airplanes are designed to handle aviation gasolines with higher vapor pressures.

Any limitation on volatility or vapor pressure not founded upon justifiable operating conditions encountered in actual service results in needless sacrifice of the more important qualities of the gasoline. To prevent the inclusion of pentanes in aviation gasoline and of butanes in motor gasoline by unduly rigid vapor-pressure specifications means not only a sacrifice of the valuable properties of the gasoline, but also necessitates an utter waste of natural resources accompanied by additional expenses for lead or a loss in anti-knock value for the same amount of lead. If the additional volatility and vapor pressure can be safely used, more, better, and lower-priced aviation gasoline will be available from the petroleum industry.

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AER-55-12

Stainless Steel in Aircraft Construction

By E. J. W. RAGSDALE,[1] PHILADELPHIA, PA.

Stainless steel will assume its position in the aircraft industry when and where the urge is strong enough to study its possibilities, to acquire its technique, and to overcome the discouragement of first efforts. The first use of stainless steel in aircraft was in England. Tons of it were used in the construction of the "R-101." In this country a trend toward stainless steel is noted, particularly for air-transport units.

PART I—STAINLESS STEEL



AVIATION, emerging from its formative stages, now faces problems of production. Its methods and materials have to be reevaluated in the light of fabrication and maintenance. Much has already been done, and more will be. The desire for improvement is sufficiently keen to preface its accomplishment. A general lack of prejudice or convention on the part of aeronautic engineers seems to assure it.

Methods are inherently bound to materials, and must be considered as a part of them. Aviation has had three materials, and it still has them. Nor is the choice an easy one, for spruce, duralumin, and high-tensile steel all equate in the formulas for structural value. The ratios of weight, strength, and modulus of elasticity leave little margin for preference. Decision must rest on a different consideration, and that consideration is the fabrication of a serviceable article.

Strangely enough, it was serviceability that first prompted the use of stainless steel for aircraft—and that at a time when structural permanence was the least of aviation's worries. A recurring interest in the material comes simultaneously with a growing appreciation of its need. Study of its possibilities then reveals unsuspected structural qualities—qualities which would be self-sufficient even without the non-corrosive feature.

The term "stainless steel" is in a way unfortunate, for its behavior is no more like that of steel than is bronze like that of copper, and its proportion of alloyed material is certainly as great.

Again "stainless" covers a wide range of composition and property; but, fortunately for the purpose of this paper, the issue can be confined to that most common alloy known as 18-8. The others seem not so likely to find wide application in structures. The 18-8 has, roughly, 18 per cent chromium, 8 per cent nickel, and a little less than 0.18 per cent carbon. It is further known by a score of trade names, such as Allegheny Metal, Nirosta, KA2, Resistal, and Enduro.

Its chief claim to structural use lies in the capacity for retaining toughness after having been cold worked to high tensile values. Tensiles of over 400,000 lb per sq in. have been known. The material, however, is not tractable above half that figure, and will give difficulty in fabrication.

In stress-strain relation, the curve of cold-worked 18-8 approximates that of cold-drawn wire. Yield point and elastic limit are poorly defined. The modulus is about 29,000,000.

Here, then, is a material which is possessed of properties superior to those of chrome-molybdenum steel and one which can be fabricated without heat treatment. In addition, it is non-corrosive and non-magnetic.

That this material requires a special technique in handling and that it may lose some or all of its virtues under abusage is true, but what material does not? Even ordinary steel requires some understanding of its behavior. Tool steel requires a definite and specific knowledge. So one can hardly resent the fact that the use of stainless involves close study of the instructions regarding it. This is particularly true in view of the widespread misconception of its nature. A few fundamentals will explain many an apparent vagary.

In the first place, 18-8 is not inherently stainless, thanks to its composition. It is stainless only when in a certain physical state; this state is known as austenite. Austenite is that condition wherein the carbides remain in solid solution with the iron. It obtains in plain carbon steels between the temperatures of solidification and dull-red heat. It could be preserved down to normal temperatures were an immediate and severe quenching possible, but as a matter of practice the carbides are precipitated out at the critical temperature around 1200 F. They become grain boundaries, and since the steel is now no longer homogeneous, but is a mixture of materials possessed of different electrolytic potential, a marked change of character results.

Austenite is non-magnetic, malleable, and susceptible to extreme cold working. There is a strong semblance, in fact, to gold, lead, or copper; its preservation seems desirable, and in 18-8 such has been accomplished. Chromium and nickel hold the carbides in solution during a reasonably short quench. In slow cooling or reheating the influence is not sufficient, and carbides of iron and chromium will precipitate. They cannot be reabsorbed without reheating to some 2000 F. The temperature range of precipitation is from 1200 to 1600 F. In all heat operations on 18-8, consideration must therefore be given to the time element, the critical range, and the corrective temperature. No complete change from or to austenite can be effected without regard to time.

The outward effect of excluded carbides is that the material becomes magnetic and is less resistant to corrosion. It is also less resistant to fatigue—a condition which is more alarming than the mere appearance of rust. Many a welded seam has eventually shown cracks at some distance from the weld, and the cause is unmistakable, though corrosion may or may not have been evident.

It stands to reason that a material having a homogeneous structure will not display the same characteristics as one composed of crystals of that material with interlarded boundaries and nuggets of carbides.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

As indicated, 18-8 can be given this condition by heating to from 1200 to 1600 F and cooling. If then it is boiled in Strauss's solution (10% $CaSO_4$, 10% H_2SO_4 + H_2O), the carbides will be dissolved out, leaving a loose mesh of metal which breaks readily in the fingers and has no metallic sound, but is again magnetic. The experiment has practical value only in determining deleterious effects due to welding or other heat operations.

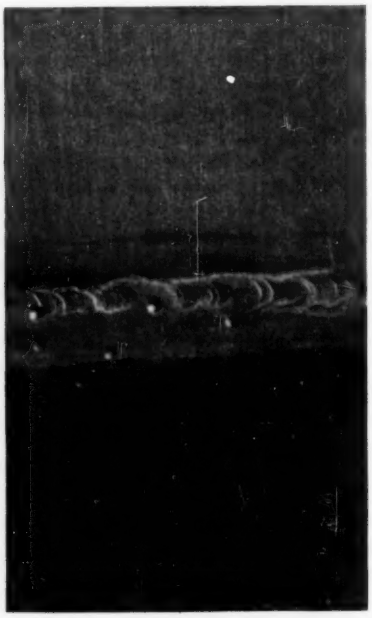


FIG. 1 GAS WELD BOILED IN STRAUSS'S SOLUTION AND SLIGHTLY BENT

(Two cracks immediately developed parallel to the weld, showing a severe and highly localized precipitation of carbides.)

It would therefore seem that 18-8 cannot be welded by any conventional method (gas, arc, or electric resistance) without producing localized precipitation of carbides. Such is true; but just how serious this is depends on a number of factors—some controllable, others not so.

Certain it is that the condition can be entirely remedied by heating the piece to some 2000 F and then air quenching. The carbides will be reabsorbed, and a homogeneous stainless structure restored.

Unfortunately, the reheating of a fabricated part is not always feasible. More unfortunate still is the fact that all beneficial effect of cold work is lost. However, 18-8 has a wide application of uses outside of structures, and welding by gas or arc has been successfully done. Some particularly good work is done with the atomic hydrogen method. A skilled welder can control his heat so that the precipitation of carbides is minimized. However, the technique required for gas or arc welding of stainless differs considerably from that which suffices for plain carbon steel.

Better results have been had with electric spot welding. The conditions favor it. In the first place, the surfaces are clean and free of oxides; secondly, the material resistance is from 8 to 14 times that of carbon steel. Both conditions make for lower current value, so that commercial welders must be regulated down below normal in order to avoid burning. The welded spot has heretofore usually been surrounded by an annular ring of precipitated carbides, but these cannot be positively detected without the aid of a microscope after sectionalizing and etching with hydrofluoric acid. This ring has been termed the "corona." Its extent is known to be a function of time and current; its existence cannot be recognized by the eye. In fact, it is axiomatic that no one can tell a good or a bad weld by inspection. Surface condition means absolutely nothing, for fusion rarely extends to the outside of the sheets. The visible marking, more often than not, is due to arcing at the electrodes or secondary heat influence. However, all argument and evidence to the contrary, welds continue to be judged at face value.

Appreciation of these conditions has led the Edward G. Budd Manufacturing Company, of Philadelphia, to develop a spot-welding machine the outstanding features of which are absolute uniformity of current supply, use of high currents in a very brief time element, and a recorder which supplies visible and audible record of performance. Results seem to justify differentiation, and the term "shot weld" has been coined.

With use of the shot welder, corona-free welds can be consistently made, which automatically opens up a large structural field to stainless steel. Without such, there is inevitably that uncertainty which has always been associated with uncontrolled welding.

The first use of shot-welded stainless steel was in connection with aircraft, which at the same time was conceived as its severest test. Development, extending over years, has been gradual and cautious. Not until last year did an entire airplane result, though many sub-assemblies had previously been fabricated. This plane, an amphibian patterned after Savoia-Marchetti design, has demonstrated its worth in test. The type seemed to offer

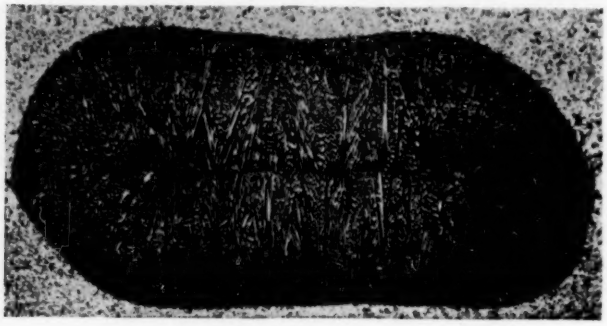


FIG. 2 OVERETCHED SECTION OF A SHOT WELD IN TWO SHEETS OF 0.030 IN. THICKNESS

(No visual inspection from the outside could tell the extent or condition of the fused material. Carbide precipitation, if any, would be found at least a weld diameter to either side of the limit of fusion and would require higher magnification.)

to stainless its ultimate utility, both as to structural value and as to corrosion resistance. Success of the venture, coming simultaneously with an increased general interest in stainless steels, presages wide application and makes justifiable a description of the structural methods employed.

PART II—THE USE OF 18-8 IN AIRCRAFT STRUCTURES

DESIGN

The design philosophy developed primarily in England for thin, heat-treated steel structures offers guidance, but cannot be too closely followed. In the first place, if high-tensile stainless be used, "springback" makes difficult the forming of sinuous curves and convolutions, such as characterize most British practice. Stainless is best formed by braking operations. Sections are apt to consist of a series of flats, sharp angles, V-beads, and flanges. Thinness being requisite, closed sections predominate in all compression members. Crinkle tests must be frequently made to determine minimum P/A, and values under 100,000 lb

per sq in. are not found to be economical. The problem of attachment to other members or fittings influences the eventual shape.

Steel structures should show an equal, if not superior, weight-strength relation to spruce or duralumin. Skilful design can do it, but the tricks learned from one material are not translatable to the other two.

Here, as elsewhere, deflection usually governs the structural design. While the modulus of stainless is some 29,000,000, this value falls off under high stress. Furthermore, if the section is built up of "flats" not properly proportioned to thickness, a waviness results and deflection becomes greater than that proportioned to the load. The capacity of 18-8 to recover itself after deformation, or even apparent failure, is phenomenal.

The 18-8 Alloy

As 18-8 differs from other alloys of the non-corrosive group, so does it differ in itself under varying conditions of cold work. When quenched from 2000 F, it assumes its most ductile and soft state. The tensile is from 85,000 to 90,000 lb per sq in., and the elongation is some 65 per cent in 2 in., but the yield point is only 38,000.

What is most important from a welding point of view is that the electric resistance is only eight times that of carbon steel and it is susceptible to corona formation. Its use in highly stressed structures is therefore limited to fittings.

On the other hand, when cold drawn or rolled to a tensile of 200,000 lb per sq in., the yield point approaches the ultimate, elongation drops to some 8 per cent, but the electric resistance becomes about 12 times that of low-carbon steel. It responds slightly to the magnet. It will not tear along the grain as will spring steel, and it displays a toughness seldom associated with material of this tensile. The resistance to corrosion, though still very high, has been lessened, while resistance to fatigue has been materially increased. Physical properties across the direction of rolling are slightly higher than those established in the direction.

This material is available in wire or strip form—the latter from 0.005 to 0.125 in. ($^1/_8$ in.) in thickness and seldom over 12 in. in width.

Then, when cold working progresses beyond 200,000 lb per sq in., utility becomes limited. The material responds even more to the magnet, and corrosion resistance falls off markedly.

Chief interest lies in avoidance of excessive working by fabrication, such as might produce these effects. Angles too sharply bent are the commonest offender. They should be examined for incipient cracks or coarse granulation of outer edges. A good test is to reflatten a bent-up section to see whether a crack develops.

Specifications

The stocks commonly carried in stainless steels are not of a gage or quality suited to aircraft use. They are confined to sheets from 0.025 to $^1/_8$ in. in gage and of soft anneal. Several mills, however, are prepared to execute orders for strip to meet apparently simple but very exacting specifications. One specification is:

Gages, 0.005, 0.010, 0.0125, 0.016 in.
Tolerances, plus or minus 0.0005 in. in gage, plus or minus 0.010 in. in width
Widths, up to 14 in.
Tensile strength, 180,000 to 200,000
Elongation in 2 in., 8 to 6 per cent
The material shall bend dead flat upon itself in a vise, both across or along the grain, without developing an open crack.

Another specification is:

Gages, 0.024, 0.030, 0.035, 0.040 in.
Tolerances, plus or minus 0.001 in. in gage, plus or minus 0.020 in. in width
Widths, up to 9 in.
Tensile strength, 180,000 to 160,000 (diminishing with thickness)
Elongation in 2 in., 8 to 12 per cent
The material shall bend at right angles in a brake with an inside radius equal to that of its thickness and shall show no incipient cracks. This is to be done along and across the grain. It shall then be flattened out without developing an open crack.



Fig. 3 Spar Fitting, Showing Compression Strut
(With attachment points for wing strut and flying wire.)

The mill practice to meet these specifications is not an easy one. Operators, too, have found that 18-8 does not follow steel conventions. The secret of success may lie in roll temperatures, for 18-8 does some funny things when worked between —100 and +100 F.

Wherever soft material is required for sheets or strip, it is found expedient to have it come with annealed and pickled finish. This contrasts with polished high-tensile stock, and confusion is not apt to result.

Forming and Machining

High-tensile stock is not readily formed in hand or power brakes. It slides too readily in the grips. Short lengths, however, can be done, but attention must be paid to springback. This is easily predetermined by braking sample strips beyond the required angle and observing the degree of return.

Fundamentally, stainless construction depends upon rolled sections, and the closest attention must be given to this department. Drawbench work has not proved satisfactory; the dies give trouble with hard material and the lubricant causes trouble in subsequent assembly by shot welding. Driven rolls have been but sparingly used. The best success has been had by pulling the strip through a series of forming rolls. Four pairs are sufficient to form any usual section. Speed and tension vary with the complexity of the formation.

Cutting of strip is done by shear, but knife edges must be maintained sharp and in close contact. The strip will show a tendency to crowd over as the shearing blade is depressed if a guide rail is not used. Sections are cut on a high-speed bandsaw having fine or blunted teeth.

Punching can be done, but is hard on dies. Drilling is more difficult, since the material hardens so quickly under the point. Great pressure, slow speed, and a sharp drill are required. The hole is apt to be ragged.

Several of the mills offer a variety of stainless steels which lend themselves better to machine operations; none are really good, being only preferable to 18-8.

Any forging or forming under heat destroys all value incident to cold work. The piece should be rectified by heating to

2000 F and air quenching. It will have 90,000 lb tensile—a fact which is apt to be misleading if one does not reflect that the elastic limit is only 38,000 lb.

Joining

The problem of joining materials, though as old as craftsmanship, has never been completely solved. It still commands our best effort.

Low-tensile stainless, especially where heat effects can be disregarded or rectified, may be joined by fusion (gas or arc) welding, electric spot or flash welding, and brazing. Shot welding or soldering do not require corrective treatment.

The high-tensile material finds a cheap and satisfactory solution in shot welding; a ready but less satisfactory one in soldering.

Riveting is not considered for normal structural work. The holes are difficult to make, stainless rivets harden too fast, and hot riveting is not yet indicated.

Folded seams, where usable, are satisfactory, and if spotted by welds every inch or so, resist considerable pull without opening up. For instance, a 1-in. test strip folded in the center opened at 370 lb, while the same seam held 1200 lb after having had one shot weld put through it.

In fusion welding by gas, arc, or atomic hydrogen, satisfactory results are to be had, once a technique is acquired. No attempt will be made here, however, to duplicate the instructions which are available from any of the sources of material supply. The warning against corona effect is repeated, and rectifying treatment is urged. In case a doubt exists, simply boil a specimen weld in Strauss's solution and bend. If no crack develops, all is well and good.

Soldering is a simple, easy, and fairly effective method of joining. No particular technique is involved. The surfaces should be cleaned and washed with a 10 per cent solution of phosphoric acid. No flux is needed. The soldered joint must then be washed free of any remaining acid.

Shot Welding

In shot or spot welding, the diameter of the fused slug should have some reasonable relation to the thickness of the sheets to be joined. No closed section can economically suffer large welding flanges, for it is here that compression failure initiates. Shot welding encourages use of a narrow flange—much narrower than would obtain in a riveted connection—and in consequence good practice dictates that a spot no greater in diameter than three times the double thickness of material should be used. The flange, in turn, should be at least three spot diameters in width.

It will be appreciated that a large corona effect would reach the edge of the flange, where, as experience shows, all fatigue failures to start. A corona-free spot is essential, and such is obtained by

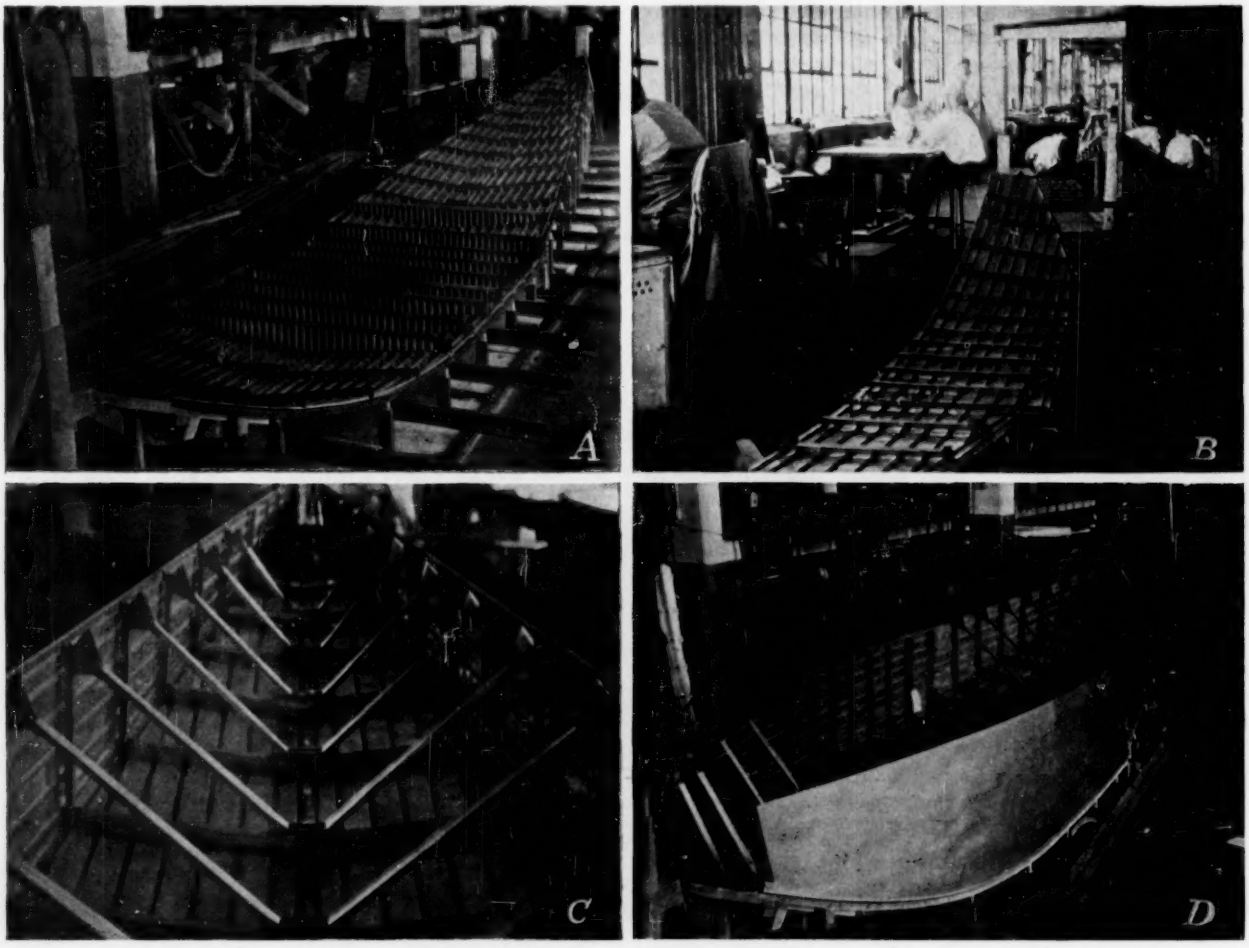


Fig. 4 Hull Assembly of an Amphibian

(*A* is the bottom, completely framed and sheathed as it lies in assembly jig. *B* is one side ready for assembly. *C* is a rear view of assembled sides and bottom before deck beams have been attached. *D* is a view of hull just after removal of side jigs.)

short time of current dwell. There is really no need for large areas of fusion, since good welds will show twice the shear of a rivet of like diameter and can be spaced 50 per cent closer together.

Shear is not the only test for welds; they must be tough—tough enough to permit rotation through at least 60 deg. Again, current and time relation will determine the character of the fused slug.

Probably the one outstanding essential of a shot welder designed for aircraft or other structural use is portability. It is rarely practical to progress the work past the apparatus. In the

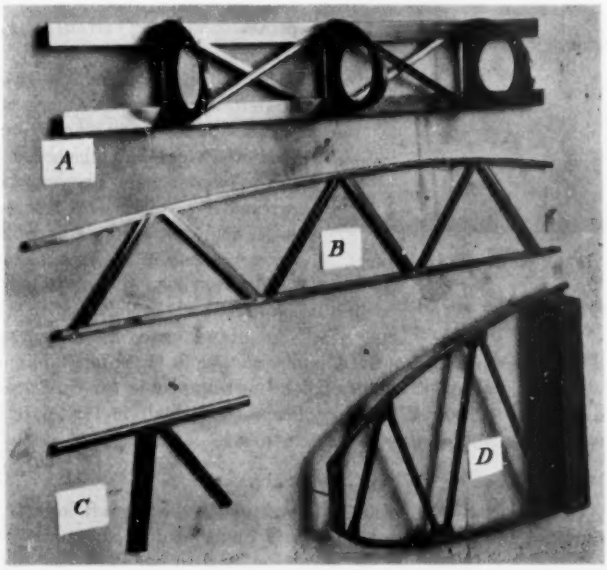


FIG. 5 FOUR FORMS OF CONSTRUCTION

(*A*, section of aileron spar before nose is covered with 0.010 in. of stainless steel. *B*, simple rib-type construction, with channel caps and Z-strip truss members. *C*, another rib-type construction, showing a compression and a tension shear member attached to cap strip. All are channel sections; the cap being of heavier gage and slightly wider. *D*, a rib nose using double gusset construction.)

Budd machine, long lines of cables are attached to hand or air tongs, and these are dragged into almost inaccessible places. The machine can be rolled about the floor or lifted by cranes to any desired position, but the current loss in cables and tongs is substantial and must be compensated for.

Since the current and dwell are established by the machine, the operator has no choice in these matters. The inspector usually makes the settings for test welds, and then adjusts the recorder to the new value. Any variance of conditions which may make for over or insufficient current-time relation causes a bell to ring continuously until the machine is reset. Every weld made or failed is recorded on a tape, the length of line being a function of the I^2Rt, where I represents amperes, R the resistance, and t the time of dwell. This is the total heating value of the electric energy supplied. It is interesting to note that the heat which actually goes into a single shot weld in two pieces of 0.012-in. material is but 0.035 Btu.

This also explains how assemblies can be made without that distortion so troublesome in gas welding.

While normal practice is to use two electrodes (one directly opposite the other, and both of similar shape), unusual conditions often dictate welding by indirection. One electrode may consist of a pad clamped to a sheet, while the other is conventionally pointed and held against the member to be welded to that sheet. Again, both electrodes may be pointed and held against the member, some distance apart, while a copper plate rests beneath the sheet. Here, two welds are made simultaneously, the current passing from one point through the member and sheet into the copper plate, then along the copper plate until it comes opposite to the other point where it exists through sheet and member. This is made possible by virtue of the superior conductivity of the plate over that of the combined sheet and member of steel.

Both methods find application in shot welding a corrugated member to a flat sheet such as is used in flooring or partitions. The sheet rests on a large copper plate, and the only care is that electrodes are held far enough apart to insure sufficient resistance in the steel pieces.

Any of the many forms of indirect welding are apt to be more uncertain than in conventional practice. Both care and control must be exercised through the operation. Even then, good design will obviate the condition where the structure is important.

Welding through several thicknesses is possible, and is frequently done, nor do the pieces have to be of equal gage.

ASSEMBLIES

The accompanying photographs must tell their own story of structural methods. This paper could not possibly picture the result of years of thought in devising ways and means, nor

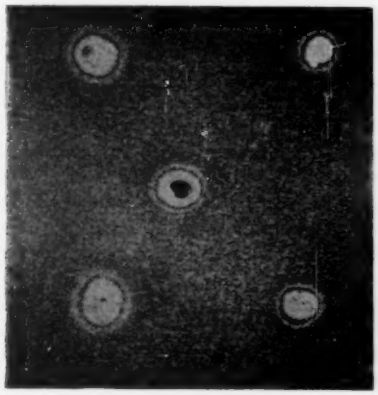


FIG. 6 FIVE SPOT WELDS VARYING IN TIME OF CURRENT

(The time varies from 1/4 to 1/2 sec. Polishing and etching developed the corona ring.)

should it attempt to guide others whose ingenuity will devise new ones. Suffice it to say that the amphibian shown incorporates type structures for every department of aircraft design.

Much novelty is to be found in the hull, where a thin material usually stands at greatest disadvantage. Here the thickest skin sheet is 0.016 in.; the thinnest 0.010 in. In order to prevent billowing or dents, the entire skin is carried by longitudinally running channels, and these attach to the frames and bulks. No cross-member touches the shell. As a consequence, all distortion results in longitudinal waves between fore-and-aft members, and these disappear upon release of pressure. The washboard effect between ribs, which is so characteristic of thinly sheathed hulls, has been successfully avoided. Again, since the shell is "hung," so to speak, it flexes under pressure and is strangely quiet.

Another feature peculiar to this design is that the bottom and sides were completely fabricated before assembly, and then swung into position, without removal from their respective jigs. Side jigs were made to dowel and bolt to the bottom jig. Assembly consisted of flanging over a lap joint at the chine and securing the side and bottom frame pieces by knee gussets. Deck frames were then thrown across and the deck sheathing attached. By

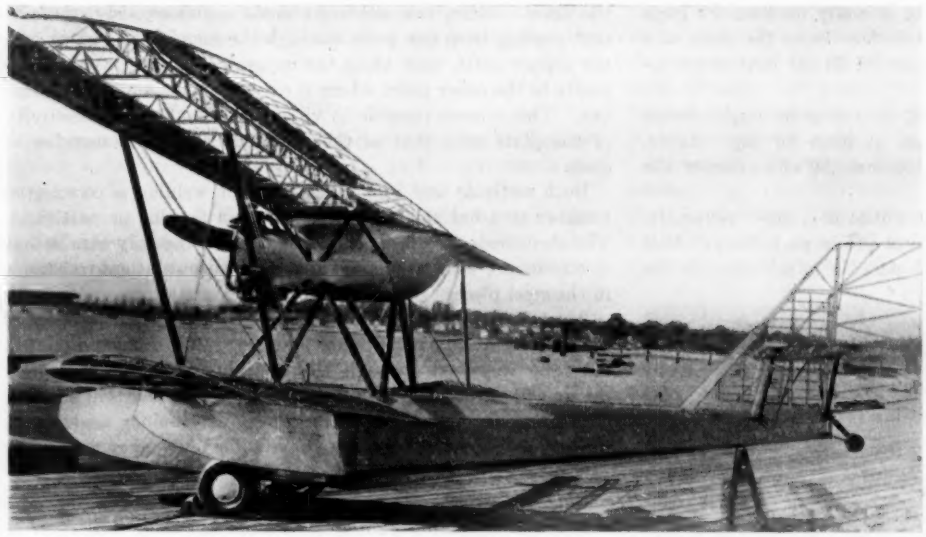


FIG. 7 THIS AMPHIBIAN IS THE FIRST COMPLETE STRUCTURE TO BE MADE OF STAINLESS STEEL AND ASSEMBLED BY SHOT WELDING
(The resulting weight, strength, performance, and quietness showed improvement over the equivalent aircraft made of wood.)

this method a minimum of inside work resulted. Work in the open is always done more expeditiously.

In this and other experimental work, it has been found that expense runs no higher than in the more common forms of aircraft structure. After all, the cost of such work can be attributed more to indecision than to any other one item.

Shot welding, however, makes the use of stainless attractive in experimental work. It is so easily and quickly done, it involves no elaborate jigs, and local reinforcing becomes simple.

For small structures, the only jig required consists of a number of wooden blocks nailed over the blueprint. For more elaborate ones, a wooden frame is erected, and significant points are notched out. The component strips which go to make up a closed section are, in themselves, flexible and can be naturally bent to even sharp curvature. They are then welded together and attain the rigidity peculiar to a closed member.

While it is futile to discuss weights of exact structures made up in different materials, since each has its own design tendency and limitation, it can be safely said that no complete design need suffer in weight or strength owing to its fabrication in stainless steel. In fact, the amphibian in question was lighter and stronger than its equivalent in wood. Had the craft been larger, an even wider discrepancy would have obtained.

COST

However tender the subject, cost cannot be ignored. The 18-8 costs in fair poundage from 40 to 60 cents a pound, depending upon gage, width, and specification. It comes in coils. Scrap lossage is very little where design cooperates. A low figure is 5 per cent; 20 per cent is high.

Draw rolling of sections is cheap—much cheaper than braking.

Shot welding is perhaps the quickest and cheapest form of attachment. One operator can average 10 welds a minute on complicated work and 40 on fairly straight seams. Machine welding has no apparent limit. Using only one pair of points, 100 welds per minute have been consistently done.

Stainless steel, however, is not a cheap way out; it is a good way. Where maintenance can be considered, it more than justifies itself. The 18-8 is a material of permanence. Shot welding is a method of rigid attachment which will not work loose with time.

Up to now, the chiefest factor of cost is engineering. Such is always the case in a pioneer development.

CORROSION

Talk of corrosion comes last, where it rightfully belongs. The 18-8 does not naturally corrode, nor does it materially age harden or show signs of instability. Certain chemicals will attack it slightly; certain abuses will render it corrodible.

When shot welded, it will show in the salt spray a pigmentation due to the rusting of iron particles which have flashed out of the fused zone. This can be arrested by washing in diluted nitric acid, but need really cause no apprehension. Certain shot welds have been in a salt immersion test for 14 months with no sign of weakened effectiveness. The pigmentation, however, is disquieting to those not familiar with the process or condition, and is responsible for continual discussion of corrodibility—a discussion which is fortunately academic rather than of serious moment.

CONCLUSIONS

Stainless steel will assume its position in the aircraft industry only when and where the urge is great enough to study its possibilities, to acquire its technique, and to overcome the discouragement of first efforts. Such an urge has not been uncommon in aviation. In fact, its brief history has been a continual search for better materials. The Wright brothers, having demonstrated a theory with a device of wooden sticks and iron telephone wires, next perfected their technique by the use of spruce and piano wire. Then with fabricated wooden sections, aircraft structures rested for a while; the material certainly was as reliable as the calculations by which it was stressed. When science stepped in to guide ingenuity and courage, the search for materials was resumed. Steel tubing offered that ideal which the formulas required, but the joining of such was a problem for the mechanic rather than the professor. Gas or arc welding, somehow, did not seem consistent with the ideal. Heat warped the structure, and the weld depended on the skill of the operator. Production dare not depend upon personal skill. Then, too, corrosion raised a question.

In the meantime, lighter-than-air machines had developed duralumin to commanding position. It answered the urge for all-metal planes, monocoque construction, and hulls. The relatively thick plates made skeleton structures simple. Heat treatment had the earmarks of magic, for the metal could be worked soft and would harden after assembly. But, again, corrosion raises its head, and this time with the added discordant note of fatigue failure.

England turned to heat-treated steel and fabricated structural members. Then, of late, one hears again of stainless; and let it be remembered that the first use of this material in aircraft was over there. They used it; they spot welded it; they built tons of it into that ill-fated *R-101*. But for some reason the development was never carried to its logical conclusion. Only now does one hear of resumed interest and a desire to learn the technique which may have been lacking before. Certainly, the urge is there.

In this country we note a trend, and it is not toward the air flivver with all that that implies, but toward transport units. With it comes added interest of governmental agencies, and that interest will spell "quality." It will inquire into conditions which may obtain in structures, not today, but five years or more from today, and that means "permanence." Then, cost of initial production can be balanced against service and maintenance, and that spells the end of long-enduring stick-and-wire construction. Then, the urge toward proper materials will not have to be stimulated.

Whether or not that urge will be toward stainless remains for time to tell. But the author of this paper, enjoying the prejudice which has been his through years of experimentation, believes it will.

AER-55-13

Recent Developments in Grading Fuels for Aeronautical Use

By W. W. WHITE,[1] NEW YORK, N. Y.

The subject of fuel grading is somewhat unknown to engine constructors, airplane operators, and others. In dealing with the grading of aviation gasoline, the author, instead of drawing on the great mass of experimental and published data on the subject, with the complexities of each of the component parts, and preparing an exhaustive discussion of one of the components, has chosen to present a general discussion of the factors involved. To this the author has added his opinion of the probable trend in the near future. Some of the points raised are to a degree controversial. The paper also includes a review of recent flight tests with various fuels.



IT HAS BEEN generally accepted by engineers, as a result of thousands of tests conducted in this country and abroad, that the tendency of a fuel to detonate in an engine is by far the outstanding characteristic of importance. Probably the next most important item is the shape of the volatility curve. Aviation gasoline must contain sufficient proportions of low-boiling (highly volatile) constituents to permit the easy starting of a cold engine. These constituents must not be present in excessive quantities, as this encourages any tendency to premature vaporization in carburetors or fuel lines, with resulting vapor lock. Likewise the presence of an undue proportion of high-boiling constituents is undesirable, as this may cause irregular distribution and excessive heat in those cylinders which fail to receive their proper proportion of fuel.

The gasoline should have a reasonably high volatility in order to insure good flexibility or response to throttle variations when the engine is below usual operating temperatures. Volatility is usually determined by means of a distillation test made under carefully specified conditions. The interpretation of results is best illustrated by the example of the United States Army's specifications for aviation gasoline, fighting grade. These specify maximum temperature limits for the 10 per cent point in the distillation range, the 50 per cent point, the 90 per cent evaporated point, and the maximum allowable distillation loss, specified as 2 per cent. The maximum limit for the 10 per cent point insures that the gasoline is sufficiently volatile at low temperatures to assure easy starting. Since "gas lock" may be encountered if the fuel is too volatile, the volatility at low temperatures is further controlled by specifying a maximum allowable vapor pressure. The limits on 90 per cent point insure against tendency toward poor distribution and excessive oil dilution. The limit on the 50 per cent point, in conjunction with the other limits, insures the desired average volatility. It will be noticed that no mention is made in the foregoing of either the initial distillation temperature or the end point. Conclusive tests, covering both the aeronautical and the automotive fields, have pointed to the conclusion that neither has any significance when related to engine performance. (See "The Meaning of the Gasoline Distillation Curve," by Graham Edgar, J. B. Hill, and T. A. Boyd, A.P.I. paper presented at Chicago, Ill., Nov. 12, 1930.) The end point does serve, however, as an excellent indicator of possible contamination, especially when viewed in conjunction with the distillation loss.

A short discussion of the other items governing present-day specifications for aviation gasolines or appearing in analyses of these fuels may be in order as matters of general knowledge, tabulated for ready reference:

Color. A comparative test performed usually with the Saybolt chromometer. A figure of +25 indicates that the product is water white and colorless. Successively lower numbers indicate increasing color content.

Doctor. Should be "negative" or "sweet."

Odor. Self-explanatory; a fuel may be characterized as having a coal-tar, aromatic, cracked, naphthenic, or paraffinic odor, etc.

Sulphur. Expressed in percentage of sulphur content by weight. It should under no circumstances exceed a value of 0.1 per cent.

Acid Heat. Of primary significance as an indicator of the manufacturing method employed. A very low number indicates a straight-run product, a high number that it is a product of cracking.

Acidity. Self-explanatory. The fuel should show a neutral reaction.

Corrosion. Determined by evaporating 100 cu cm of fuel in a polished copper dish, after which no gray or black corrosion of the copper should result.

Gum. Usually determined in a copper dish, as in determining corrosion. Weight of residue should not exceed 3 mg.

Accelerated Oxidation. Determined by subjecting gasoline to accelerated aging at 212 F in presence of oxygen at 100 lb pressure for 4 hr. Subsequent evaporization of 100 ml of oxidized sample should yield residue not to exceed 10 mg—an indi-

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

cator of the stability of the gasoline in storage and in service.

Breakdown Period. Used as substitute for, or supplementary to, accelerated oxidation test. Technique is similar in general to the latter, except that test is continued until gasoline becomes unstable. A period of 3 hr without breakdown is equivalent to at least one year's storage. This test is thought by many to be more severe than the accelerated oxidation test.

Reid Vapor Pressure. Determined by heating a small bomb containing the sample to 100 F. It is generally conceded that the pressure rise should not exceed about 7.5 lb to insure safety from vapor lock in all installations. This stipulation reacts to limit the 10 per cent point in the distillation range, as the highly volatile fractions are those which raise the vapor pressure.

Freezing Point. Self-explanatory. To insure against clogging of jets and icing in manifolds, this figure should be below —76 F. Gasolines containing 20 per cent or more of benzol seldom meet this requirement.

Gravity. Expressed either in A.P.I. units or as specific gravity. It is as valueless in the determination of desirable qualities or the constituents of gasoline as using the total weight of a pile of dirt as a measure of the proportion of sand, gravel, or stones that the pile contains. The Government does not in a single instance consider this item in its specifications.

Octane Number. The great variations in anti-detonating characteristics which have been noted in gasolines which are perfectly satisfactory for aeronautical use in all other respects have been responsible for a vast amount of research and test work. Oil companies, engine builders, governmental agencies, and technical societies have labored, both individually and co-operatively, to arrive at a basis for measurement of this all-important factor which can be applied directly as a criterion of the adaptability of the fuel to engines in actual operation. A direct result of these intensive investigations has been the adoption of the present American method for determining and expressing anti-knock properties. This practice, which is receiving international recognition, and which it is hoped will shortly be adopted as a universal anti-knock yardstick, is to compare the fuel in question with a basic chemically pure fuel. This basic standard is a mixture of iso-octane (2-2-4 tri-methyl pentane) and normal heptane, two hydrocarbons which can be obtained in a chemically pure state. Iso-octane has exceptionally good anti-detonating properties, well above all classes of current clear gasolines. Normal heptane possesses pro-knock characteristics to an equally marked degree. By mixing these two substances in proper proportion, a fuel of any desired knock characteristics may be obtained, against which a given sample of gasoline may be compared. The anti-knock quality of the fuel may then be expressed in terms of "octane numbers," the number designating the percentage of iso-octane in the octane-heptane mixture which corresponds in knocking characteristics to the fuel sample under test. Since the octane is the knock suppressor in the mixture, it follows that the higher the octane number of a gasoline, the better its anti-knock qualities.

After having established the basic comparison fuel, it has remained to define and perfect the test conditions under which the fuels are to be compared. These conditions, if the ideal is to be realized, must predict accurately the performance of the fuel in a full-scale engine, under the most strenuous conditions of operation. At the same time it is necessary that they be readily and inexpensively realized by "run of the mine" laboratory personnel, and that the results of tests be reproducible by widely separated agencies.

The vast amount of cooperative experimentation which has been expended upon the problem has produced results which are extremely encouraging. Proceeding from the groundwork established through developing the technique of rating automotive fuels, it is now usual for several separate laboratories to agree on the octane rating of a particular aeronautical fuel to within one octane number, under the current commercial method in vogue. Briefly, this method employs an Ethyl Gasoline Corporation series 30 single-cylinder test engine, operated at a jacket temperature of 300 F and at 600 rpm, with 15 deg spark advance. Substitution of the Cooperative Fuel Research Committee's variable compression engine, operated under similar conditions, gives results of comparable accuracy. The detailed conduct of these tests has been widely discussed before the Society and elsewhere, and may be referred to by those interested.

The correlation of data obtained on the anti-detonating characteristics of fuels in the laboratory, with actual full-scale performance in aeronautical engines, still remains a problem of the first magnitude. It is relatively simple to determine, for a gasoline from a selected crude and produced under a particular set of manufacturing specifications, the octane number required by any engine or series of engines for satisfactory operation. An excellent example of this type of correlative work is the fuel recommendation chart prepared by engineers of the Stanavo Specification Board, Inc.

The situation is enormously complicated, however, when we consider that gasolines from various crude sources, or of varying methods of refining and manufacture, or variously blended with lead tetra-ethyl, benzol, or other anti-detonant, refuse to cooperate with the investigator. He finds that a series of gasolines of diversified antecedents which have identical octane numbers when tested, say, at 300 F jacket temperature, 600 rpm, and 15 deg spark advance, fail by a large margin of being identical in octane rating if the test conditions are changed to 375 F jacket temperature and 900 rpm. Moreover, he discovers that these gasolines, which have so neatly catalogued themselves in the same anti-knock bracket in the laboratory under the 300 F, 600 rpm conditions, fail just as dismally to perform comparably in a full-scale test in an engine which is supposedly critical at that octane number. Horsepower, cylinder temperatures, smoothness, and fuel economy vary in a manner which is inexplicable, until the investigator notes that those fuels whose octane numbers have been reduced the least by the more severe 375 F, 900 rpm method of test, are the products which are in the same degree satisfactory to the engine.

A sequence of events such as the foregoing was noted by technical experts at the Matériel Division, Army Air Corps, and sufficient corroborative data were amassed to warrant the establishment of an amendment to its specification Y-3557-D. This amendment requires the determination of octane numbers in the series 30 Ethyl Gasoline Corporation engine, at 900 rpm, 375 F jacket temperature, and 15 deg spark advance. Mr. S. D. Heron has described the preliminary investigations which led to the adoption of this amendment in a paper delivered before the S.A.E. Aeronautic Session, Chicago, August 26 to 28, 1930, entitled, "Fuel Requirements of the Gasoline Aircraft Engine."

The Naval Bureau of Aeronautics has likewise signified its acceptance of the fact that these more stringent test conditions are better representative of full-scale engine requirements by insisting on similar speed and jacket temperature conditions. Its schedule 7567, domestic aviation gasoline, includes these stipulations in its specifications.

The Detonation Sub-Committee of the Cooperative Fuel Research Steering Committee has realized the importance of this problem, with particular reference to automotive engines. This committee is sponsoring investigations which are similar in purpose to the cooperative test at the General Motors proving ground, carried out by the General Motors Corporation, the

TABLE 1

Sample symbol	Crude source	Manufacturing method	Blending	Octane ratings 300°, 600 r.p.m.	Octane ratings 375°, 900 r.p.m.	Specification[a]	Remarks
A	California	St. run	None	67.5	63.3		Standard reference fuel B-2[b]
B	Columbian	St. run	None	70.0	66.7	D. A. G.	Limited commercial availability
C	Select Coastal	St. run	App. 5% benzol	73.6	70.3	F. G.	Available commercially
D	Select Smackover	St. run	None	74.0	70.6	F. G.	Available commercially
E	Select Mid-Continent	St. run	App. 2 cc t.-e. lead	75.5	74.0	F. G.	Available commercially
F	California	St. run	None	73.8	70.4	F. G.	Experimental blend
G	Coastal	Highly cracked	None	72.3	60.3	D. A. G.	Available commercially
H	Special Smackover	St. run	None	76.3	75.5	F. G.	Special experimental product
J	Mid-Continent	Cracked	None	67.5	56.6	D. A. G.	Limited availability
K	Select Smackover	St. run	0.7 cc t.-e. lead	80.0	77.2	F. G.	Experimental blend
L	Coastal	Highly cracked	1.0 cc t.-e. lead	80.5	72.1	D. A. G.	Experimental blend
M	Columbian	St. run	1.85 cc t.-e. lead	80.1	78.8	F. G.	Experimental blend
N	Mid-Continent	St. run	6.0 cc t.-e. lead	80.4	79.7	D. A. G.	Experimental blend
O	Select Smackover	St. run	27% 90s benzol	80.1	65.8	F. G.	Experimental blend
P	East Texas	Highly cracked	30% 90s benzol	80.5	59.2	F. G.	Experimental blend
Q	East Texas	St. run	43% 90s benzol	80.5	64.7	F. G.	Experimental blend
R	Mid-Continent	St. run	56% 90s benzol	80.5	64.9	D. A. G.	Experimental blend
S	California	St. run	App. 35 benzol	80.6	70.0	.....	Special experimental blend

[a] F. G. signifies that the fuel meets all of the provisions of U. S. Army specification Y-3557-E, aviation gasoline, fighting grade, with the exception of octane number. D. A. G. signifies that the fuel fails in one or more items to meet F. G. specifications, but that it meet domestic aviation gasoline specifications in every particular. It should be borne in mind that any fuel meeting the fighting grade specifications will of course meet D. A. G. specifications.

[b] Standard reference fuel B-2 is an especially prepared secondary testing fuel, checked against pure octane and heptane. Secondary standard fuels are used in routine test work to avoid the expense of rating against the chemically pure octane and heptane.

Standard Oil Development Company, and the Ethyl Gasoline Corporation, arranged for the purpose of verifying preliminary conclusions based upon the work of the Ethyl Laboratory in this connection. Their results show that the present automotive test conditions, which are even less stringent than the commercial aeronautical conditions (600 rpm, 212 F jacket temperature), fail just as completely to predict gasoline performance in the automobile engine. (Ethyl Gasoline Corporation Report, "Correlation of Knock Testing Engine Ratings With Knocking in Automobile Engines," February 5, 1932.)

Table 1, which is taken from investigations carried out by the Standard Oil Development Company, shows the effect upon octane ratings of various types of aviation gasolines, of varying the knock test procedure from the commercial standard of 300 F, 600 rpm, to the new military standard of 375 F, 900 rpm.

Full-scale flight tests have been carried out by the author on gasolines B to J, inclusive, in one of the test airplanes of the Standard Oil Company of New Jersey. The engine had been rated by the manufacturer as requiring a fuel of 73 octane number, at 300 F and 600 rpm, and it was found to be extremely critical. Fuels C, D, E, F, and H were found to operate with normal cylinder-head and base temperatures and approximately equal maximum rpm and fuel economy, under full-throttle level-flight conditions. Fuel G did not give audible detonation on the ground, but full-throttle, level-flight conditions at ground could be maintained for less than two minutes before cylinder temperatures reached dangerously high figures, with an accompanying loss in rpm of about 200, necessitating immediate partial closing of the throttle. Fuel J gave audible detonation on the ground, and such violent detonation at full throttle in level flight that continuation of the test was deemed inadvisable. Fuel B showed no detonation on the ground. Flight tests showed gradually increasing head temperatures during four full-throttle runs at the ground over a two-mile speed course.

Flight-test data are always subject to criticism because of the inability of the pilot exactly to duplicate all flight conditions, to say nothing of being able to control temperatures, humidity, and other atmospheric variations. It appears to the author that this very lack of uniformity in flight test work, augmented as it undeniably is in routine military and commercial flight operations, provides the most excellent case in favor of choosing a test basis for gasolines which is, as closely as possible, comparable to the most severe engine requirements in service.

Further investigation of the correlation of laboratory test data to engine requirements is at present under way. Through the cooperation of the Wright Aeronautical Corporation, all of the fuels tabulated are at present undergoing full-scale dynamometer tests, under the supervision of Mr. R. F. Gagg. The engine selected for examination of fuels A to J, inclusive, has been found to be very critical of octane-number variations from 73, while that assigned to the test of fuels K to S, inclusive, is equally critical at 80 octane number. In these tests, of course, it is possible to control practically all variations for which corrections cannot be made. Results to date indicate further substantiation of the growing mass of data in favor of changing commercial test conditions to correspond to those required by the governmental agencies for military use. These dynamometer tests will be supplemented by flight tests in one of the Stanavo "Eagle" Lockheed Vega airplanes, powered with a Pratt & Whitney Wasp "C" engine with 10-to-1 supercharger gearing. This engine is supposedly subject to detonation using gasolines rating below 80 octane at 300 F and 600 rpm.

If the results of these investigations and of those now in process by other agencies are in further substantiation of the conclusions which have grown from the previous work, it would seem that the next and a very long step forward would be the adoption, as standard commercial test basis, of rating aviation fuels on octane number at 375 F, 900 rpm, and 15 deg spark advance, in accordance with current Army practice. Further, it would seem logical and practical to grade these fuels in brackets of ten octane numbers, i.e., 50, 60, 70, 80, and 90, assigning each gasoline to the next lower bracket in every case. Under these conditions a fuel which tested in the laboratory at 67 would fall in the 60-octane-number classification, one which tested 84 in the 80-octane bracket, etc. The adoption of such classification would insure that the purchaser of fuel would receive a product which was at least as good as and very probably better than that for which his engine had been proved to be critical by the builder, if he demanded his supplies in accordance with the manufacturer's recommendations. The present commercial method of rating is apt to give fictitiously high values, so far as full-scale engine operation is concerned, which would certainly be materially reduced under the proposed conditions to octane values more nearly representative of engine requirements.

Erratum

On page 38 of AERONAUTICAL ENGINEERING for January-March, 1933, in paper AER-55-5 ("Air Compression With Temperatures Above Adiabatic With Special Reference to Airplane Superchargers," by Sanford A. Moss) in the last line of column (2), "T" should be "T_2."

AER-55-14

National Air-Tour Formulas and Other Methods of Handicapping

By P. ALTMAN,[1] DETROIT, MICH.

Methods of scoring and handicap used in aircraft competition events are given. The first part contains a historical review, and the second contains criticism of handicap formulas, these having a bearing on aircraft design and the improvement of airplanes. In many cases airplanes are especially designed for air races, and if the design is good, it increases progress. The paper takes up the important formulas for handicapping and considers them from a race and design point of view. Useful curves are plotted.



NO TWO contestants in an air race will agree that the handicap is entirely fair. It really is difficult to devise a handicap to suit all competitors, with widely varying equipment and piloting skill, especially in such events as tours, derbies, and free-for-all races. In the National Air Tour for the Edsel B. Ford reliability trophy, the airplanes entered ranged from a single-place airplane with a two-cylinder engine of 20 to 30 hp to a sixteen-passenger airplane with about 1200 hp, and this year a forty-passenger airplane of about 2500 hp might have been entered.

These conditions in an event open to all aircraft require a method of handicapping suitable to the smallest "flivver" airplane and to the biggest air liner, with the necessary factors of simplicity and ease of application.

It is true that handicaps are not always equally just to all entrants, and the fact that events where obviously there was partiality were completed without official protest shows the contestants' high standard of sportsmanship. Without this, all the slide rules and regulations developed by the handicapper or referee would not make for a successful event.

The answer as to why there is a handicap is a simple one, and much has been written on its origin, development, and failings (1).[2] Its origin probably dates back to the earliest organized competition between fellow men and is because in the development of man his stature, ability, and character have had as many forms as the equipment with which he has become accustomed to work. As a result no two groups of men in developing a machine will produce exactly the same structure or performance characteristics. But in their own minds they believe that their equipment is the best for the purpose. This was true in the earlier machines of transportation, and the condition is even more pronounced in the airplane. One designer may believe that low landing speed is essential, another that high speed should be the criterion, and still another that the major requirements are convenience of schedule, comfort of travel, and low cost, regardless of the factors of pay load and speed range.

To judge the varying equipment collectively in any contest presents a problem to the handicapper, whose instructions are to apply the handicap so as to place all contestants on an equal basis. The handicap may take the form of:

(*a*) Personal judgment by a group of authorities, a method practiced with fair success in England (1).

(*b*) A mathematical formula, employed at one time or another in most countries.

(*c*) A point system developed from the characteristics of a hypothetical airplane having ideal performance for a particular case, such as the point system used for the Guggenheim Safe Airplane Competition (2) and many contests in Germany (3).

In archery tournaments (4), one of the oldest of sports, the contestants are handicapped on their record of previous contests and also upon the improvement shown between contests. Automobile (5) races in America are seldom handicapped. Contests are usually divided into classes, with the division based on piston displacement. There are automobile contests, such as the cross-country tours so popular ten or twelve years ago, that are conducted on a handicap basis, the method depending largely upon the objective to be achieved. The figure of merit of each car is obtained by a formula based on the weight of the car in tons multiplied by the miles covered and divided by the gasoline consumed. This can be reduced to a formula familiar to some aircraft efficiency contests:

$$\cdot \frac{W \times S}{C \times \text{bhp} \times t} = \frac{WV}{P \times C} = FM$$

where

FM = figure of merit
W = weight of car in tons
S = distance in miles
C = specific fuel consumption
P = bhp = brake horsepower of motor
t = time in hours
$V = S/t$ = speed in mph.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Numbers in parentheses apply to references at the end of the paper.

In speed events entered by cars of varying size and power, qualifying time is usually the basis of the handicap, and the slowest car is started first. For example, if ten cars are entered in an event on a 1-mile track and the fastest qualifies at 37 sec for one lap and the slowest at 42 sec, the time differential is 5 sec, and for ten cars, we can assume, if the differential between cars is equal, that on a handicap of $^1/_2$ sec per car per lap, all cars would finish at the same time. If the time differential varies, the handicap has to be varied accordingly between cars. Also, consideration is given to the fact that it is difficult for a car to maintain its qualifying speed on a race of considerable length, and the handicap is usually modified an amount determined from experience.

In Europe, more attention has been given to the mathematical handicapping of automobile races, some involving elaborate methods (6). Handicapping by formula (7) is applied to power-boat racing. Golfing, horseracing, sailboat races, and motor-cycle races have received attention in handicapping. It is therefore not surprising to find the most modern of sports, the aircraft contest, conducted upon some form of handicapping, with the method similar to that in other automotive events, but developed to meet the characteristics of the airplane, such as take-off, speed, landing run, and rate of climb.

In aircraft competitive events, the National Air Races and the National Air Tour are major contests which have received the attention of handicapping committees and judges. The events in both contests have gained in prominence from year to year, yet seldom has either been conducted under the same rules for succeeding years. The rapid development of the equipment entered in these events has been the primary factor dictating such changes.

In 1921, in the Air Races at Omaha, the handicapping formula for the John M. Larsen Efficiency Trophy was in the form:

$$FM = \frac{(W - T) \times S}{G}$$

and can be seen to have no particular significance if reduced to the form:

$$\frac{W_u}{C \times P \times t} \times V = \frac{1}{C} \frac{W_u}{P} \times \frac{V}{t}$$

where V/t appears to have no relation to the performance characteristics of the airplane.

FM = figure of merit
G = fuel consumed = $C \times P \times t$
C = specific fuel consumption
$S = V$ = average speed
t = time in hours
$(W - t) = W_u$ = useful load
P = bhp of engine.

Subsequent efficiency contests modified this formula variously but again in another year it is found in almost the same form, and given as:

$$FM = \frac{W_u \times V \times 3}{\text{gas consumed}}$$

The similarity to the 1921 formula is obvious, if we substitute for gas consumed $C \times P \times t$.

In other years, the winner in the efficiency contests of the air races was determined by various handicapping formulas, all of them possessing certain merits and most of them stressing particularly the ratio of useful load to the power consumed, but very few giving any consideration to the safety or reliability of the airplane. In one case, we find the winner determined by the formula:

$$FM = \frac{\text{average speed} \times \text{constant}}{\text{piston displacement}}$$

On analysis, this formula does not appear to have any practical or theoretical basis, but evidently served some purpose for which it was intended. In another instance, the handicap was applied by the formula:

$$FM = \frac{\text{weight}}{\text{gas consumed}}$$

And so on, other contests employing such formulas in many different forms. If we substitute in the latter formula for gas consumed its equivalent given by the product of $C \times P \times S/V$, with the terms having the same significance given before, we obtain:

$$FM = \frac{W \times V}{C \times P \times S}$$

Further, if the contest is conducted over a course which is of the same length for all contestants, then S is constant and

$$FM = \frac{W \times V}{K \times C \times P}$$

which is a familiar form and quite popular in many contests such as the automobile economic races. The figure of merit is a function of the specific fuel consumption, a factor difficult to measure accurately even under the best of conditions and particularly in contests in which a large number of airplanes are entered.

A handicap used for several years in the Detroit News Efficiency Contest of the National Air Races is:

$$FM = \frac{W \times V}{P}, \quad \text{or} \quad \frac{W \times V}{\text{piston displacement} \times K}$$

differing from the other in that the specific fuel consumption has been eliminated. These formulas do not contain any factor governing the rate of climb or stalling speed of the airplane. Under this method of handicapping, it is entirely possible to win the contest by loading up the airplane, provided a large enough field is available for take-off. In one case it was possible to overload an airplane 50 per cent without decreasing the maximum speed more than 2 per cent and resulting in an increased figure of merit of over 100 per cent, all accomplished at the expense of increased stalling speed, decreased rate of climb, and increased take-off run, the latter factor jumping from about 400 ft to 1600 ft, or 300 per cent. Therefore, since take-off distance and the other factors affected by high-wing loading are not accounted for in the formula, winning the contest was a "walk-away" for this particular airplane. This condition will appear obvious if one examine the factors involved in the formula. For a given airplane, the power and wing area are constant, and on the basis of Warner's formula (8) for maximum speed, where $V = K \sqrt[3]{\frac{P}{S}}$, it can be seen that the speed is also constant and independent of the weight. Therefore the figure of merit becomes a direct function of the weight of the airplane, and its magnitude depends largely on the size of field available for take-off. Of course, the formula $V = K \sqrt[3]{\frac{P}{S}}$ assumes constant propeller efficiency and rpm, an assumption which is not entirely correct. However, the variation in these factors is small.

The 1931 National Air Races adopted the 1930 National Air Tour formula for its efficiency contest. This formula was de-

veloped from the experience and data obtained in the National Air Tours from 1926 to 1929, and is the result of the combined efforts of the Rules Committee of the tour under the chairmanship of Mr. Wm. B. Mayo and the suggestions and constructive criticisms from contesting pilots and members of the aircraft industry. In 1929 and some previous years, the method of handicapping in the National Air Tour was applied by the formula:

$$FM = \frac{W \times V \times K}{(S/2 + US) \times \text{piston displacement}}$$

By this formula, it was attempted to remove the objectionable features of the efficiency formula $FM = K \times (WV)/P$, so often used in the National Air Races, by introducing in the denominator the factor one-half stick time plus unstick time, where stick time was the time in seconds required to stop the airplane after landing, measured from the instant the airplane touched the ground to the instant it stopped rolling, and unstick time was the time in seconds required for the airplane to take off, measured from the instant the wheels began to turn to the last time that the wheels touched the ground. These terms will be referred to as S and US, respectively.

By the inclusion of the factors of stick and unstick time in the older formula of $FM = (W \times V)/P$, the figure of merit at once became an index to not only the weight-carrying efficiency and speed but of safety and reliability as well. The weight-carrying efficiency is represented by the ratio of the useful load to the horsepower, the speed enters directly in the term maximum speed, the reliability through the length of the tour and penalty imposed for forced landings, repairs, etc., and safety is excellently represented by a liberal reserve of power and low landing speed, both of which enter into the determination of the term of stick and unstick time (10).

Yet at the end of the 1929 tour, practically all contestants clamored for a change of some kind. It is true that the formula contained all the essential factors necessary for a just and fair

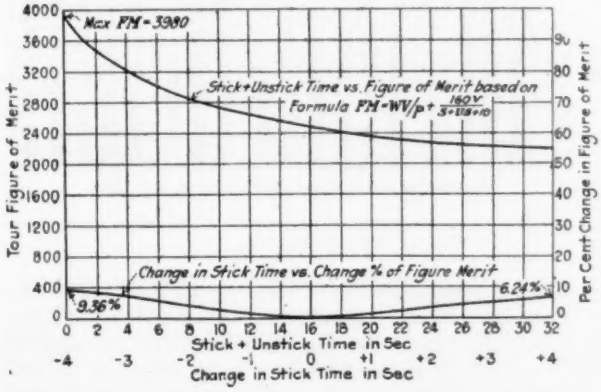


FIG. 1 EFFECT OF SMALL DIFFERENCE ON FIGURE OF MERIT FOR EACH TOUR FORMULA DUE TO ERRORS IN TIME OR SKILL OF PILOTS

handicap and considered all the factors pertaining to both aerodynamic and structural efficiency, but it was questioned whether these factors were in exactly the correct form and with each given its proper credit.

The terms of useful weight per horsepower and maximum speed were accepted without argument, but not so with the term of stick plus unstick time. The effect of this term on the total figure of merit is of primary interest. It appears that an unduly large percentage of the credit is given to this factor, and is shown by Fig. 1. In this is plotted the figure of merit against stick plus unstick time, on the assumption that the term $(W \times V)/P$ remains constant. The intention is to show the effect of changing values of stick plus unstick time on the final score as a result of the improvement of brakes, changes in landing angle, and the condition of the use of brakes and no brakes. It will be seen that the curve is asymptotic and that if the term $S + US$ could be made to approach zero, the figure of merit would approach an infinite value, although the useful load would be very small. While it is improbable that the term $S + US$ would ever be zero, yet it is clear that for small values of this term the figure of merit is unduly influenced by it. This is also shown by Fig. 2, where is plotted the percentage increase in figure of merit due to small changes in the term $S + US$. A decrease of 1 sec changes the figure of merit about 9 per cent. Primarily, this chart indicates the effect of small errors on the part of the timers and either poor

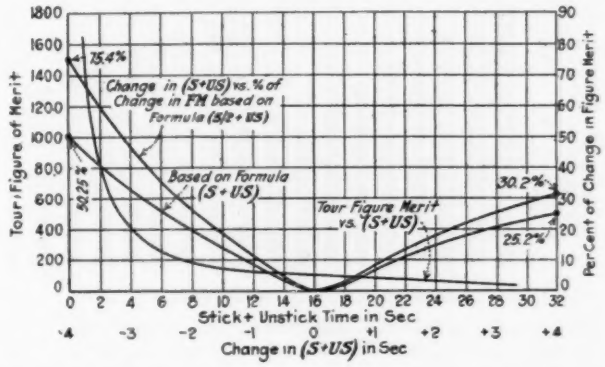


FIG. 2 EFFECT OF SMALL DIFFERENCE ON FIGURE OF MERIT FOR EACH TOUR FORMULA DUE TO ERRORS IN TIME OR SKILL OF PILOTS

judgment or bad brakes on the part of the pilot. In general, this effect is too great and the penalty paid by other contestants is too large compared to the questionable fact of the additional safety obtained by a decrease of 1 sec in either the stick time or the unstick time, which might sometimes easily be accounted for by error in timing.

In an attempt to eliminate some of the objectionable features, the stick time in the formula for 1929 was changed to one-half stick time. This, however, instead of improving conditions, actually made them worse. The term $S/2 + US$ was now smaller than $S + US$, and thus the effect of small differences such as 1 sec or less became even more pronounced. A decrease of 1 sec in $S/2 + US$, whether due to error or otherwise, now resulted in a change of about 11 per cent in the figure of merit.

However, what is true for the low values of the term $S + US$ is not the case at high values. The conditions are entirely reversed.

If we again assume the factor $(W \times V)/P$ as constant and take into account the effect of changes in the term $S + US$ due to pilot's error, poor brakes, timer's error, etc., we can see from the curve that increasing the value $S + US$ has a much smaller effect on the figure of merit of the airplane. In other words, the penalty paid at high values of $S + US$ is considerably less than at low values.

The desire on the part of any one faced with the problem of eliminating the objectionable feature of this method of handicapping or the development of some other method would naturally be to set up conditions to be fulfilled and determine as accurately as possible how the various factors in the formula obey these conditions and whether they are arranged in the order in which particular stress will be laid upon those items which the contest is aiming to develop. It is axiomatic that no contest is conducted unless some specific good is to be obtained from it. In speed contests, it is the refinement of design to obtain increased speed without increasing power; in efficiency contests,

it is the improvement in performance for a given structural weight and power; in safety and reliability contests, it is the ability in the former to land and take off safely in and from restricted areas; and in the latter, to reduce the occurrence of forced landings to a minimum.

From what has been stated, it is obvious that the tour formula as used up to 1930 laid particular stress on the effect of the factor $S + US$, and it did this well and is indirectly responsible for the development of efficient wheel brakes on commercial aircraft.

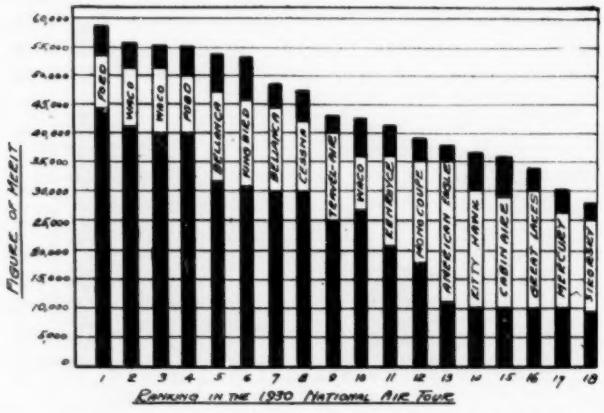


FIG. 3 RANKING IN THE 1930 NATIONAL AIR TOUR

Now that this has been accomplished, it is logical that its effect in the formula should be reduced. Hence, this would form one of the first factors to be considered in changing the present formula or adopting a new one.

Other factors of paramount importance to be considered in this case or any other are:

1 The formula should be simple for computation and simple in appearance and should be easily handled by the average pilot.

2 All factors should be easily obtained in the field and should involve a minimum chance of hazard due to crashes. Factors involving dimensions of the aircraft which are difficult to measure accurately on different types of airplanes should be avoided. Factors which require maneuvers close to the ground should be avoided for the sake of safety and adverse public opinion in the case of a crash.

3 Every factor in the formula should be effected by the variation in the airplane speed along the route in the case of a derby or tour and by every lap in the case of a closed-course event.

4 When a formula has the necessary factors to accomplish the desired results of the contest but not arranged in the proper order, the new formula should not differ materially from the old one. In some cases, however, conditions sometimes warrant new factors and a complete new formula. This is particularly true where the old handicaps have outlived their usefulness.

5 In order to develop a competitive spirit along the whole route of a tour or derby or on each lap of a closed-circuit contest, it is essential that the factors of the formula should be so arranged that the grouping of the contestants be so close that the standing of each entrant may vary from lap to lap, without of course neglect of aerodynamic or structural efficiency.

6 Due consideration should be given useful load, maximum speed, power, take-off time, landing time, reliability, and in the case of a tour passenger comfort.

7 The effect of any factor, such as take-off time, on the final score should not be excessive, except in such contests where it is highly desired to lay stress on this particular factor.

8 The value of any factor in the formula should be permitted to approach zero without the final score approaching infinity.

9 The formula should be as theoretically and fundamentally correct as simplicity will permit.

10 The factors in the formula should be so arranged that the winner will not be known until the end of the contest.

From these requirements the tour committee evolved the following 1930 formula, which was approved by the majority of the manufacturers and is believed to approach very closely to the specifications outlined:

$$\text{Figure of merit} = \frac{\text{Contest load} \times \text{Speed}}{\text{Horsepower}} + \frac{160 \times \text{Speed}}{\text{Stick} + \text{Unstick} + 10}$$

Contest load is the Department of Commerce allowable contest load, equal to the gross weight of the airplane minus the weight empty and corrected for cabin accommodations and amphibian landing gear

Speed is the actual average measured speed on each leg of the tour

Horsepower is the Department of Commerce rated horsepower for each airplane and corrected in the case of the multi-motored airplane

Stick is the time required to stop the airplane from rolling after landing

Unstick is the time required for the airplane to leave the ground during the take-off.

It should be noticed that the formula is divided into two parts, the first relating to weight-carrying efficiency and speed and the

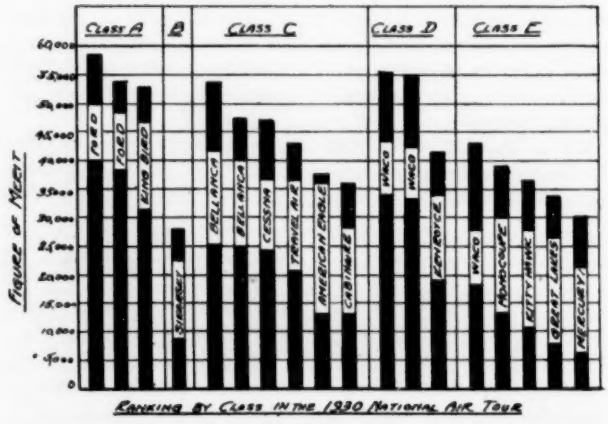


FIG. 4 PER CENT OF BRAKE-HORSEPOWER WITH MOST EFFECTIVE MOTOR STOPPED

second to reliability and safety in landing and take-off in restricted areas, as well as being somewhat connected with speed range. Proof that this formula has accomplished the results requested by the majority of the contestants is in the records of the 1930 tour. For instance, among the airplanes winning from first to tenth place is found a representation of every type of airplane, from a tri-motored Ford and twin-motored Curtiss to the small 90-hp Waco and Cessna. The competitive aspect is seen by the difference in standing between contestants and is illustrated in Fig. 3. Second place was only 5 per cent behind first, third place only seven-tenths of 1 per cent behind second, and fourth place only one-half of 1 per cent behind third. As a flying laboratory, the tour cannot be excelled. New designs are tested under actual service conditions, and old designs show up their faults, such as poor reliability of engine, airplane, and propeller. One manufacturer stated that the tour brought forth a fault in his design which otherwise might have taken six months to determine. The tour is to the airplane what the proving ground is to the automobile.

Further analysis of this formula in the light of additional performance figures seems to indicate that while the formula is fundamentally sound, yet the additional handicaps given to multi-motor airplanes should receive further attention. The multi-motor handicap, as given in the rules for 1929, 1930, and 1931, is as follows:

Multi-Motor Handicap, 1929: An entrant of a multi-motored ship will be allowed a lower cubic inch displacement figure than that which actually exists and that used in the formula providing he can demonstrate to the satisfaction of the contest committee that his plane is capable of performance as follows: (*a*) it will be required that the pilot of a three-motored ship climb from 1000-ft altitude to 2000-ft altitude as indicated on a calibrated altimeter carried by the contest committee observer as follows: With the center engine dead, he shall climb from 1000 ft to 1400 ft; with the right engine dead, he shall climb from 1400 ft to 1700 ft; and with the left engine dead, he shall climb from 1700 ft to 2000 ft level. If he shall accomplish this to the satisfaction of the contest committee observer, he will then be allowed a cubic displacement figure of two-thirds of the actual cubic displacement of all motors. (*b*) Where a twin-motored ship is employed, the pilot must climb from 1000 ft to 1300 ft with one motor dead and from 1300 ft to 1600 ft with the other motor dead. If successful, he will be allowed a cubic displacement figure of one-half of the actual cubic displacement of the two motors. In the event of failure to do this in either case to the satisfaction of the contest committee observer, the total cubic inch displacement of all engines will be taken as the displacement figure in the formula.

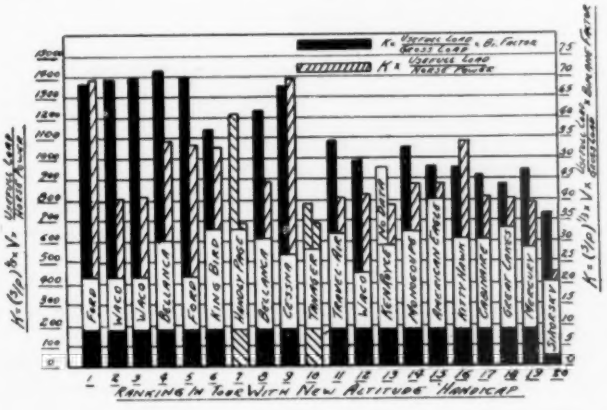


Fig. 5

Multi-Motor Handicap, 1930 and 1931: An entrant of a multi-motored ship will be allowed a lower horsepower figure than that which actually exists, and this will be used in the formula providing he can demonstrate to the satisfaction of the contest committee that his plane is capable of performing as follows: (*a*) A multi-motored ship will be required to climb immediately on all engines to 6200 ft above sea level with full contest load, and during the next 15-min interval the plane must not descend below 6000 ft when flying with the most effective engine dead. If this is accomplished the total rated horsepower of the remaining motors will be used in the formula. The most effective engine of any combination will be determined by the contest committee. (*b*) In the event of failure to do this, the total horsepower of all engines will be taken as the horsepower figure in the formula.

From the rules of 1929, it can be seen that a multi-motored airplane is required to demonstrate its ability to climb to certain altitudes, but no stipulations are made on the time required to reach these altitudes. In other words, the airplane which is heavily loaded with gasoline, usually about 30 per cent of its useful load, could take ten minutes or an hour to gain the desired altitude by climbing at a very slow initial rate and thus use up enough gasoline to reduce its weight sufficiently to enable it to reach the handicap altitude with any engine stopped.

The effect of reducing weight in the multi-motored handicap also involved the fact that no definite gross weight was specified with which the altitude tests were to be conducted. It was

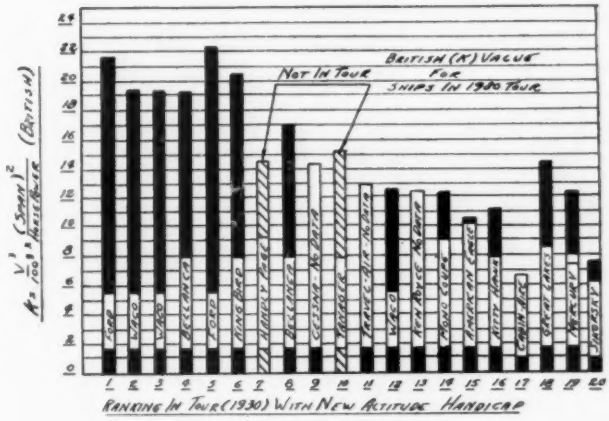


Fig. 6

possible to reduce the useful load in certain airplanes a sufficiently small amount to materially increase the absolute ceiling for performance with any engine stopped and receive a much greater benefit in the final score due to this handicap than that lost due to reduction in useful load. In the case of a twin-motored airplane this might represent a net gain of about 45 per cent.

The multi-motor ruling of 1930 and 1931 attempted to avoid this condition by setting a time limit on this performance and specifying the weight with which it was to be demonstrated. In addition, the handicap was changed from the previous basis of ability to climb on part power to the basis of settling ceiling on part power. Although the new requirements were an improvement, the factors involved worked a hardship on certain airplanes and insufficient credit was given to others. For instance, the airplane either had to complete the performance test successfully or forfeit the handicap credit entirely. Other airplanes which had ample reserve power at the settling ceiling were not given additional credit for it.

Additional investigation indicates that the multi-motored handicaps should be altered, and the following changes have been suggested:

1 The handicap credit allowed multi-motored airplanes should be based on a sliding scale. A method based on experience gained from previous contests and tempered by fundamental considerations is shown in Fig. 4, and has been suggested for future tours.

2 A standard method of correcting altitude readings for temperature and pressure changes should be adopted. For this, Diehl's correction has been suggested, where $h = h_p - k(h_p - h_d)$, where h_p and h_d are the pressure height and density height, respectively, and K has an average value of 0.36.

3 The credit allowed multi-motor airplanes should be reduced. A suggested reduction is shown in Fig. 4.

The effect of these suggested changes (as shown in Figs. 5 and 6) indicates the change in position of the contestants produced by these revisions.

Another change of importance often suggested is to divide the contestants entered in the tour into different classes, the divisions to be based on the type of airplane and its power rating. A defi-

nite plan of this type proposed for future tours or derbies is as follows:

1 Multi-motored land airplanes, any gross load, any power.
2 Amphibians, single or multi-motored, cabin or open cockpit, any gross load, any power.
3 Single-motor cabin airplanes, engine 160 hp or greater.
4 Single-motor open-cockpit airplanes, engine 160 hp or greater.
5 Single-motor cabin or open-cockpit airplanes, engine 80 to 159 hp inclusive.
6 Single-motor cabin or open-cockpit airplanes, engine 10 to 79 hp inclusive.

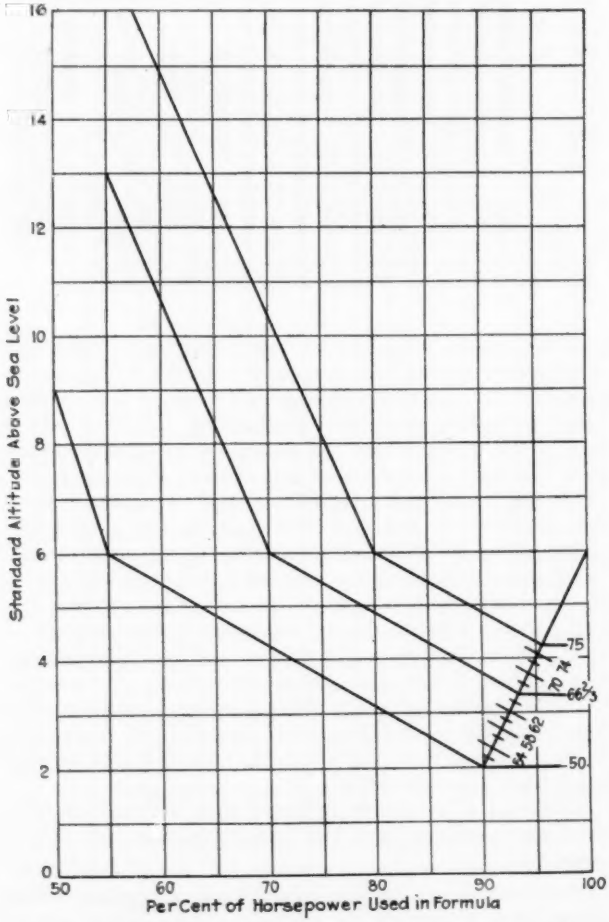


FIG. 7 RANKING BY CLASS IN THE 1930 NATIONAL AIR TOUR

The effect which this change would have had on the 1930 Tour contestants is illustrated in Fig. 7. From the arrangement of the classes, it will be seen that there are more divisions according to power in the lower powered classes than in the higher ones. This division is based on a study of the results of previous tours, arranged in a manner suggested by Warner (12), and illustrated in Fig. 8, where is shown the winning figure of merit and winning speed plotted against the corresponding horsepower of the airplane for the entrants in the 1930 tour. In addition, Warner's (12) curve of winning speed against horsepower for the A.T.C. airplanes in the 1931 National Air Races is also shown. From this chart, it will be seen that the division is fair and should result in a very interesting contest between the airplanes in each class.

A performance chart developed by the author to study the possible characteristics of new airplanes which might be entered in the tour is shown in Fig. 9. A solution for the figure of merit for any airplane by the Tour Formula is shown by the dotted line on this chart, for which the procedure is as follows:

(*a*) Obtain the values of the gross weight, wing area, and power of the airplane and also the ratio of the useful load to the gross load. If the last factor is not shown, it can be assumed to have an average value of 45 per cent of the gross weight.

(*b*) Calculate the maximum speed by the formula $V = K\sqrt[3]{P/S}$, using a conservative value of K (135 to 150).

(*c*) Enter the chart at the value of weight/horsepower (*a*), and project horizontally to the corresponding value of W, then vertically upward from (*B*) to the unstick scale S, and continue to the intersection (*C*) of the curve in the upper right-hand section of the chart. From this point, project horizontally to the intersection of the corresponding value of Useful load/Horsepower or W_1/HP (*D*), then drop vertically to the correct value of maximum speed (*E*) and horizontally to the figure of merit (*F*).

The section of the chart for unstick time is based on Diehl's formula for take-off run in a calm which is given by the equation (3):

$$X = KV_s^2 (W/P)$$

By assuming constant acceleration and substituting $X = V_s t/2$ and $V_s = K\sqrt{W/S}$, we get:

$$t = K_1 \sqrt{\frac{W}{S}} \cdot \frac{W}{P}$$

where

K and K_1 = constants and were checked by actual values measured in the tour
t = unstick time
V_s = stalling speed taken equal to landing speed
W = gross weight of the airplane
X = take-off run in feet
S = wing area in square feet
P = brake-horsepower of the motor.

The upper right-hand section of the chart is simply the ratio of $S + US + 10/US$, which appears to have a very constant value, as will be seen from the plotted points.

The curve is obtained by plotting the term $(S + US + 10)$ as ordinate and US as abscissa from data obtained in the 1929, 1930, and 1931 tours.

The rest of the chart consists of simple addition and multiplication curves, which are necessary to obtain the solution of the equation

$$\left(\frac{W}{P} + \frac{160}{S + US + 10}\right) V = FM$$

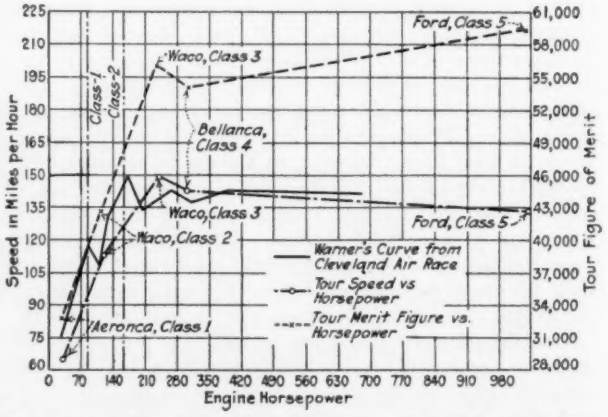


FIG. 8 CURVES FOR FIGURE OF MERIT AND SPEED VERSUS HORSEPOWER

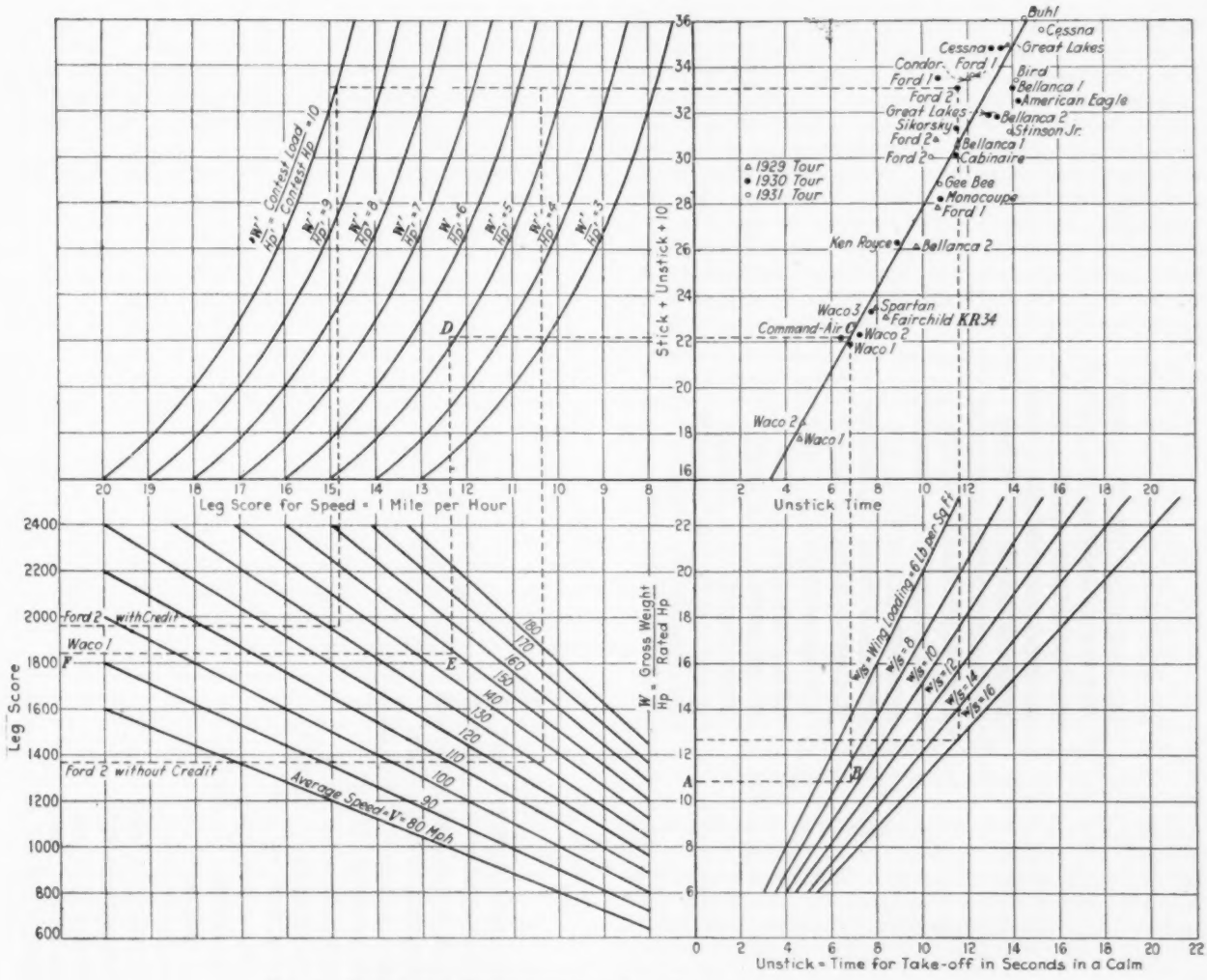


FIG. 9 CURVES FOR COMPUTATION OF LEG SCORE IN NATIONAL AIR TOUR

(Formula: $\frac{W}{HP} + \frac{160}{\text{Stick} + \text{Unstick} + 10} \times \text{Leg speed} = \text{Points.}$)

Other methods of handicapping have been suggested from time to time for both the National Air Tour and National Air Races. Most noteworthy of these has been a proposal to adopt Warner's formula for maximum speed as a handicapping formula. This appears to have considerable merit to it, from both a theoretical and practical viewpoint. The formula 8 as presented by Warner is:

$$V_{max} = K \sqrt[3]{P/S}$$

where

V_{max} = maximum speed of the airplane; the units can be either feet per second or miles per hour

P/S = the ratio of rated brake horsepower to the wing area of the airplane

K = assumed as a constant for all aircraft (14).

This constant is really a function of (*a*) the drag coefficient of the whole airplane, (*b*) the propeller efficiency, and (*c*) the maximum lift coefficient of the airplane. Thus the better the streamlining, the higher the propeller efficiency, and the higher the value of the maximum lift coefficient, the greater will be the value of K. If the maximum speed is measured and the ratio of the horsepower to wing area is known, then this value of K should be a good index to improvement in design and performance, and in this form could be used as the figure of merit in such airplane contests as tours, derbies, and even speed events. The fact that the formula considers the stalling speed of the airplane through the wing-area factor, as well as accounting for the weight, power, and speed, should make it well suited for these events. In events open only to A.T.C. airplanes, the value K computed on the basis of horsepower per unit of area is probably the best suited for the purpose. But in events open to group 2 and experimental racing airplanes, the value of K should be based on engine displacement per unit of area, to avoid the questionable amount of the power of the engines with which these airplanes are equipped and to afford a ready means of checking in the case of a protest.

However, the value of any formula for handicapping can only be judged from the results which it produces. To obtain a comparison of this formula with others, values of K have been calculated from data obtained in the National Air Tour and are shown graphically in Fig. 10, where the ranking according to K is plotted against the ranking according to the tour formula. It appears that the ranking obtained by the two methods do not agree very closely. One obvious reason for this condition is that the tour formula is based on the useful load, while the value of K is based on the gross load of the airplane. Since one of the requirements of the tour is efficiency, the useful load should govern instead of

the gross load. New values of K based on the ratio of useful load to gross load were calculated by multiplying the first value by this ratio, and are shown graphically in Fig. 11. The agreement by the two methods is now better throughout a great part of the range, but the ranking still differs in the first part of the range and seems to favor the biplanes. This is probably due to the wing-area factor in the formula, which for biplanes is higher for the same take-off and landing speed than for monoplanes. Correcting for biplane interference further improves the agreement, and replotting the ranking based on the new values of F against the ranking under the proposed new Tour rules shows a remarkably good agreement between the two methods. The correction for biplane effect is a function of the induced drag, and since the induced drag is small in magnitude at high speed, it would seem at first that this correction is unnecessary, and this is so if maximum speed is the only criterion. But where high speed is desired in combination with a good cruising speed, the induced drag is an important factor, and in speed efficiency contests its effect should be accounted for in the handicap. The formula in its final form, then, is:

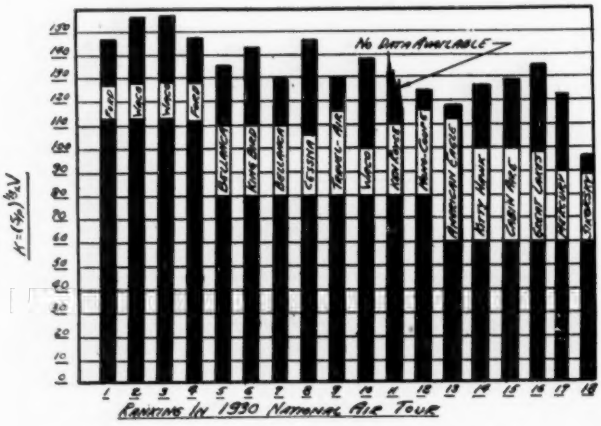


Fig. 10 Ranking in 1930 National Air Tour

$$K = V\left(\frac{S}{P}\right)^{1/3} \times \frac{W_u}{W_g}$$

where W_u is the useful load, Wg the gross weight, and S the equivalent monoplane area. Due to the simplicity of this formula $K = V \times (S/P)^{1/3}$, it would appear that it could be substituted for the tour formula and other efficiency handicaps. However, its very simplicity makes it rather difficult to apply, which is caused primarily by the work involved to obtain the correct wing area of all the types of aircraft that may be entered in a contest and for which wind-tunnel data are not available.

Consider, for instance, the case of a monoplane with lift struts of large area and perhaps in addition a stub wing at the fuselage. Here the determination of the equivalent monoplane area would present quite a problem; consider also the case of the various possible arrangements of the biplane as well as the effect of large tail areas and different fuselage shapes. All these complicate the problem and form the primary reason why a wing-area factor was not included in the tour formula. Another factor which would militate against the use of K as a tour or derby handicap would be that it does not allow any credit for the use of efficient wheel brakes. The method, however, is highly recommended in such cases where the equivalent monoplane area could be accurately determined and effect of landing run is not desired.

Another method of correcting the value of K for the weight-carrying efficiency of the airplane consists of multiplying the value of K by the ratio of the useful load to the total horsepower instead of the useful load to the total gross weight. The latter is a function of the structural efficiency, while the former is an index of the efficiency on the basis of horsepower. Both methods have their merits, and it has been suggested that the product of both ratios be used.

Expressed in symbols having the same significance as before, the formulas now become:

$$K_1 = V_{max}\left(\frac{S}{P}\right)^{1/3} \times \frac{W_u}{W_g} \quad \ldots\ldots\ldots\ldots [1]$$

$$K_2 = V_{max}\left(\frac{S}{P}\right)^{1/3} \frac{W_u}{P} \quad \ldots\ldots\ldots\ldots [2]$$

$$K_3 = V_{max}\left(\frac{S}{P}\right)^{1/3} \frac{W_u}{W_g} \cdot \frac{W_u}{P} \quad \ldots\ldots\ldots\ldots [3]$$

Values of K have been calculated by all three methods and are shown plotted against the tour ranking in Figs. 11 and 5. The relationship between the methods using the tour as a standard is self-evident from the charts. These formulas were compared with the tour formula, not because the author believes that the latter should be used as a standard of comparison, but for the simple reason that more accurate data were available for this event than any others with which the author is familiar.

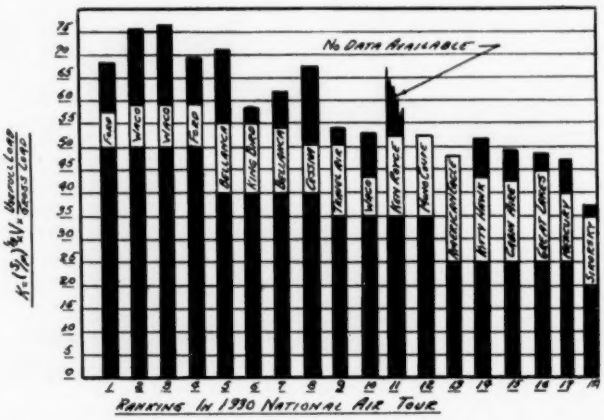


Fig. 11 Ranking in 1930 National Air Tour

In 1926 and 1927, the King's Cup Race was conducted in England on handicapping formula (*15*), somewhat similar in form to the one proposed by Warner except that wing span was used instead of area. The handicap speed is given by the equation:

$$V = 100\sqrt[3]{K \times \frac{P}{b^2}}$$

where b is the equivalent monoplane span and for biplanes of equal span was equal to 1.265 times the span of the biplane; V is the maximum speed in mph, P the brake horsepower of the engine, and K a constant. The span-correction factor used in this formula is actually an area-correction factor, so that the term P/b^2 becomes, as mentioned before, similar to P/S.

In this formula, as in Warner's, K could be made the index factor of figure of merit, but a study of this formula in comparison with other methods indicates very definitely, as shown in Fig. 6, that it unduly favors the monoplane and is subject to the criticism that correcting the span of a multiplane airplane to that of a monoplane is at best a difficult problem.

The point system of handicapping in which characteristics of

a contesting airplane are compared with a set of requirements of a hypothetical airplane has received considerable study both in America and Europe. Noteworthy of the European events was the German Seaplane Contest (3) of 1926, which was based upon such characteristics as maximum speed, take-off speed, rate of climb, fuel consumption, and endurance. In America, the outstanding event of this kind was the Daniel Guggenheim Safe Airplane Competition, in which were stressed such factors as low landing speed, gliding angle, take-off run, and useful load.

The method has considerable merit for certain contests, but depends largely on the experience of the handicappers in allotting the proper number of points for each performance factor. In addition, it usually involves factors which are very difficult to obtain in the time available for such measurements. To illustrate the effect of credit allowed for certain factors, the figure of merit for the airplanes winning first and second place in the Guggenheim competition was calculated according to the tour formula and Warner's K factor and are shown plotted with the tour airplanes in Figs. 5 and 6. It is to be noticed that although these airplanes possessed some excellent characteristics, yet the fact that certain unusual performance factors were required of them, placed them at a disadvantage with respect to other airplanes in such events as a tour, derby, or efficiency contest.

A method of handicapping proposed for the 1932 Transcontinental Derby and which has been used in other contests (5) with considerable success, consists of starting the fastest airplane an interval of time after the departure of the slowest airplane, so that with given equal conditions, both should arrive at the end of the lap at the same time. A similar procedure is to be followed for the second fastest, third fastest airplane, etc. The time in seconds in which the contestant arrives at the end of the lap ahead of his schedule is to be the figure of merit.

The maximum speed of each airplane is to be determined at the start of the derby by disinterested pilots, who will fly each airplane over a speed course with the machine in the same condition in which it will be flown in the contest. This method requires a tremendous amount of speed testing before the derby leaves, and the test, unless corrected for atmospheric conditions, is subject to error from this source in addition to other errors.

Since the length of each lap is a function of the distance between good airports on the route, it is obvious that the length of laps will vary from point to point and will thus require a new handicap figure for each lap. However, the speed of each airplane will be known before the contest starts, and it should therefore be a simple matter to compute the handicap for each lap before the derby actually leaves the first control point.

A large number of the airplanes will be closely matched on speed, which will necessitate very close starts and require extreme alertness on the part of both the timer and pilot, not only at the start of the lap, but even more so at the finish. The proposed handicap can be expressed as:

$$\text{Handicap} = \frac{\text{Length of lap}}{\text{Speed of airplane}} - \frac{\text{Length of lap}}{\text{Speed of slowest airplane}}$$

This system of handicapping is of course applicable only to such contests where the skill of the pilot is the primary factor in determining the winner. No attempt is made to allow for aerodynamic or structural efficiency or even safety, the object being almost entirely a close and spectacular finish between contestants. The factor of reliability does receive some consideration, but only from the fact that the route is a long one and motor trouble on any lap would severely handicap the contestant concerned.

A factor of paramount importance in any method of handicapping is the requirement that the condition of the aircraft remain exactly the same during the whole contest as it was during the handicapping trials. Propeller-blade setting, carburetor adjustment, type of fuel, position and amount of ballast, streamline fairings, and throttle position for maximum speed should not be altered. The throttle position for maximum speed refers of course only to airplanes equipped with supercharged engines, a type which is becoming quite popular on commercial aircraft. Airplanes equipped with these engines are required by the Department of Commerce to install throttle spots, which limit the speed of the engine on the ground and also at different predetermined altitudes. It is entirely possible, and the act can be easily carried out with such an installation, to pull the throttle beyond the spot while the airplane is either on the ground or at low altitude and thus obtain an increase of about 20 per cent or more in the horsepower of the engine. The exact amount depends on the degree of supercharging and the type of fuel used.

It is therefore possible to obtain a substantial increase in speed or decrease in take-off time by such a procedure. Where the tests on flights are to be made at low altitudes, three possible procedures present themselves as a solution of the problem, and they are:

(*a*) Handicap the airplane having this installation an arbitrary amount.

(*b*) Seal the throttle position at the carburetor at the point at which the tests are to be conducted.

(*c*) Conduct the flight tests with the throttle at its maximum open position beyond the spots and base the handicap on the speed obtained under this condition.

Of the three methods suggested the last one is recommended, due to its simplicity and ease of application. However, the procedure to be followed should be decided upon by the contesting pilot.

In all three methods and also for all other speed trials the propeller-pitch position should be sealed immediately after these trials, except in the case of propellers whose pitch can be adjusted automatically from the cockpit. The sealing can best be accomplished by marking the propeller hub bolts with a paint of special color.

The correct determination of maximum speed presents a problem of great importance in a contest in which speed is the factor deciding the winner and where these speeds are measured over a short straightaway or closed course. The accuracy and the method employed depend upon the magnitude of the speed to be measured and the length of the course on which the trials are made.

For instance, for a 1-mile course an error of 1 sec at speed of 100 mph corresponds to an error in speed of 2.78 per cent, while for a speed of 300 mph a difference of 1 sec corresponds to an error of 8.34 per cent. For longer courses, the error is reduced proportionately, so that the difference in time of 1 sec in the latter case on a 2-mile course produces an error of 4.17 per cent and corresponds to a distance of 220 ft traveled by the airplane. It is therefore obvious that stop-watch timing is not applicable to such speeds, and some more reliable and accurate method is required which is less subject to the human element. Various methods have been employed for this purpose, all of which have been a combination of photographic and electric timing. Noteworthy among these is the method employed in timing of the Schneider Cup Race (16) of 1931 and the method of timing in use on the French official speed course (17). A modification of the latter method has been employed in America in the timing of high-speed events, such as the Bayles trials to set a new mark for the world's speed record for landplanes.

This method consists essentially of two high-speed cameras, each mounted on a base located to one side of the start and the finish pylons of the course, respectively. The position of the base is arranged on a line perpendicular to the course and passing

through the center of the pylon and is perpendicular to the line of the course. In the shutter of the camera is included a circuit breaker which forms part of an electric circuit connected to an accurate electric timer. Thus it is possible to photograph the airplane as it passes over the pylon and the position of the airplane with respect to the pylon at which the photograph was taken and the corresponding time printed on the timer. This procedure is repeated at the end of the course, with the electric circuit of both cameras connected to the same timer.

While this method eliminates the human element in correcting by means of the photographs for exactly the point at which the time was taken, yet unless the operator at the camera has had sufficient experience with this method, he is likely to miss the airplane entirely and therefore obtain no photographic record at all.

For this reason, the British method in which continuous photographs are taken by means of a motion-picture camera is to be preferred. In such contests where take-off or landing time are factors in the handicap the effect of head wind on these factors should receive due consideration and corrections should be made for the magnitude to this effect on each test.

In conclusion, the writer wishes to make clear that in this presentation it was not attempted to solve the problem of handicapping for aircraft nor to advocate that handicapping by formula is the only feasible method available, but the intention was simply to present such information with which the author is familiar for the good which others might get from it for future work in this direction.

It must be remembered that any handicap determined by formula will favor the airplanes of better aerodynamic performance and structural efficiency and usually does not consider the effect of the pilot's skill. Whether this should be so depends entirely upon the character of the contest.

Formulas which depend upon arbitrary constants are subject to the criticism that the constants do not provide for the case of the "freak" airplane or one specifically designed for the event by taking advantage of the weak features of these constants through special arrangement of the design. If such constants are necessary and cannot be eliminated from the handicap formula, then they should be made a function of the handicapper's judgment and should not be published before the event, except in a very general form.

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Discussion

E. W. CLEVELAND.[3] There is but one reason for handicapping, and this is for competitions. The handicap permits a larger number of competitors to enter an event.

In previous years the racing events scheduled by the National Air Races were based on cubic inch of piston displacement. This could not be entirely equitable to the motor manufacturers, and in free-for-all events, after scheduling a 1000 cu in. event, the others above that were practically duplicates. In 1932 an attempt was made to group the A.T.C. races according to top speeds. The races grouped in accordance with the advertised top speed proved that the pilot is an important factor in handicapping, and coincides with Mr. Edward P. Warner's ideas on handicapping. However, one would be at a loss as to a method of working this factor into the handicap.

The National Air Tour supplied experience in handicapping and had the appearances of being a fair formula, but due to the wide range and purpose of the airplanes involved, it could not satisfy all competitors. Putting the formula into words: "The airplane which can take off the quickest and land the slowest, fly the fastest, with the most load and the least amount of horsepower." This approaches an ideal. However, the tour was supposed to have a semblance of reliability; therefore speed trials were run before the start, in the earlier days of the tour, and a schedule was set up requiring but 85 per cent of the speed obtained on the speed trials for cross-country flying. This was abandoned later as most of the pilots ran their motors wide open, and the tour took on a racing aspect, as it became a case of not only maintaining the schedule, but a direct competition as to which pilot could get to the next scheduled stop first. This again brings up the fact that, regardless of the formula, the pilot plays an important part. On all tours with which the writer has been associated, there have been contestants who were prone to wander off the course. This wandering about after the adoption of the leg speed into the formula resulted in the airplane losing points through no fault of its own. There were cases where an airplane with an earth inductor and compass would be followed by others whose pilots knew that the craft ahead was so equipped, and therefore could not be wrong as to direction, and were thereby led astray. Then there were cases where the pilot would miss the airport, although in sight of it, by enough to affect the total score. This comes under the heading of navigation, but in cross-country events it plays an important part, as usually the course is laid down without taking into consideration present-day facilities furnished by the airways.

The tour formula is more of an efficiency formula than one to demonstrate reliability, although a remarkable degree of reliability was obtained. Only on rare occasions was the scheduled take-off deferred, and with but few exceptions all contestants finished, and in two or three cases the finish was in the face of adverse weather.

It is too bad that Professor Altman could not have accompanied the tour on its various trips—an explanation as to why the speed in 1930 was greater and also the stick time in seconds greater might have been forthcoming. The writer's explanation as to why the 1930 tour showed a superiority over the 1931 tour in speed was favorable winds over most of the distance flown. Variables are bound to creep into each handicap, as shown in a case where seven airplanes, all of the same make and horsepower, competed in the same event, starting from a racehorse start on a $3^1/_2$-mile course, six laps being flown, and there was a variation in range of speed of 1:46.79 between the fastest airplane and the

[3] Automotive Division, Cleveland Pneumatic Tool Company, Cleveland, Ohio; Chairman, Contest Committee, 1931-1932, National Air Races.

slowest. This proves that certain methods of mathematical handicaps are reasonably correct as to the equality of its effects on the airplanes involved. Following are the results of the purely handicap event, No. 8, flown over the $3^1/_2$-mile course, four laps, the handicap being absorbed at the start. The handicapping was figured mathematically by three people, and the values of K submitted by each member of the handicap committee were checked against each other and a compromise was reached:

Plane	Finish time
92, Taylor Cub	13:39.61
57, Waco	13:48.50
59, Monocoupe	13:50.78
70, Aero Klemm	14:05.77
23, Monocoupe	14:07.71
31, Aristocrat	14:09.71
99, Monocoupe	14:18.56
49, Heath Parasol	14:29.03
103	Out on first lap

Event No. 8 was a handicap for open or closed airplanes, carrying Department of Commerce commercial license, powered with an engine of not more than 510 cu in. piston displacement. All entries were handicapped by a handicap committee, and the announced handicap was absorbed at the start of the race.

As evidence that classifying airplanes according to the manufacturers' advertised top speed is superior to the old method of grouping according to cubic inch piston displacement for A.T.C. airplanes, there is the following comparison: Event No. 11, 1932 National Air Races, C or NC planes, 140-mph basis, 6 laps on a $3^1/_2$-mile course: The difference in speed between the first plane to cross the finish line and the fifth was 2:26.09. The corresponding A.T.C. event in 1931, based on a cubic inch piston displacement, showed a variation of 3:22.85 between the first airplane and the fifth.

In some cases the manufacturer's advertised top speed is overrated, in which case a handicap would be unfair. Also there is a variation in the speeds obtained by different pilots. From the writer's experience, the same speeds cannot be obtained from the same airplane operating under full throttle in level flight consistently, as the same airplane has been found to vary as high as 5 miles per hour when flown over a speed course on five different occasions in as many days. This variable is hard to compute and can only be reckoned with by taking the average.

One thing noticeable in the speed tests covering the 1932 Handicap Air Derby was the slight variation between the test pilots' speed as recorded by carrying a stop watch strapped to the pilot's leg and the speed as recorded by the camera. A total of 28 ships of 14 different makes were speed tested with a maximum variation in average speed of 2.3 miles per hour as the greatest difference, which is also evidence of the fact that elaborate timing apparatus is not particularly required to obtain very satisfactory results.

George B. Patterson.[4] Regarding the quoted statement that the original inclusion of stick and unstick times in the National Air Tour formula was "a stroke of genius," it is believed that the genesis of this formula was a remark by Major R. W. Schroeder, "The best airplane is the one which will carry the largest load the fastest, at the least expense, and will get off and on."

As to measuring performance for computing multi-motor handicap allowance, the writer differs from the statement that "A standard method of correcting altitude readings for temperature and pressure changes should be adopted. For this Diehl's correction has been suggested."

The methods of performance testing and computation which the writer developed in 1918 and 1919, and which are the basis of the present methods of the Army, Navy, and N.A.C.A., were designed to permit quantitative comparisons of performances measured under widely varying atmospheric conditions. In the present case, qualitative results are desired, and the tests are run so nearly simultaneously that atmospheric conditions are practically identical. A stop-watch and a sealed sensitive aneroid (the same instrument being used for all tests) should give sufficiently accurate results for handicap purposes. Of course, no high-altitude performance tests should ever be made when conditions such as temperature inversions or scattered cumulus clouds indicate the presence of vertical air currents.

In cross-country derbies and closed-course races where skill is to determine the winner, a wind allowance should be included in the handicap. Assuming all airplanes in a straightaway race to be handicapped so that all would reach the finish line together in still air, a tail wind would give the race to the slowest airplane, while a head wind means victory for the fastest. Any wind during a closed-course event favors the fastest airplane. In other words, the slower the airplane, the more it is affected by wind.

Application of the wind-correction factor can be greatly simplified by using previously prepared charts which take account of wind speed and direction and the shape of the course. This method was used for the first time at the National Air Races this year with promising results.

Carl F. Schory.[5] Handicapping is justified only where it is the desire of the race management to encourage a sufficient number of entries to warrant a contest which could not be otherwise organized on a free-for-all-class basis. Handicap contests have been generally unsatisfactory from the viewpoint of the contestants, as obviously all contestants other than the winner are inclined to the feeling that they would have won had the handicap been more favorable to their particular entry.

Author's Closure

Answering Mr. E. W. Cleveland's discussion, the author agrees that stop-watch timing is sufficiently accurate for slow-speed airplanes, but where the speed succeeds 160 mph and the trials are conducted over a mile course, a more accurate method should be used. Mr. Cleveland's reference to the close agreement between the stop-watch and elective timing in the speed trials of the 1932 Transcontinental Derby refers to tests on slow-speed airplanes. The maximum speed in these trials did not exceed 150 mph and the average speed was about 120 mph.

Mr. Patterson's statement referring to a correction for wind velocity and direction in the handicap of events based on speed is an important one and should be included in all such events. Referring to the statement that standard altitude correction for atmosphere conditions is not necessary, in the 1930 National Air Tour the multi-motor altitude tests were conducted at a temperature of over 90 deg. The specified test altitude was based on standard atmospheric conditions, and in order for most of the multi-motored airplanes to complete these tests successfully it was necessary to correct the altitude performance for temperature on a basis of an equivalent pressure altitude. After the correction was determined, the tests were conducted with a sensitive altimeter and stop-watch, as Mr. Patterson suggests. A barograph was used as a check in some of the tests. Various methods are available for this correction, but in any contest one method should be specified. When the time to conduct these tests is limited, it is not usually possible to wait for correct weather conditions.

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AER-55-15

Mechanical Equipment for Handling Large Rigid Airships

By C. M. BOLSTER,[1] LAKEHURST, N. J.

The paper describes the development of the equipment used at the Naval Air Station at Lakehurst, N. J., for docking and undocking the U. S. S. *Akron*. The main feature of the plan is a stern handling beam or long horizontal girder lying athwartship beneath the stern of the airship, holding the ship from each side with a cable system from heavy trucks moving along outward-lying tracks, and a floating support resting on a castoring taxying wheel and placed under the reinforced lower fin. This stern handling beam moves out of the hangar following the mobile mast to which the airship is moored. When a circular track is reached, the airship is positioned in the proper wind direction.



THE development of the equipment used at the Naval Air Station, Lakehurst, N. J., for docking and undocking the U. S. S. *Akron* was undertaken during the fall of 1930. At that time it had been decided that the bow of the ship would be held by a mobile mooring mast of the rail type—that is, built to travel on railroad tracks. However, it was not known what type of equipment would be used for holding the stern.

Experience in the development and use of the partial mechanical handling equipment used in docking and undocking the *Los Angeles* had produced several definite ideas as to the desired features of mechanical docking equipment. These ideal characteristics were used as a guide in the development of the equipment for the U. S. S. *Akron*. Briefly, those which appeared to be most desirable at that time were:

(*a*) That when leaving the hangar the airship should be held securely from both vertical and lateral movement relative to the mechanical handling gear until the ship is clear of the hangar and the stern has been swung with the prevailing wind direction. Conversely, when entering the hangar the ship must be held securely from the time it is connected to the mechanical handling gear on the landing field and during the time the stern is hauled around against the wind to a direction parallel to the hangar and while the ship and the mechanical handling gear, including the mobile mast, enter the hangar.

(*b*) The system should be so devised that there would be no necessity for disconnecting one set of equipment and connecting up another after the stern had been hauled up against the wind. Such a condition would exist if a system were used by which the stern of the airship were hauled around by a locomotive or a winch with a long bridle and when parallel to the hangar be connected to ordinary docking trolleys. This shifting from one system to another at a time when the loads were the greatest was considered both dangerous and undesirable. Such a system also did not hold the ship from both sides while hauling up the stern, which is essential in shifting winds.

(*c*) The equipment should be of such a type that the time taken to shift from the hauling up or swinging condition to the docking condition would be as brief as possible.

(*d*) Provision must be made in the equipment for supporting the stern of the ship in such a manner that the castoring taxying wheel could be used to hold it above and clear of the stern handling equipment after the equipment itself had been disconnected. This was essential, since after the ship had been taken out of the hangar and had swung to the prevailing wind direction, it had to move across the landing field to a mooring and riding-out circle which was about 2000 ft away from the hangar.

(*e*) The height of the stern handling equipment, including the hauling-up equipment, must be kept well below the bottom of the lower fin when supported on the taxying wheel in order to avoid any possibility of the fin striking it while the ship was swinging freely before it was connected to the mechanical equipment.

(*f*) The original design of the *ZRS-4* (*Akron*) provided for six reinforced points on each side of the after part of the ship, two at frame 0, two at frame 17.5, and two at frame 35. The mechanical handling equipment had to be devised in such a manner that it could utilize these six attachment points.

(*g*) The *ZRS-4* also had been designed in such a manner that the vertical load on either method of ground handling (that is, a taxying wheel or mechanical handling attachments) had to be applied through a reinforced area directly beneath frame 17.5 in the lower fin. The equipment therefore had to support the fin at this point.

(*h*) Since this particular equipment was being devised for use at Lakehurst, it had to be so designed that it would permit docking or undocking the *Akron* with the *Los Angeles* in the hangar. Owing to the limited space available, this complicated matters somewhat.

It was fairly obvious that, whatever type of stern handling equipment was to be used, it should travel during the hauling up or swinging period on a circular track, the mobile rail mast providing the center of the circle about which the ship could swing. It was also realized that an extremely powerful locomotive of low height and delicate control would have to be available for hauling up or swinging the stern of the airship.

The first thought as to the type of equipment to use for holding the stern was to provide heavily weighted cars on each side of the ship which would act somewhat as docking trolleys, except that they would be capable of holding the side load of the airship by their weight and the flanges on the wheels. A third car was to be used to support the lower fin. It was thought that some system could be devised whereby these heavily weighted side cars and the center supporting car could be shifted or transferred from the hangar tracks to the circular or hauling-up track when the ship had reached the center of the circle and was ready to swing. During the period that these heavily weighted side cars and the center car were on the circular track, the use of long compression members tying the three together was contemplated.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Some time was spent on the development and study of this system, and several difficulties and complications arose. It had always appeared that the simplest and most dependable method by which these heavily weighted cars could be shifted from the hangar tracks to the circular tracks while under side load would be by means of some method of vertical jacking whereby one set of wheels would be lifted off as the other set made contact.

This immediately resulted in complications due to the fact that one car could not move vertically relative to the other without greatly altering the tension in the side cables. Another undesirable feature of this method was that it involved the construc-

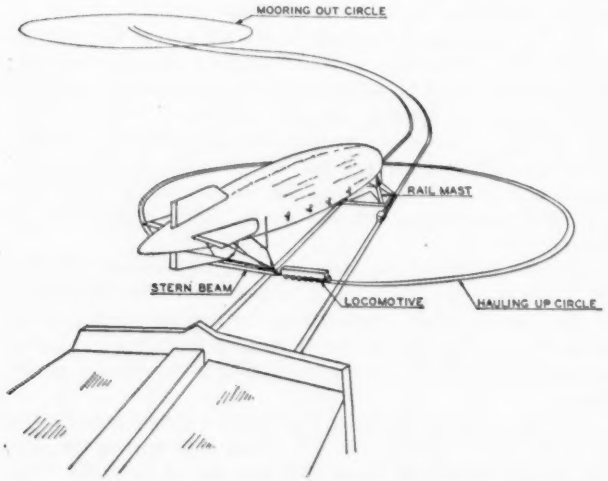


FIG. 1 GENERAL PLAN OF HAULING-UP AND MOORING-OUT CIRCLES

tion of three more standard-gage railroad tracks completely through the hangar and for a distance of about 1200 ft out on the landing field. In other words, our trackage system would have consisted of five sets of standard-gage railroad: one in about the center of the ship's path on which the lower fin support would travel, the two which already existed at a gage of $64^1/_2$ ft on which the rail-type mobile mooring mast was to travel, and two more—one at the extreme edge of the hangar on the side of the *Akron* berth and one on the other side as far over toward the *Los Angeles* as possible on which the heavily weighted side handling cars or trolleys were to travel. The expense and the difficulty of laying these three additional railroad tracks were considerable. It also appeared that there might be a dangerously high concentrated side load on the rails which supported the side handling trolleys under the 63,000-lb side load upon which the design was based.

It was while trying to improve on this system in such a manner as to permit the use of the two tracks through the hangar on a $64^1/_2$-ft gage which already existed that the idea of the stern handling beam was evolved.

Basically, the stern handling beam consisted of a long beam which reached out to the two end points where the side handling trolleys were intended to travel while utilizing the two railroad tracks which already existed through the hangar. It also gave the desired center support without the necessity for a track at that point. It thus provided the three points, two end and a center point, without the cost of laying any more track through the hangar. In order to investigate the possibilities of such an arrangement, a model was made of the proposed stern handling beam, and a small section of hangar track and a circular track were constructed on the same scale as this model. The whole arrangement was used with a small airship model that we had at Lakehurst. Work with this model showed that the stern handling beam system was perfectly feasible and that it should provide a suitable type of mechanical handling equipment for the stern at relatively low cost.

There was one interesting complication, however. It was found while checking back with models that it would be impossible to bring the *Akron* in the hangar with the stern and bow concentric to the existing hangar tracks without the tail of the *Akron* striking the *Los Angeles*. This was solved by moving the stern of the *Akron* 10 ft 3 in. off the center of the existing tracks. The *Akron* could then enter or leave the hangar and still have ample clearance between the hangar wall and the *Los Angeles*.

Such an arrangement meant that a beam must be provided for supporting and holding the ship at two possible points each 10 ft 3 in. from the center line of the existing hangar tracks. This was necessary because the ship set toward one end of the beam when entering the hangar stern first, while when entering bow first the beam had to be swung 180 deg, and the stern therefore had to be moved toward the other end of the beam. This complication meant a dual installation of the stern supporting points and of the two side handling points. It did not, however, destroy the feasibility of the beam arrangement.

In checking back on the characteristics which were deemed most desirable for the stern handling equipment, it was found that the stern handling beam adequately met all of them:

(*a*) It held the airship securely from both vertical and lateral movement relative to the mechanical handling gear until such time as it was desired to disconnect it. (*b*) Having once connected the ship to the stern handling beam, there was no further necessity for connecting or disconnecting any other system. (*c*) The period of transfer from the hauling-up circle to the hangar tracks was reduced to a relatively short time. (*d*) The castoring taxying wheel was attached to the ship throughout the operation and was available for use in holding the ship clear of the mechanical gear whenever required. (*e*) The height of the stern handling equipment, both beam and locomotive, could probably be kept within the 6-ft over-all height required. (*f*) The six attachment points on the ship itself could be utilized in holding the ship. (*g*) The vertical load from the ship to the stern handling beam was applied through the reinforced point at frame 17.5. (*h*) The equipment permitted docking or undocking with any direction of the ship in the hangar, with any wind direction, and with the *Los Angeles* in the hangar. Thus all of the requisite features were adequately met and preliminary drawings and specifications were prepared.

The stern handling beam was constructed by the Wellman Engineering Company, of Cleveland, Ohio. Complete with all winches, trucks, and miscellaneous equipment, it weighs about 178,000 lb. The main strength member consists of a steel built-up girder of two heavy I-sections, with ample section modulus for the bending loads involved. These I-sections are 42 in. deep and are mounted parallel to each other about 7 ft apart. The sections are tapered at the ends and suitably braced laterally. The resulting girder is 186 ft 6 in. long and lies horizontally in such a manner that the 42-in. dimension is vertical.

When traveling in and out of the hangar, the beam is supported on four four-wheeled trucks, two toward each end, each pair traveling on one of the two railroad tracks laid through the hangar on a $64^1/_2$-ft gage. These trucks are well braced to take the side load and are rigidly mounted. The wheels themselves are of a roller-bearing type and are only 14 in. in diameter. Special flanges are used on the wheels for taking the side load, the flanges being machined from rolled steel.

For traveling on the circular track, the beam is supported by

two four-wheeled trucks located outboard of the hangar trucks and built to conform to the curvature of the circular track of 650-ft radius on which they travel. The axles which carry these wheels are supported by a power-driven screw linkage mechanism which permits the axles and wheels to be moved vertically 8 in. relative to the beam. This movement is so arranged that while the beam is traveling on the hangar tracks the circular track wheels are in their top position, or are 4 in. above the level of the ground and hangar tracks. When the beam has reached the circular track and it is desired to shift from the hangar track to the circular track, the circular track wheels are jacked from their top position to their bottom position. In so doing the circular track wheels have landed on the circular track at the mid-point of their travel and have then lifted the beam, ship, and hangar track wheels 4 in. upward. The beam has thus been transferred from one set of tracks to the other and is perfectly free to swing on the circular track. This operation is performed with brakes on the circular track wheels and with the hauling-up locomotive connected up in order to prevent the side load on the airship from running away with the beam as soon as the hangar track wheels have disengaged. This jacking operation is done by a gasoline engine and requires less than a minute.

While traveling in and out of the hangar, the beam is pulled by the ship itself through suitable towing cables. The ship is moved by the mobile rail mast, which is provided with a gasoline-electric locomotive. The decision to tow the beam in this manner was based on extensive towing tests conducted with flanged wheels under side load and towing tests of the beam itself under high side load. These tests showed that the pull required to start the beam under extremely high side load was about 5000 lb, and that the running load was only about half this amount. Although this method has functioned satisfactorily, it is considered desirable to remove all unnecessary loads from the ship's structure, and at the present time a system of interconnecting members traveling on small trolleys is being developed which will rigidly connect the beam to the mast.

When swinging on the circular track, the beam is connected to a gasoline hydraulic locomotive especially designed for the purpose which controls the beam and the ship during this period. This locomotive is a most interesting mechanism. It was designed and built by The H. K. Porter Company, of Pittsburgh, Pa., and is probably capable of exerting a greater drawbar pull than any locomotive in the world of comparable size and power. It is 35 ft long, 6 ft overall in height, and conforms to the standard railroad requirements as to width. It weighs 266,000 lb and is driven by a 250-hp Leroy engine, which drives the wheels by siderods and gears through a Waterbury hydraulic transmission unit. In low gear this locomotive develops a drawbar pull of 63,000 lb at 94 ft per min. and due to the hydraulic transmission it has remarkably delicate control, speed being capable of reduction to such a creep that the motion is hardly perceptible.

The original design of the *ZRS-4* (*Akron*) provided six reinforced attachment points on each side of the after part of the ship for attachment of the side handling cable system. Suitable bridles were made up which would secure to these points, the bight of the bridle hanging down to where it could be reached from the ground. Large snatchblocks were procured which could be quickly attached to these side handling bridles, and a cable system was provided which connected the beam to the ship's bridles by means of a continuous cable anchored at one end and rove back and forth between the snatchblocks and sheaves on the beam and then to a power-driven winch on the beam. The motors which drive these winches are mounted on the after side of the beam out of the way of the taxi wheel and cradle. A $^{9}/_{16}$-in.-diameter extra-flexible cable having a breaking strength of about 30,000 lb is used, and the system is so arranged that each side of the ship is held by from six to eight strands of this cable, the tension in each strand being approximately the same except for the slight friction in the sheaves and snatchblocks. The sheaves which restrain these side handling cables are mounted on removable pins placed at two possible locations on each end of the stern handling beam. Two of these locations are at the extreme ends of the beam and the others are 20 ft 6 in. in from the ends, thus providing the two possible attachment systems required by the necessity of moving the stern of the *Akron* off the center of the beam itself.

In connecting up or disconnecting the beam cables with the ship's bridles, the beam cable is slacked by the power-driven winch until the bridles can be reached from the ground, at which

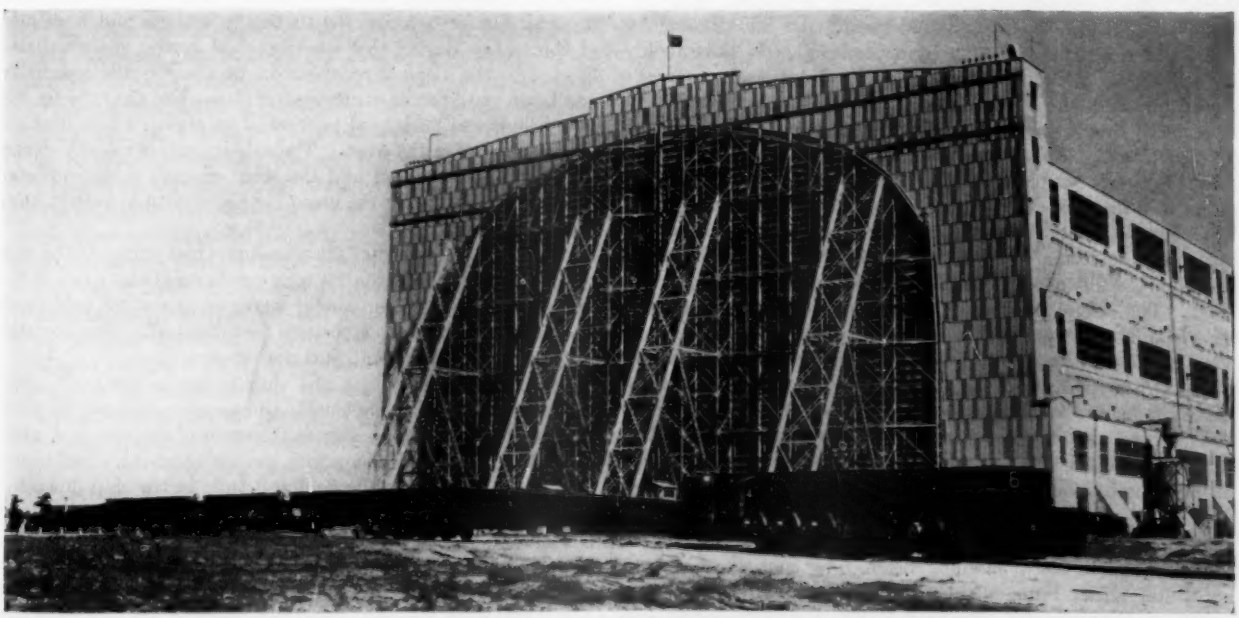


FIG. 2 STERN HANDLING BEAM AND LOCOMOTIVE

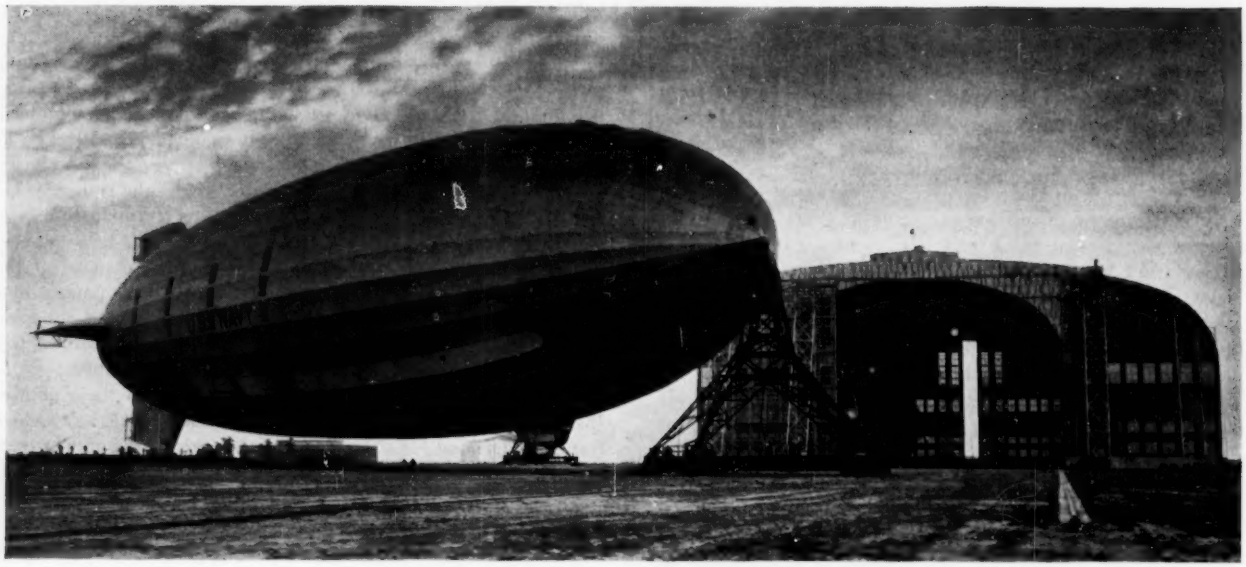


FIG. 3 THE "AKRON" BEING HAULED UP TO ENTER THE HANGAR

time the snatchblocks are removed. Quick-release devices are also introduced into the cable system between the ship's bridles and the beam cable in order to permit rapid disconnecting of the side handling cables in gusty weather. Due to the whipping action of the cables under tension, these quick-release devices are normally never released until the cable has been slacked down. It will be noticed that this cable system, when connected up, holds the ship from both sides, and that once the ship has been placed on the beam and the side handling cables are set up, the ship is held from any side movement relative to the beam. The ship, however, is not restrained from rolling.

In providing the support for the lower fin, two objectives were carried out. The first was that the support for the lower fin should be so arranged that the castoring taxying wheel could be left in place on the ship while the ship was attached to the beam. This attachment is made in such a manner that the beam support can be moved underneath the fin attachment point and raised mechanically by a powerful jack, thus avoiding any necessity for lifting the ship by hand during the connecting-up period. The support for the lower fin is so made that it is perfectly free to move laterally in order that the ship could be free to roll. This is essential in order to avoid side loads on the bottom of the lower fin. The support also is free to move 36 in. in a fore-and-aft or radial direction relative to the beam. This fore-and-aft freedom is necessary in order to allow for inaccurate centering of the mast relative to the circular track and to allow for inaccuracies in the radius of the circular track itself. The lateral freedom of the fin support removed all side load from the bottom of the fin, the ship absorbing the rolling effect from the side load by rolling over until the couple due to the static stability of the ship equals the rolling moment of the wind load. Actually this rolling is not as great as might be supposed and should normally not exceed 4 or 5 deg. The actual connection between the supporting trolley and the lower fin is made by clamping a swiveling yoke on the beam trolley to a cradle which has previously been secured to the lower fin. The castoring taxying wheel is secured to this same cradle.

The complete stern handling beam thus consists of a long horizontal girder lying athwartship beneath the stern of the airship, holding the ship from each side with a cable system and supporting the ship on a floating support through the reinforced portion of the lower fin. This girder or stern handling beam moves out of the hangar with the airship, and when the mobile mast reaches the center of the circular railroad track it stops. The beam is connected to the stern locomotive, and with brakes on is transferred to the circular track by means of the jacking trucks. The stern of the airship is then swung around until the ship is headed approximately into the prevailing wind. It will be noticed that the airship may be swung in this manner through any required angle, and the ship may leave the hangar either bow first or tail first with any wind direction and can be swung through any angle up to or beyond 180 deg if required.

When headed approximately into the prevailing wind, the stern cables are partially slacked off, and it is immediately apparent whether or not the ship is actually in the wind by movement of the ship relative to the beam. This is most evident by lateral movement of the fin-supporting trolley along the beam. A signal device is connected to the fin trolley, and movement of the trolley during this slackening off period is immediately relayed to the locomotive operator, who moves the locomotive and beam until the signaling system shows him that the fin support has moved back to the center of its travel, which indicates that the ship is in the wind. The cables are now rapidly slacked down and disconnected and the stern support is disconnected, the locomotive moving the beam back and forth to keep it under the ship in gusty weather.

As soon as disconnected, the taxying wheel supports the after part of the ship, and the beam is run around the circle out of the way as the mast and airship move on across the field to the mooring-out circle or to a position for taking off. The operation for docking is, in general, just the reverse.

During the period that the ship is being "walked" across the landing field, a single handling-line spider, manned by about 30 men, is used on each side of the stern of the ship. A group of about 12 men walk along with the after taxying wheel to help it over rough spots in the field and to hold the ship down if it becomes light. These men also assist in holding the ship while it is being connected or disconnected from the beam. They perform no function during the hauling up and docking operation, and this may truly be said to be entirely a mechanical operation, since the only personnel involved are the eight men on each side who connect the side handling cables, the group

mentioned who connect the lower fin support, the three winch operators, and the locomotive operator.

Actual use of the stern handling beam and rail mast in docking and undocking the *Akron* required the solution of several incidental problems. The most important of these were:

(*a*) When leaving the hangar how best to stop the mast so that the beam wheels could be lined up with the circular track rails. An automatic stop-light system was first tried, but did not work satisfactorily. Experience showed that a hand-operated stop light at the beam and seen by the locomotive operator at the mast gave a better control, and such a system works fairly well. The beam usually stopped within 3 or 4 in. of the required point and is moved along the track by suitable equipment through this distance before the jacking operation is begun. Further improvement of this feature is now under way and will be included in the improved type of towing equipment.

(*b*) In hauling up the stern it is necessary to stop the beam when the hangar trucks are lined up with the hangar rails. Due to the ease with which the hauling-up locomotive can be controlled, this operation is a simple one. It has now been improved to the point where the locomotive operator stops his locomotive at a predetermined point which automatically lines the wheels up as required.

(*c*) One of the most difficult features of the beam handling operation is that of connecting up and disconnecting the side handling cables, especially in gusty weather. This operation makes it essential that the beam be kept fairly well centered under the ship during the period that the side handling cables are slacked. The signaling system which controls the locomotive operation by movement of the fin and fin trolley relative to the beam has proved to be the solution of this problem and has worked out fairly well. A system of automatic control would be even better and can undoubtedly be included in future designs. Actual connecting and disconnecting of the snatch-blocks themselves is a matter of experience, and with properly drilled personnel can be performed very rapidly. Under such conditions it requires about 2 min to disconnect the blocks and from 4 to 5 min to connect them up.

Several important lessons have been learned during the relatively brief experience with this handling equipment. Undoubtedly many more will be learned in the future. Briefly, these lessons may be summarized as follows:

(*a*) Mechanical docking and undocking is entirely feasible, and the stern handling beam and rail-type mobile mast provide suitable equipment for this purpose. However, as shown by the accidental breaking away of the *Akron* from the stern handling gear on February 22, 1932, there are definite limitations as to safe docking and undocking conditions imposed by the strength of the airship structure. Present indications are that the maximum permissible cross-hangar wind velocity for docking and undocking the *Akron* is about 20 miles per hour. This is of course a great improvement over non-mechanical docking methods. However, this limitation must be just as carefully adhered to as was the old limitation of 5 to 6 miles per hour for manual handling in former days. Such a limitation will always depend on the strength of the ship's structure, for the mechanical handling gear can be designed to withstand any loads that the ship can carry.

(*b*) Apparently a quartering wind imposes greater loads on the stern handling gear than a beam wind. Accurate confirmation of this fact will be obtained in the future. At any rate, undocking stern first in a quartering wind probably requires the best weather conditions of any maneuver.

(*c*) The stern handling equipment should be permanently equipped with suitable dynamometers to indicate the load imposed on the ship at all times. Loads existing at any time can then be observed and the maneuver be stopped before dangerous conditions arise. At all times, but especially in gusty weather, there seems to be little relation between observed wind velocities and loads imposed on the ship. A dynamometer will give the true story as to the load which the ship is carrying, which is the important feature of the operation, the wind velocity of course serving as a guide as to whether such an operation should be begun.

(*d*) Although actual breaking away of an airship from mechanical handling gear should normally be a rare occurrence, it seems desirable that every precaution be taken to minimize the danger to the ship when such an accident occurs. To accomplish this it would probably be desirable to have the pendants which secure the side handling bridles to the ship's structure the weak link in the system, so that these pendants will break before the ship's structure will fail, and at the same time any whipping of the heavy cables against the ship's structure will be avoided.

(*e*) It would be desirable to have fewer attachment points on the airship, and thus fewer bridles and less equipment to connect up and disconnect. This is a matter of airship design and local reinforcement of the airship structure. Undoubtedly, considerable progress toward reduction of the number of these attachment points can be made in future ships. If this means a reduction in permissible loads which the ship can withstand while being handled mechanically, it would not be desirable, for the slight advantage gained in making the handling operation an easier one would not be sufficient to warrant a reduction in the strength of the ship and equipment.

(*f*) The use of a type of handling equipment such as the stern handling beam, in which all parts are below the ship, is still considered a vital feature. Use of vertical booms or masts which would attach to reinforced points well up on the side of the hull on each side has been proposed several times, but it would seem that in case of unexpected high winds which cause the ship's structure to fail, the damage resulting to the hull would be much more severe in the case of the side masts than in the use of cables, since they would cut completely through the hull at that point. It may be feasible to design such booms so that they will fall down out of the way when a breakaway occurs. However, at the present moment it is difficult to see how such a boom can be relied upon to hold in tension from one side and in compression on the other and at the same time release itself should the ship's structure break. Any type of manually operated quick-release equipment for such an emergency is of little or no value since the break will occur before any one can act. Danger of damaging the ship while connecting and disconnecting would also be ever present unless the booms were used in combination with a cable system. A most desirable improvement in the low type of equipment such as the beam, which holds entirely in tension from the windward side, would be to have the fin-supporting trolley and leeward cables automatically releasing under conditions when the ship breaks away. Such features are believed to be perfectly feasible.

(*g*) It would be desirable to have the hauling-up locomotive built into the stern handling beam as an integral part of it to avoid the unnecessary complication of coupling and uncoupling it and to reduce the necessary pieces of equipment. Such a design was considered when the beam was designed, but at that time it was felt that due to the experimental nature of the whole project it was much more conservative to procure the locomotive and beam separately, since the locomotive could be used for hauling up the stern whether the beam worked properly or not.

(*h*) The improved towing methods, removing all towing loads from the ship's structure, are considered desirable.

(*i*) A two-ended hangar with circular tracks at each end would be desirable for an ideal arrangement. Undocking of an airship stern first out of the windward hangar door is a difficult and hazardous operation, the danger of which would be considerably reduced if the ship could go out of the leeward door and swing on a circle at that end of the hangar.

Based on our experience thus far, it appears that the ideal stern handling equipment for handling large rigid airships should consist of a stern handling beam much like the present one except improved in arrangement, as already outlined. At a location where meterological forecasts can be relatively accurately made (that is, where desirable weather conditions for docking can be depended upon for an hour or so) it would be desirable to move the hauling-up circle a considerable distance from the hangar and combine the hauling-up circle and mooring circle into one circular track. The beam for such a layout would be changed to permit the fin-supporting trolley to travel directly upon the top of suitable trackage along the top of the beam, and when the ship had left the hangar and had been swung into the prevailing wind, transfer of the ship from the beam to the riding-out car could be made entirely a mechanical one by movement of the trolley along the top of the beam to the top of the riding-out car. In this case the trolley which supported the lower fin would have to permit some movement of the fin relative to this trolley, and this movement would in turn automatically control the movement of the beam and would thus move it out from underneath the ship as the trolley transferred from its position on the beam to the riding-out car.

For locations where weather conditions are changeable and difficult to forecast, it seems very desirable to have the hauling-up circle as close to the hangar as possible in order that the period during which the ship is held against the side-wind load will be reduced to a minimum. For such locations, a beam like that now used at Lakehurst except for the slight improvements mentioned would be preferable. Such an arrangement, of course, will always require a small ground crew, since during the period that the airship is being "walked" across the landing field a small detail of men will be required to hold and control the after part of the ship.

Thus it appears that at an airship base where relatively good meteorological conditions exist (namely, where maximum winds of 20 mph or less can be assured for an hour or so) the docking and undocking operation can be made almost completely a mechanical one with only a few (perhaps 20 or 30) men required for mooring, riding out, or entering the hangar. For unfavorable locations, although the undocking itself can be made mechanically, a small ground crew will always be required for "walking" the ship across the field.

Discussion

W. Klemperer.[2] The docking and undocking of large rigid airships by man-power has been looked upon by many as not in keeping with the trend in mechanization in this machine age. Although it may be argued that even the docking and undocking of ocean liners between piers with the aid of tugs and a considerable number of men handling ropes on shore is even today not completely mechanized, there is no doubt that substitution of machines for man-power in handling large rigid airships will increase the safety for men and equipment, the reliability of operation, and the economy of commercial application of airships.

The motorization of the bow mast has been accomplished for some time. However, the holding of the aft part of the ship by mechanical means offered considerably greater problems, and it has been extremely gratifying to all interested in lighter-than-air craft to watch the remarkable progress made at the Naval Air Station at Lakehurst, where undoubtedly the greatest experience along this line has been concentrated.

One is impressed that one of the most ambitious parts of the problem is to maintain the proper distance between the bow mast and the stern handling gear without throwing stresses into the ship's structure, considering the gear having the mass of 180,000 lb.

The author has pointed out that mechanical links connecting the tail beam with the bow mast have been considered, and mechanical links have already been successfully used between individual docking trolleys in handling the U. S. S. *Los Angeles*. However, the alternative idea of providing the stern handling gear with a power drive and servo-control obviously has also suggested itself. It is understood that any such device is looked upon as too delicate and dangerous to be relied upon entirely. However, one may visualize the power drive being servo-controlled with or without the aid of a human element from a moderate permissible movement between the beam proper and the ship's support substructure, any displacement of the two movable parts, fore and aft, activating the beam motors in such a direction as to reduce this displacement. Such an arrangement would act as a brake as well as a reversible motor.

Evidently the system would have to be fully operative with the servo-control drive out of operation, and the latter would only serve to keep the routine stresses away from the airship. Whether the expense involved in this motorization would be justified by this limited usefulness is a matter of opinion and experience.

It is of course of great interest to accumulate data of actual handling forces in order to compare them with those assumed under more or less fictitious meterological conditions. So long as airships were handled mainly by man-power, the introduction of dynamometers was always a disagreeable complication of a delicate operation. Now with mechanical means measuring is greatly facilitated, and it is noted with great gratification that the introduction of permanently installed force recorders in all of the important load-carrying members is contemplated by the Navy.

Although the author has vividly pointed out the disadvantages of any stern handling structure with solid parts extending high above the ground, there is one such alternative which has been given serious attention. This is a single stern mast on which the stern of the ship would be moored somewhat similarly as the bow is on the bow mast. In this case, our studies have shown that the stern attachment could possibly be worked out so that it permits some relative motion and so that, in case of breakaway, the stern could swing clear of the structure. Then the mast need not be collapsible.

Of course, a mutual fore-and-aft tow system with provision for ample permissible relative motion or an efficient and reliable servo-drive equipment must again be worked out. Neither solution seems to offer insurmountable difficulties. A stern mast equipment requires some strengthening of the stern of the ship, which investigations have shown would not necessarily be prohibitive in weight.

Most of the difficulties in docking or undocking large rigid airships arise from sudden wind shifts encountered more or less against anticipation during the docking operation. Obviously it will mean much for the elimination of unnecessarily high handling loads when elaborate automatic wind-shift warning systems are installed at airship ports. A number of automatic wind velocity and direction recorders placed at strategic distances around the airport, especially on the weather side, sufficiently far away to give at least a quarter of an hour's warning, will enable the docking officer to change his plans before a sudden emergency arises.

[2] Goodyear Zeppelin Company, Akron, Ohio.

Author's Closure

The suggested motorization of the stern handling equipment to provide movement in and out of the hangar, keeping it properly spaced away from the mast by some type of servo-control, is not considered a safe solution to the problem. It is believed that one of the most important basic principles for safe operation is that while going in and out of the hangar there should be only one source of power. The soundness of this principle has been demonstrated twice at Lakehurst, when use of such a servo-control might have resulted in damage to the ship.

Since the original paper was written a system of interconnecting tension and compression members has been designed and built which rigidly holds the beam a fixed distance from the mast and is stronger than the full tractive effort of the mast locomotive. On one occasion while the ship was being walked out of the hangar, the beam fouled on a protruding part of the hangar floor and was definitely stopped against further motion. Using the interconnecting spreader-bar system of towing, this merely resulted in stopping the mast and airship, the locomotive spinning its wheels but doing no damage to any part of the ship or handling apparatus. On another occasion while operating in a snowstorm, the tracks were partially coated with ice and the locomotive wheels turned with no motion of mast, beam, or ship. Under such a condition the beam wheels might be properly controlled and receive ample torque, but no movement would result, although at that moment the mast wheels might be taking hold. This would result in movement of the mast relative to the stern equipment, with resultant damage to the ship. The spreader-bar system is simple and relatively inexpensive and is considered to be the safest arrangement.

The use of a single stern mast which would secure to a suitably reinforced point on the stern, much as the bow is secured forward, is considered to be an excellent alternative to the beam. The *Akron* and *Macon* are not provided with such a point, and the beam was designed to utilize the strong side handling points available. The stern mast probably would require some added structural weight in the tail of the ship and would be a more costly unit in itself in addition to the added trackage required. It has the important advantage, however, that it removes all handling stresses from the lower fin. Such a stern mast, if used, should be on a track radius large anough to insure that it would be well clear of all parts of the airship as soon as disconnected.

AER-55-16

The Economics of Air Transport

By LOUIS BREGUET,[1] PARIS, FRANCE

In this up-to-date analysis of the economic problems of aerial transportation, the author presents a revision of the basic formula for calculating the net cost of air transport, gives examples of its application, and discusses the following problems: (*a*) Application to a simple case; (*b*) the importance of overhead expense; (*c*) the influence of speed on net cost; and (*d*) flight at high altitudes.

IN A PAPER read before the Fourth International Congress of Aerial Navigation, held at Rome, Italy, in October, 1927, the author presented a formula which permitted the calculation of the net cost of aerial transportation; he also discussed several examples of the application of this formula.

It now appears advisable, after an interval of four years, to raise the question again and bring it up to date, taking into account the progress made in the operation of air lines and the changes in the economic situation. Advantage will also be taken of this occasion to complete to a certain degree the main formula which the author previously proposed for the calculation of the net cost of aerial transportation.

I—REVISION OF THE BASIC FORMULA FOR OBTAINING NET COST

This formula gives the net cost X per *ton-kilometer of pay load available*. For scheduled operations with an efficiency of 100 per cent, the net cost is obtained by dividing the total expenses charged to each flying hour by the pay load and by the cruising speed per hour in kilometers.

For practical operation with an efficiency less than 100 per cent, the net cost is obtained by multiplying the cost of 100 per cent operation by a coefficient i, which takes into account irregularity of operation and sundry incidental expenses. This coefficient varies appreciably with operating conditions and the type of traffic available. Some years ago i had a value of approximately 1.30 on most routes, but it is now more usually between 1.0 and 1.10 for air lines which are efficiently operated and well patronized.[2]

The main formula in question has reference to the operation of an air line in stages of a practically uniform length E, flown by airplanes of the same type fitted with the same engine. The supply parks, exclusive of the supply of spare parts which are in stock at the main repair depots, will carry the following classes of spares:

Airplanes, excluding engines and their fittings
Fittings, accessories, and spares attachable to the bare airplanes
Engines, including their accessories, spares, and propellers.

A distinction is drawn between current maintenance, comprising cleaning, washing, and occasional repairs needed to keep the airplane in service, and major overhauls made at intervals as required.

Similarly, one should distinguish between repair or overhaul work done by the operating company (or by a third party paid by the company) and not covered by insurance, and the repair work caused by accidents and covered by insurance. The cost of the second type of repair work is covered in the coefficient i, being an irregular or incidental expense. The different categories of material include the equipment in regular service, the equipment "out of action," and the equipment in reserve to meet emergencies.

The depreciation of the material in each category (airplane or engine groups with their accessories and spares) should be calculated differently for equipment in regular use and for equipment "out of action" or in reserve. Equipment in use should be depreciated on a flying-hour basis, while that not in use should be written off on the basis of age in months or years, to cover obsolescence.

It should be appreciated that the same material passes by successive stages through the phases of active use, out-of-action, and reserve, the whole being the useful life of the equipment. The complete depreciation of equipment can be calculated from the numbers allotted to these different stages without considering individually the items themselves, which are for simplicity assumed to be all of the same type. It is admitted, on the other hand, that the initial cost value to be placed on an airplane or engine group for depreciation purposes must include the cost of accessories and spares; it should also include the financial charges incidental to purchase, e.g., the interest on capital locked up (either paid up or tied up) in the equipment.

Repairs or overhauls are made at intervals of H_p or H_m hours of running for planes and engines, respectively, and entail an expenditure which may be represented by a fraction K_p or K_m of the purchase price.

To sum up, the equipment costs can be broken down into:

a The cost of amortization of the purchase price, including interest on capital
b The expenses of repair or periodical overhaul (not covered by insurance) including the total expenses of the repair shops, and/or payments to third parties for repair work.

To these expenses must be added the costs of running the service, which comprise:

a Fuel and oil
b Pilots and crew, including salaries, flying pay, insurance, uniforms, etc.
c Airports, including ground personnel, executive, clerical, mechanical and accounting; hangar charges, radio, lighting, etc.

These expenses can be considered on an annual basis as proportional on the one hand to the tonnage of the fleet, and on the other hand to the product of the number of flights per annum multiplied by the pay load of the aircraft employed.

Finally, we must add to the foregoing the administration expenses, including legal and insurance costs, plus the expenses of the traffic department. The total of these last two categories of expense can be termed general charges, and will depend on numerous factors.

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[1] Director General and Founder, the L. Breguet Aviation Works.

[2] The net cost per ton-kilometer available, designated by X, does not take into account the actual pay load carried. Airplanes do not by any means always fly with a pay load equal to their maximum capacity; the pay load actually carried should be considered in the calculation of actual operating receipts to be set off against X, the net cost per ton-kilometer for operation at maximum capacity available.

Contributed by the Aeronautic Division and presented by the translator, C. H. Biddlecombe, New York University, before the Metropolitan Section of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, New York, N. Y., November 22, 1932.

In the case of a specific air line using specific equipment, we can consider these general charges as comprising two parts: one constant and the other proportional to the traffic. The latter will vary linearly with the number of trips flown annually on the scheduled service, the coefficients of the linear function depending principally on the number of stopping points on the route and the useful tonnage of the aircraft employed, or, more simply, of the total tonnage of these aircraft.

The sum total of the foregoing items represents the cost of operating an air line. Regarding the pay load FP of the aircraft employed, this is obtained by using a coefficient $(1 - a)$ where a represents the fraction of the disposable load absorbed by the cabin fittings and furnishings; the disposable load being the difference between the "all-up" weight P of the airplane fully loaded, and the "dead weight" represented by the sum of the weights of the bare airplane structure, the power plant group or groups, the crew, the flying equipment and/or instruments not otherwise included, and the fuel and oil with tanks and pipe lines. Finally, the cruising speed over the ground, as an average of all flights made over the length E of the trips, is generally a little less than the air speed V; it can be represented by γV, γ being a coefficient slightly less than unity, and which allows for head winds and the time lost in taking off and landing. The variable winds encountered in flight have very little effect on this coefficient, but it may be noted that in the formula the amount of fuel and oil needed for a given operation is calculated by using a factor v, representing an average wind always deducted from the air speed V.

Taking into account all the foregoing remarks, the formula which the author presented in 1927, now suitably revised, can be written in the manner, shown in the adjoining column, X being the actual cost of the ton-kilometer of pay load available, expressed in French francs as defined by the stabilizing law of 1928.

$$X = i \times \frac{100}{\lambda V}\left[\left(\frac{1}{M_p} + \frac{(n-n_1)\lambda V}{A_p N E} + \frac{K_p}{H_p}\right)x_p + \left[\frac{r}{M_m} + \frac{(n'-n'_1)V}{A_m N E} + \frac{rK_m}{H_m}\right]\frac{\chi V \tan\varphi'}{r \cdot 270}x_m + \left(\frac{100}{P} + 0.035\right) + \frac{V}{270}\tan\varphi' \cdot mx_c + p + \frac{\lambda V}{E}\left[g\frac{n}{N} + g'f\right] + \frac{\lambda V}{E}\left[\frac{G}{NP} + G'f\right]\right]$$

$$\text{Payload Capacity} = (1-a)\left[1 - \alpha - \chi\frac{V}{270}\tan\varphi' \cdot q - \frac{m}{270}\tan\varphi'\frac{V}{V-v}E - \left(\frac{130}{P} + 0.03\right)\right]$$

Purchase, Depreciation, and Overhaul of Equipment — Airplanes With Spares and Accessories — Depreciation — Service: $\frac{1}{M_p}$; Reserve: $\frac{(n-n_1)\lambda V}{A_p N E}$; Overhaul: $\frac{K_p}{H_p}$ — x_p

Engine Groups, Spares, Propellers, Etc. — Depreciation — Service: $\frac{r}{M_m}$; Reserve: $\frac{(n'-n'_1)V}{A_m N E}$; Overhaul: $\frac{rK_m}{H_m}$ — $\frac{\chi V \tan\varphi'}{r \cdot 270}x_m$

Operating Expenses — Flying Personnel: $\left(\frac{100}{P} + 0.035\right)$ — Fuel and Oil: $\frac{V}{270}\tan\varphi' \cdot mx_c$ — Running Maintenance: $p + \frac{\lambda V}{E}\left[g\frac{n}{N} + g'f\right]$ — Airport Expenses: $\frac{\lambda V}{E}\left[\frac{G}{NP} + G'f\right]$ — Administration and Traffic Expenses

Relative Weights — Cabin Fittings: a — Airplane Without Power Plant: $1 - a - \chi\frac{V}{270}\tan\varphi' \cdot q$ — Power Plant: $\frac{m}{270}\tan\varphi'\frac{V}{V-v}E$ — Fuel and Oil With Tanks: $\left(\frac{130}{P} + 0.03\right)$ — Crew and Equipment

France per Ton-Km Available

The letters and symbols employed in the formula are as follows:

- i = coefficient of increase to allow for irregular operation and certain incidental expenses, including insurance costs. The value of this coefficient ranges between 1.0 and 1.5, and is at present between 1.05 and 1.10 for air lines with efficient and regular operation
- M_p = depreciation of the bare airplane and accessories in hours of flying time. Its value varies between 2000 and 5000, depending on conditions
- M_m = depreciation of engine and accessories in flying hours and varies usually from 1000 to 2000
- n = total number of bare airplanes in the supply depot
- n_1 = total number of bare airplanes in regular operation on the air line
- n' = total number of engine groups in the supply park
- n'_1 = total number of engine groups in regular operation on the line
- A_p = mean time of depreciation, in years, of bare airplanes out of action or in reserve; usually between 5 and 10
- A_m = mean time of depreciation in years, of motor groups out of action and in reserve; usually between 5 and 10
- H_p = running time in hours of flight prior to an overhaul which puts the airplane structure out of action, and of the order of 300 to 1000
- H_m = running time in hours prior to overhaul of the engine group; usually from 200 to 600.
- x_p = purchase price, increased to take care of interest on and liquidation of capital invested in the plane and its accessories, spare parts and equipment, expressed in francs per kilogram of total weight of bare airplane. This specific price x_2 so defined is at the moment be-

tween 100 and 200, and varies according to the construction (wood, duralumin, or steel) of the plane

x_m = same as x_p, but for the engine group (i.e., power plant), expressed in francs per nominal hp. This specific price so defined is at the moment between 300 and 500 francs, according to the type of engine and quality of construction.

K_p = a fraction of the original purchase price of an airplane (minus engines) which represents total cost of periodic overhaul, including all the charges of the workshops and spare-parts depots doing the work. It varies from about 0.15 to 0.25

K_m = the same as K_p, but for the engine group, and varying from about 0.15 to 0.25

r = number of engine groups in each airplane

V = mean air speed of flight in regular service in kilometers per hour

λV = effective commercial speed (with regard to the ground and as an average of all flights) of aircraft in service; λ is of the order of 0.92 to 0.99

$\tan \varphi'$ = "fineness" of the aircraft at nominal flight speed, and equal to the aerodynamic "fineness" $\tan \varphi$ of the plane divided by the efficiency n of the propellers.

χ = relation between the nominal horsepower and the mean horsepower, $PV \tan \varphi'/270$, actually developed by the engines in service and at the normal altitude of flight

m = actual consumption of fuel and oil in kilograms per actual horsepower-hour. m is of the order of 0.23 to 0.28, and varies with the type of engines and the speed (i.e., rpm) at which they are operated in service

x_c = specific price of fuel and oil in francs per kilogram, taking into account all supplementary expenses such as distribution, storage, insurance. In France this is about 3 francs per kilogram for aviation spirit, and in the French colonies about double this figure. Conventionally the price should be increased by a small amount added to the actual cost to take care of additional casual consumption, such as during test flights

p = cost of line service and running maintenance of the airplane expressed in francs per hour of flight and per kilogram of all-up weight P. p ranges in value from about 0.07 to 0.12. Actually the expense of current maintenance is clearly proportional to the duration of flight and to the size of the airplane. The costs of line service for the planes (cleaning, inspection, etc.) and of their storage for the night are proportional only to the tonnage of the aircraft and to the number of flights, and not to the length of flights, so that one could break down the term p into two parts, one constant and the other varying inversely with the length of the stage. In practice, and to avoid unduly complicating these calculations, one can take a global figure p, fixed for a definite operation

g, g' = coefficients of the annual general airport expenses (other than expenses otherwise specifically enumerated). For air lines of the type in general operation today (i.e., with more than 10,000 flying hours a year and 2 to 6 airports), one can take $g = 8$ to 10 and $g' = 0.3$ to 0.5

E = length (taken as uniform) of the stages expressed in kilometers and assumed to be between 400 and 1500 km

N = total number of trips run per year on a scheduled service

F = pay-load ratio; FP = actual pay load

G, G' = coefficients for the total annual costs of the administrative and traffic departments. For the average air line, with more than 10,000 flying hours a year, one can take G as 1,000,000 to 3,000,000 and G' as 0.8 to 1.8

a = fraction of total disposable load devoted to the necessary fittings, etc., for the transportation of passengers. This fraction varies between 0.2 and 0.3 for passenger-carrying planes according to the comfort, etc., provided

α = relation of the weight of the bare airplane (without engine groups, tanks, fittings for the cabin, or special equipment for pilotage or navigation) to the total weight P. α varies from 0.25 to 0.35 according to the type, size, and construction of the plane

q = specific weight of the engine groups expressed in kilograms per nominal hp. q varies from 1.1 to 1.4 or from 1.0 to 1.3 for engine groups, water- or air-cooled, not supercharged, and of a capacity greater than 250–300 hp. This specific weight increases slightly as the hp of the engine decreases, and on the other hand decreases when the normal hp is increased by supercharging

v = speed (in kilometers per hour) of the contrary wind agreed on in order to determine the weight of fuel and oil to be carried.

In addition, the number of airplanes and engines necessary for a given service can be determined by formulas as follows:

The total number of airplanes

$$n = n_1 + n_2 + n_3$$

where n_1 = airplanes in regular service
n_2 = airplanes out of action for repair or overhaul
n_3 = airplanes in reserve to meet the emergencies of the service.

In the same way the total number of engines n' includes n_1', engines in regular service, n_2' engines under overhaul, and n_3' engines in reserve. The number n_1 of airplanes in regular service is equal to N/N', where N is the total number of trips budgeted for per year on scheduled service and N' is the number of trips which each airplane should fly per year in regular operation.

The number n_1' of engines in regular service is equal to the product $n_1 r$, i.e., the number of planes n_1 in regular service multiplied by the number r of engines in each plane.

The numbers n_2 and n_2' of airplanes and motors out of action for repair and overhaul are given by the formulas

$$n_2 = \frac{J_p N \lambda V}{365\,EH_p}; \qquad n_2' = \frac{J_m N \lambda V}{365\,EH_m}$$

where J_p and J_m represent the average time in days per year during which a plane or engine is out of commission for repair or overhaul.

As for the numbers n_3 and n_3' of airplanes and engines held in reserve, they depend essentially on the disposition of the stops and the nature of the traffic, as well as on the actual variables more or less unforeseeable. As a general rule, at least one complete airplane and one engine group ought to be held in reserve at each regular stopping point.

II—EXAMPLES OF APPLICATION—DISCUSSION OF CERTAIN PROBLEMS

1 Application to a Simple Case

Let us take an air line with two 400-km stages, each of them traversed three times a day in each direction; i.e.,

$E = 400$ km
$N = 12 \times 365 = 4380$ trips a year.

Assume that there are 6 single-engined planes ($r = 1$) running regularly on this line; i.e., $n_1 = 6$; $n_1' = 6$, that the air speed is 200 km per hr ($V = 200$), and that the commercial effective speed is reduced 5 per cent ($\lambda = 0.95$, or $\lambda V = 190$).

The "flying life" in service is taken as 3000 hr for each airplane, ($M_p = 3000$), obsolescence is reckoned at 5 yr ($A_p = 5$), the interval between overhauls is taken at 500 hr ($H_p = 500$), and finally the elapsed time between overhauls is 120 days ($Jp = 120$). Under these conditions the number n_2 of airplanes regularly out of action for overhaul is

$$n_2 = \frac{120 \times 12 \times 365 \times 190}{365 \times 400 \times 500} = 1.37$$

or, roughly, $n_2 = 2.0$.

One plane is held in reserve at each terminal, and one at the intermediate stop, i.e., $n_3 = 4$.

In the same way for the engines, $n_2' = 2$, and $n_3' = 4$.

The supply park then has:

$n = n_1 + n_2 + n_3 = 12$ airplanes, of which $n_1 = 6$, are in regular use

$n' = n_1' + n_2' + n_3' = 12$ engines, of which $n_1' = 6$, are in regular use.

Let us now calculate the pay-load ratio F, taking the relative bare airplane weight as 32 per cent ($\alpha = 0.32$), the specific weight of motors installed as 1.4 kg per hp ($q = 1.4$), a coefficient of excess power of 2/3 or ($\chi = 1.666$), a specific consumption of 0.26 kg per hp-hr ($m = 0.26$), a contrary wind of 40 km per hr ($v = 40$), a relative commercial fineness of 13 per cent ($\tan \varphi' = 0.13$), the relative weight of fittings in commercial flight as 20 per cent of the whole disposable load ($a = 0.2$), and finally a total weight of 4000 kg ($P = 4000$).

With these conditions, we calculate the nominal (rated) capacity of the engine at 640 hp, and the weight distribution of the plane is as follows:

		Weight, kg		Relative weight
Airplane		1280		0.32
Engine group		890		0.223
Crew and equipment		249		0.0625
Fuel, oil, and tanks		251		0.063
Total disposable load		1330		0.3315
being	cabin fittings 270		0.0665	
	pay load 1060		0.265 = F	
		4000		1.0

For the engine groups let us allow a service life of 1800 hr ($M_m = 1800$), an obsolescence life of 5 yr ($A_m = 5$), an interval between overhaul of 400 hr ($H_m = 400$).

For the cost of material let us allow a specific price of 150 francs per kg for the airplane ($x_p = 150$), of 300 francs per hp for the engine ($x_m = 300$), of 3.20 francs per kg for fuel and oil ($x_c = 3.2$), and finally a depreciation figure of 1/5 of the cost price for the airplane ($K_p = 0.2$) and 1/4 for the engine ($K_m = 0.25$).

Let us further decide on a coefficient for line service and running maintenance for the equipment of 0.075 franc per hour of flight and per kilogram of airplane weight ($p = 0.07$) and on coefficients $g = 8$ and $g' = 0.35$ for the general service expenses of the airports, and finally on $G = 1{,}000{,}000$ and $G' = 0.08$ for the costs of the administrative and traffic departments.

We have thus defined all the elements for the calculation of the net cost for an ordinary service on a short and easy line with very regular operation, using airplanes and engines corresponding to the present-day normal construction of commercial airplanes of moderate size and good quality. Speaking generally, all the conditions assumed are, although realizable, rather optimistic at the moment.

We can now calculate the net cost X per ton-kilometer *available*, taking into account incidental expenses of operation by giving the coefficient i a value corresponding to anticipated irregularities and the incidents and accidents common to average airplane operation. Including insurance costs and other incidental expenses for a sanely planned line, we may take the value of i as being about 1.06.

We then obtain (Table 1) the elements of actual expenses referred to either the ton-kilometer available or to the kilometers flown (with an available load of 1060 kg).

TABLE 1

			Total Expenses: Francs per ton-km	Francs per km	Per cent	
Equipment						
Airplanes	Depreciation	In use	0.990	1.05	8.60	22.22
		In reserve	0.387	0.41	3.34	
	Maintenance and overhaul		1.191	1.26	10.28	
Engines and propellers	Depreciation	In use	0.530	0.56	4.56	10.75
		In reserve	0.123	0.13	1.06	
	Maintenance and overhaul		0.595	0.63	5.13	
Operations						
Flying personnel			1.191	1.26	10.28	45.17
Fuel and oil			1.582	1.68	13.68	
Line service and running maintenance			1.385	1.47	11.97	
General airport and other expenses			1.065	1.13	9.24	
General administrative and traffic expenses			2.531	2.68	21.86	21.86
Total			11.57	12.26		100
Allowance for contingencies such as extraordinary repairs and insurance costs			0.69	0.72		6
Grand Total			12.26	12.98		106

2 Importance of General Expenses

In the foregoing example the author has assumed that the hypothetical service is operated economically, particularly with regard to administrative, traffic, and airport costs. In other words, he has assumed that this 800-km line laid out in 2 stages of 400 km each (i.e., 2 terminal airports and 1 intermediate landing ground) was established under very favorable operating and traffic conditions.

Such is not the case for the majority of the services being run today. Very often air lines operate between several different countries, and this fact involves complications in the running and additions to the general expenses which are easy to understand. On the other hand, air-transport companies, in spite of the development of air lines, have relatively small fleets and little traffic as compared with the great railway or shipping companies. The fact is that commercial aviation is still only in its early stages, nevertheless the organization of each company must include a considerable number of departments to study the numerous problems, technical, economic, legal, and otherwise, which are encountered in operating an air line. These problems can only be solved by incessant experiment, research, and practical operation. The expense of maintaining these departments is generally found to be heavier than has been assumed in the preceding example.

In order to keep in our minds the importance of these general charges in the actual state of commercial aviation, let us modify the assumptions made in the example so that these expenses will conform to all intents and purposes to the figures realized in 1930 on a French air line.

For the general expenses of the airports we can then substitute for the values of g and g' (viz., 8 and 0.35) the values 9 and 0.40; as regards the general expense of the administrative and commercial services, we should double the figures for G and G', that is, instead of 1,000,000 and 0.8, call them 2,000,000 and 1.6.

In these circumstances, if all the other factors were unchanged, Table 1 of the net cost of transportation would appear in its revised form as Table 2.

TABLE 2

			Total Expenses: Francs per ton-km	Total Expenses: Francs per km	Per cent	
Equipment						
Airplanes	Depreciation	In use	0.990	1.05	6.71	17.42
		In reserve	0.387	0.41	2.62	
	Maintenance and overhaul		1.191	1.26	8.09	
Engines and propellers	Depreciation	In use	0.530	0.56	3.57	8.43
		In reserve	0.123	0.13	0.84	
	Maintenance and overhaul		0.595	0.63	4.02	
Operations						
Flying personnel			1.191	1.26	8.09	37.67
Fuel and oil			1.582	1.68	10.73	
Line service and running maintenance			1.385	1.47	9.38	
General airport and other expenses			1.396	1.48	9.47	
General administrative and traffic expenses			5.385	5.71	36.48	36.48
Total			14.755	15.64		100
Allowance for contingencies such as extraordinary repairs and insurance costs			0.878	0.94		6
Grand Total			15.633	16.58		106

We can conclude that the increased expense estimated for the general charges translates itself into an increase of 28 per cent on the net cost. Such an increase is altogether in accordance with the actual influence which peculiarities of the organization and running of a line can exert on the net cost of operation independently of the equipment used. It is clear that an increase in these general charges of every kind has the effect of reducing the relative importance of the purchase price, and the maintenance and operating costs of the flying equipment. It follows that the importance of technical progress will be felt the more as the general costs of air transport can be reduced with the development of more profitable air lines.

3 Influence of Speed on the Net Cost

Let us go back to the formula which is the basis of our calculation and look into the influence of V on the different figures.

Consider the numerator only, of which the figures represent the costs in relation to each hour of flight and to each kilogram of the gross weight of the airplane (or to this weight in tons when one considers the coefficient 1000 used as a common factor of the numerator).

These figures fall into two categories; those in the first class are independent of V because they represent hourly expenses (such as obsolescence and maintenance of the airplanes, crew expenses, ground expenses, and maintenance) which are independent of the distance covered by the airplane in any given time of flight.

Those in the other class are proportional to V because they represent hourly expenses which are either exactly proportional to the power of the engines (partial obsolescence, overhaul, and fuel and oil consumption of the engine group) or inversely proportional (obsolescence of material in reserve, other expenses of airports, expenses of administrative and traffic departments) to the length of the stages because the corresponding annual expenses are either fixed by or proportional to *the number of trips flown* and do not essentially depend on the duration of those trips.

Divided by the factor λV which appears at the left of the denominator, the figures of the numerator then represent the costs in relation to the kilometers flown and to the gross weight of the airplane in kilograms (or tons if we multiply the figures of the numerator by the common factor 1000).

In accordance with the foregoing, we see that the figures in the first class decrease as V grows larger, while those of the second class become independent of V, that is to say, they are constant, always excepting those which depend on the rated or effective horsepower of the engine. For these latter χ, $\tan \varphi'$, and even m can vary more or less as we increase V. However, this last influence is inconsiderable, and if we neglect it we can make the statement that in general the total of the expenses per kilometer decreases with increasing speed, at least up to a certain limit.

As for the denominator (the factor λV having been dealt with), it represents the pay-load ratio F.

In this, the increase of speed of flight entails an increase in the rated and used horsepower of the engine and can bring about a modification of the manner of flight. It follows that the relative weight of the engine group varies not only because it is directly proportional to V but also by reason of the variation which may have to be made to χ, $\tan \varphi'$, and q.

As for the relative weight of fuel, oil, and tanks, it is affected only by the influence of the possible variation of $\tan \varphi'$ and by the influence (favorable in sign) of the lessening of the quantity $V(V - v)$ which takes care of the increased factor necessarily affected by the importance of contrary winds.

Therefore, in general, the pay-load ratio diminishes as V increases, and this effect is opposed to that which we have seen above for the expenses per kilometer.

The result is, therefore, that the net cost affected by these two opposing influences can vary but little with the speed, and can exceed a very small minimum only when the speed is very high. This particular property of the airplane, *eminently characteristic of its fitness for economic transportation at great or extreme speed*, can be specially developed if we increase the speed of flight by adding as little as possible to the weight of the engine while increasing as much as possible the effective power which it can give.

In order to study this problem we can consider two kinds of flight, namely, at low and at high altitudes; we can consider flight at 1000–2000 meters, which is normal in commercial aviation today, or flight at an altitude much higher, e.g., above 4000 meters.

In the latter case, remaining at such a height for a long flight involves the provision of special arrangements for the crew and passengers, and in consequence we can equally well consider regular flight at heights already reachable by commercial airplanes (4000 to 6000 meters) or at much greater heights (6000 to 12,000 meters, for example). The influence of high-altitude flight on the net cost will be considered later.

For flight at ordinary heights an increase of V involves, for any given airplane, an increase of $\tan \varphi'$, for the speed actually employed by commercial airplanes is already somewhat higher than the so-called "economic" speed which corresponds to the minimum of the apparent "fineness," $\tan \varphi'$.

One can nevertheless see the possibility of an appreciable increase of V without alteration of φ', either by increasing the weight per square foot (progress in this direction is soon stopped by the increase in landing speed and the risks such increase brings) or by "cleaning up" the airplane, particularly by reducing all forms of parasite drag.

To show the influence which the increase of V may have on the net cost, let us take the first example given above and vary only the factor V, giving successive different values to $\tan \varphi'$. We can then plot a suitable graph from which to ascertain the influence of the variation of V, both when supposing V constant and when an arbitrary law of variation is given to $\tan \varphi'$ as a function of V.

We then obtain, with all the assumed figures of the previous example (see Art. 1 of this section), for the pay-load ratio F the values given in Table 3, and for the net cost X per ton-kilometer available those in Table 4. These results are shown by the curves for $\tan \varphi'$ in Figs. 1 and 2.

The graph of Fig. 1 shows very clearly that the variation in net cost X (as a function of speed), with $\tan \varphi'$ constant, decreases with lower values of $\tan \varphi'$. In the example in question, when $\tan \varphi'$ has a value of 0.13, X only increases about 22 per cent when the speed increases from 160 to 320 km per hr. If $\tan \varphi'$ is 0.11, the extreme spread in the net cost falls to 5.5 per cent, which is practically insignificant.

TABLE 3

V, km per hr	tan φ' = 0.11	0.13	0.15	0.17
	Values of F			
160	0.329	0.297	0.267	0.237
200	0.300	0.265	0.230	0.195
240	0.271	0.231	0.191	0.150
280	0.242	0.197	0.152	0.106
320	0.214	0.162	0.112	0.062

TABLE 4

V, km per hr	tan φ' = 0.11	0.13	0.15	0.17
	Values of X			
160	11.08	12.37	13.86	15.75
200	10.77	12.26	14.17	16.79
240	10.79	12.67	15.34	19.50
280	11.13	13.57	17.71	24.97
320	11.70	15.18	21.69	39.00

The decrease in the net cost X obtained, for any given speed V, by reducing the value of tan φ', either through "cleaning up" the airplane or improving the power plant, is also clearly visible on the diagram. In order that it may be better appreciated, the author has added a curve for the ideal and unrealizable case where tan φ' would be nil. This curve also shows the absolute low limit possible for X. It is seen that if tan φ' is taken as low as 0.10 to 0.12 (which can be expected from really good modern aircraft, at least at normal speed of the order of 200 km per hr), we already approach closely to the theoretical limit of X.

Moreover, we must not look alone to technical progress in the design of flying equipment for a reduction of the net cost. Such reduction will in fact depend above all on the improvement in the

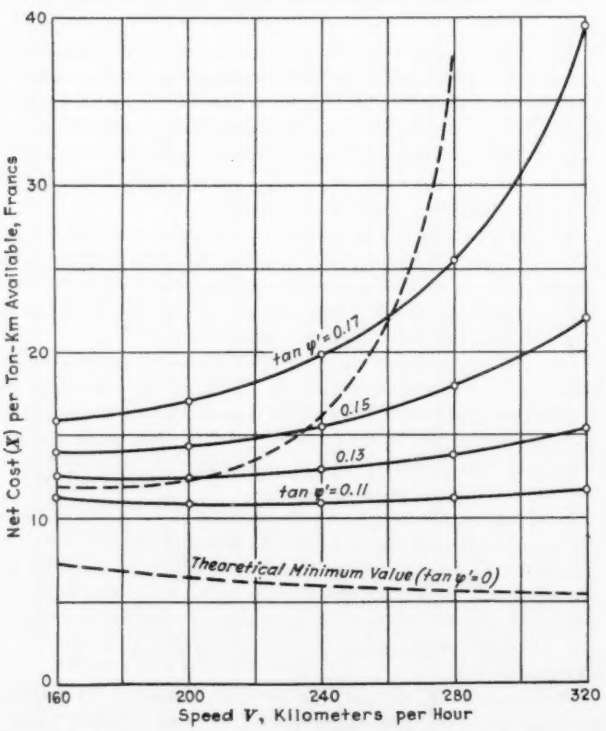


FIG. 1 NET COST PER TON-KILOMETER AVAILABLE (IN FRANCS)

life of the equipment, on the reduction of wear in service, on a decrease in initial and maintenance costs (all qualities more or less compatible), and finally on the reduction of general charges of all kinds, including organization and traffic costs.

As outlined above, for any given airplane, tan φ' increases with V when the economic speed is exceeded, which has already been done in the commercial speeds in common use today. Returning to the preceding example (tan $\varphi' = 0.13$; $V = 200$) we may admit that the variation of tan φ' with V for the airplane under consideration will be as follows:

V, km per hr	200	240	280	320
tan φ'	0.13	0.153	0.187	0.240

In accordance with these values of tan φ' we can trace by extrapolation a curve showing the variation of X as a function of V (upper broken line in Fig. 1). We get the following values of X:

V, km per hr	160	180	200	220	240	260	280
X, francs	11.7	11.7	12.26	13.4	15.9	21.3	36.7

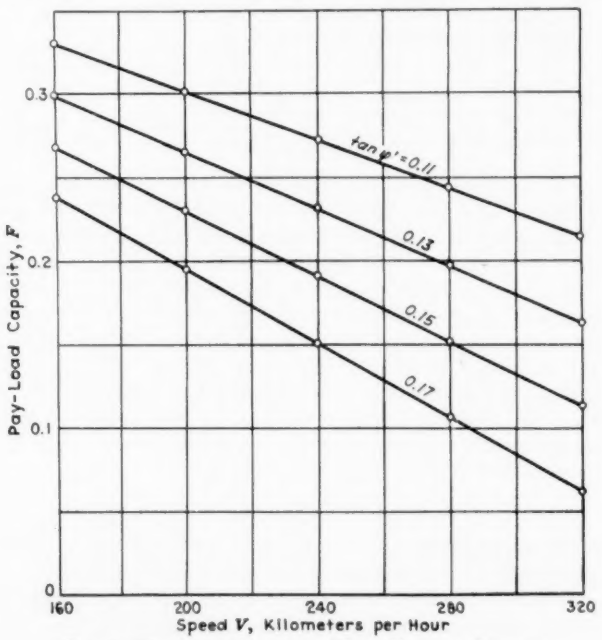


FIG. 2 PAY LOAD CAPACITY (F) AT VARIOUS SPEEDS

which show the very rapid growth of the net cost when V exceeds, under the assumed conditions, 220–240 km per hr; this results from the rapid growth of the values of the fineness tan φ'.

We see clearly then from this example that *the problem of economic flight at very high speeds is absolutely bound up with the realization of very small values for tan φ'*. These small values must be at such speeds (outside of the cleaning up of the airplane and the improvement of the power plant) as will allow us to lower tan φ' at a constant load or speed, or to reduce the load and increase the speed at a constant tan φ'. An increase in the altitude of flight will permit increasing the speed at a given load, and this without impairing tan φ', if the propeller efficiency is maintained.

We shall now examine the question of flight at high altitude.

4 FLIGHT AT HIGH ALTITUDE

Let Z be the normal height in commercial flying. Let us allow that, thanks to supercharging, the engine can maintain a constant thrust up to the altitude Z. In principle, the nominal hp of an engine of this kind is the power which an ordinary engine would have developed at ground level, where thrust at altitude Z would have been reduced by the natural falling off with height to the value estimated above.

Instead of the nominal hp, which here becomes altogether imaginary, let us consider the basic hp or the maximum power actually obtainable at all altitudes between zero and Z.

The effective power necessary at altitude for commercial flight at a speed V is always PV tan $\varphi'/270$ (hp) and the basic hp is equal to this multiplied by a certain factor of increase which we shall call χ_z.

This factor is not of the same order as the coefficient χ used earlier for a normal engine and for flights at low altitude.

In the first case, the value given to χ should be sensibly larger than unity—for example, of the order of 1.5 to 2, so that the excess of power over that required for commercial cruising will give the airplane a quick take-off, rapid climb, and sufficient ceiling to insure safety in flight. This would allow us, while cruising, to reduce the strain on the engine, which works considerably below full power, and from this again we get more freedom from engine trouble and a decrease in corresponding costs (the engine life M_m and the interval H_m between overhauls are increased).

This reduction in costs again tends to compensate, and sometimes more than compensates, for the loss of pay load which results from an increase of the engine power.

On the contrary, in the present case, even if we give to χ_s the lowest value ($\chi_s = 1$), the airplane must necessarily have a ceiling at least equal to Z, and in consequence, if Z is large enough, sufficient climbing speed. But in that case an engine would be working at Z at full throttle and thus be liable to excessive trouble. To avoid these conditions it is necessary to give to χ_s a value larger than unity, between 1.2 and 1.4 for example.

On the other hand if the basic hp, which is defined below, is taken as a term of reference, the specific weight q and specific price x_m of the engines have no longer the same values as for a normal engine for which one takes them in reference to nominal hp.

Let q_s and $(x_m)_s$ be the coefficients to employ in the case of a supercharged engine. Let us compare them with the coefficients q and x_m of a similar engine not supercharged and of a nominal power equal to the basic power of the first one.

Obviously supercharging adds to the weight of an engine and increases its price, but to a moderate degree. For the purposes of approximation we can consider that we have for different values of Z the approximate value shown below for the relations q_s/q and $(x_m)Z/x_m$:

Z, meters..	0	5,000	10,000	15,000
q_s/q.....	1	1.2	1.4	1.6
$(x_m)_s/x_m$...	1	1.3	1.6	1.9

Let us take these figures and review the assumptions of the first example studied above on the supposition that the cruising speed is made at the same efficiency (tan $\varphi' = 0.13$), which again obviously presupposes the employment of a variable-pitch air-screw, the extra weight of which is included in the increase of the specific weight just fixed for the engine groups.

Let us assume, on the other hand, that the increase in the basic power be taken at the low value $\chi_s = 1.2$ for each altitude Z above 5000 m.

Finally, the special equipment needed to make an airplane habitable during flight in rarefied air increases the weight of the cabin fittings. Instead of expressing this increase in cabin weight as a constant fraction $a = 0.2$ of the total disposable load, it will be assumed for clarity that, in a passenger plane, the total disposable load should be reduced by a constant amount increasing linearly with the height. This amount is 200 kg when $Z = 5000$ m, and 300 kg when $Z = 10{,}000$ m. We should also increase the value of the coefficient a to 0.3.

The author will consider separately the case of a mail plane, allowing the same fixed increment for the weight of fittings, but reducing the value of the coefficient a to zero. In these circumstances, we can calculate that there can be no pay load when $Z = 12{,}000$ m, which is the altitude at which the speed V reaches 395 km per hr.

The speed, the pay load, and the cost X for high-altitude flying between 5,000 and 12,000 m is as given in Table 5.

The cost thus calculated is represented as a function of V by the two curves in Fig. 3 representing high-altitude flying by passenger and mail planes, respectively.

TABLE 5

Altitude (Z) in meters..	5,000	7,000	9,000	10,000	11,000	12,000
Speed (V) in km per hr..	258	288	324	345	369	395
Pay-load ratio (f) of airplane:						
With passengers.....	0.179	0.142	0.095	0.0668	0.033	0
With mail.....	0.257	0.206	0.138	0.0958	0.05	0
Payload (kg) of airplane:						
With passengers.....	716	569	380	266	132	0
With mail.....	1,025	823	552	382	200	0
Cost (X) in francs per ton-km:						
With passengers.....	13.0	16.4	22.25	29.05	....	....
With mail.....	10.8	12.75	17.30	21.50	32.75	....

A third curve has been traced on Fig. 3 by the author which corresponds to flight at low altitude with the same value of tan φ' (0.13), which is supposed to be constant. He has also traced on the same figure a curve corresponding to an airplane with a constant speed having a "fineness" factor tan φ' equal to 0.13 at a

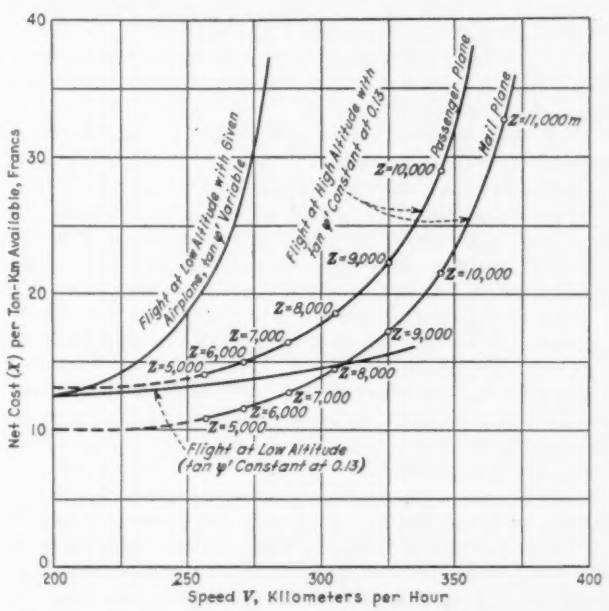


FIG. 3 NET COST PER TON-KILOMETER AVAILABLE (IN FRANCS)

speed V of 200 km per hr. It can be seen that tan φ' varies with the speed.

For example, if we take a passenger airplane for which $X = 12.26$ francs per ton-km, when $V = 200$ km per hr, and increase the speed to 260 km per hr, we obtain the following results:

(*a*) X becomes 21.4 francs per ton-km (an increase of 74 per cent) if we increase the horsepower in order to attain 260 km per hr, while still flying at low altitudes.

(*b*) X would not become greater than 13.2 francs per ton-km (an increase of only 8 per cent) *if* we could *at low altitude* increase the speed from 200 to 260 km per hr without changing tan φ'.

This would necessitate increasing the landing speed by 30 per cent (by reducing the lifting surfaces 41 per cent) or else decreasing the parasite resistance and profile drag by 36.5 per cent without changing the wing load or aspect ratio. The third alternative would be a compromise between the two methods, but all these suggestions are impossible in practice.

(*c*) X would reach 14.1 francs per ton-km (an increase of 15 per cent) if without changing the airplane or the rated sea-level speed, we could attain a speed of 260 km per hr by flying at 5100 m with a supercharged engine.

(*d*) X would *decrease* to 10.9 francs per ton-km (a *decrease* of 11 per cent) if we adopted solution (*c*) but carried only mail or freight and no passengers.

It is evident that it is advantageous and relatively easy to in-

crease the speed of flight from 200 to 260 km per hr by flying at 5100 m with a supercharged engine.

Let us go further and consider a speed of 320 km per hr; the curves of Fig. 3 show that solution (*a*) become absolutely prohibitive, X reaching an enormous value. On the other hand:

(1) X would not exceed 15.2 francs per ton-km (an increase of 24 per cent) if we could at low altitude increase the speed from 200 to 320 km per hr without changing tan φ'. This would mean either increasing the landing speed 60 per cent (by reducing the lifting surfaces 61 per cent) or reducing the parasite resistance and profile drag 56.5 per cent without changing the wing loading or aspect ratio. A third alternative is a compromise between these two, but all three are again impossible in practice.

(2) X would not exceed 21.1 francs per ton-km (an increase of 72 per cent) if without changing either the airplane or the rated sea-level speed, we reached a speed of 320 km per hr by flying at 8700 m with a suitably supercharged engine.

(3) X would not exceed 16.3 francs per ton-km (an increase of 33 per cent) if we flew at 8700 m, but carried only mail or freight and no passengers.

From these observations we can conclude that as soon as we consider commercial speeds above 300 km per hr, the efficiency of air transport is dependent on high-altitude flying.

This involves the use of supercharged engines, variable-pitch propellers, and special arrangements for making the airplane habitable for the crew and the passengers in the rarefied atmosphere.

Flight at high altitude can appreciably affect the reduction coefficient φ, by which we must multiply the average air speed to obtain the ground speed. In effect, the ground speed is reduced during the climb to high altitudes, but on the other hand we must consider the increase in frequency and intensity (at least up to 11,000 m) of the so-called "regular" winds encountered at high altitudes.

It seems generally agreed that these regular winds of from 60 to 100 km per hr velocity can be expected at altitudes of from 6,000 to 10,000 m. Nevertheless, in the foregoing calculations the author has allowed for all flights a value of 0.95 for λ, which can easily be modified if regular favorable winds can be relied upon in future high-altitude flights.

Discussion

JEROME LEDERER.[3] The formula appears to be involved, but actually it reduces to the well-known fact that the cost of operating an airline equals the sum of overhead plus operating expenses. This total is then divided by the total available pay load and by the miles flown to get the cost per ton-kilometer. However, some items in the overhead will vary with mileage flown, and this formula offers the advantage over usual rule-of-thumb methods by providing for variation in overhead with miles flown. Unfortunately, several factors in the formula would have to be obtained by actual operating experience, and by the time the operator obtained this data he could find out what his costs were by simply referring to his books. However, the formula, with simplifications and assumptions, might be used to make comparisons of different types of aircraft flying over the same route.

One's first reaction to the formula is that it goes into too much detail; for example, why should the weight of cabin fittings and of tanks be separated from the weight of the airplane. However, when one considers that the same airplane may be used for relatively long or short hops and over high or low terrain, the justification for this becomes evident. Long hops, for example require greater cabin comfort and larger tanks; hence the weight of fittings and tanks is a variable factor.

The method of calculating depreciation seems ambiguous. What are the "numbers" referred to in the sentence reading, "The complete depreciation of equipment can be calculated from the numbers allotted to the different stages." If exactness is desired in figuring depreciation and obsolescence, the following reasoning is suggested. The value of an airplane will depreciate by obsolescence and/or by use. An airplane which is kept in a hangar continuously will be obsolete at the end of, let us say, 6 yr; an airplane in constant use might be said to have a life of 3000 flying hours. Briefly it is good for a certain number of hours, and/or a certain number of years. However, these factors of time and use are seldom combined in figuring depreciation.

It does not seem particularly difficult to take both factors into consideration. Assume an airplane worth \$6000 is good for 3000 flying hours and is obsolete at the end of 6 yr. Then it will depreciate at the rate of \$2 per hr. Whether it flies or not, it will depreciate \$1000 per yr, due to obsolescence. Consequently, if the airplane is used 500 hr per yr, it is being used to the full extent of its obsolescence—that is, \$1000 (\$2 × 500)—and its value would be \$5000 at the end of the first year.

If the plane were used less than 500 hr, its depreciation would still be \$1000, due to obsolescence. If it were used more than 500 hr, say 1500 hr, its depreciation due to obsolescence would be \$1000 as before, and for use would be (1500 — 500) × \$2 = \$2000, or a total of \$3000. The minimum depreciation is fixed by obsolescence; the maximum depreciation is fixed by obsolescence plus use. Expressed in general nomenclature, let D = value of plane less engine, in dollars; H = life of plane, in hours; Y = life of plane, in years; and h = number of hours it is used per year. Then D/H = rate of depreciation due to use in D/hr; and D/Y = rate of depreciation due to obsolescence in D/yr.

When $h \times D/H = D/Y$, the plane is being used to the full extent of its obsolescence, or H/Y hr per yr. If h is less than H/Y, the depreciation is D/Y dollars per yr. If h is greater than H/Y, by say A hr, the depreciation is $(D/Y + A \times D/H)$ dollars per yr.

This method automatically takes care of the depreciation of an airplane due both to use and to obsolescence, and determines the minimum number of hours a plane should be used to obtain full advantage of its growing obsolescence.

AUTHOR'S CLOSURE

The author sympathizes with the feeling of dissatisfaction which Mr. Lederer expresses—i.e., that the proposed equation contains certain conditions of which the numerical value cannot be supplied otherwise than by practical experience. But any formula for the precalculation of net cost price will be exposed to the same criticism, because the elements of the price to be calculated are naturally of an empirical kind and cannot be known with certainty before real expenses have been incurred for the case under consideration.

No great importance should be placed on this remark, however. Every human "prevision" always supposes a certain extrapolation of the required results. This is an extrapolation which can be safely made. Moreover, my equation is for the purpose of making intelligent comparison possible, and likewise it is for the study of variations of the net cost price by changing its principal factors at will. The observations presented by Mr. Lederer with regard to effects of age, use, and depreciation value of a piece of equipment are entirely correct. His remarks furnish a useful comment on a point that I have purposely omitted to discuss in order to make my explanation simpler and shorter, and in which I have made each of these elements of depreciation to bear only on the material in reserve or on that which is in regular service.

[3] Chief Engineer, Aero Insurance Underwriters, New York, N. Y. Assoc-Mem. A.S.M.E.

AER-55-17

Shaft Drives for Airship and Airplane Propellers

By J. L. GOLDTHWAITE,[1] INDIANAPOLIS, IND.

General recognition of the possibilities of gear and shaft transmissions would free the designer from the limitations imposed by the fixed-unit power plant and appreciably broaden the field of aircraft design. Against these advantages are the weight and many engineering problems. The author discusses the most important problems from the standpoint of transmission design and suggests methods of meeting them. Working stresses, torsional vibration, dynamic balancing of long shafts, flexible couplings, and splines receive attention.



A UNIT POWER PLANT consisting of a propeller closely coupled to the nose of a heavy engine fits the typical airplane of today. It seems a perfectly natural arrangement, until some one asks, "What would you think if the designer of a ship put the propeller in front to throw the water back over the hull, impeding its progress, and if he put the motors out in the water instead of inside the hull?" And then one begins to suspect that the arrangement appears natural only because the airplane has developed around that kind of a power plant.

There are several types of aircraft for which the conventional arrangement is distinctly awkward. Amphibians and other watercraft properly take geared engines with large propellers, but it is difficult to secure sufficient clearance above rough water, particularly with flying boats; and engines over the wings are objectionable because of drag and interference, not to mention structural problems. A better arrangement would be to place the engines in the pontoons or floats, with shafts running up to the propellers. One engine can drive two propellers, or two engines can be geared to one propeller. By this plan the engines are entirely enclosed, noise and exhaust are far removed from the cabin, and the struts are relieved of the weight of the engine in landing.

In large multi-engine monoplanes, the thick wings provide free and frictionless space for the engines, but it is then necessary to provide shaft extensions to the propellers. The latest Ford plane has two such wing engines; the advantages are obvious.

Several years ago our company built for the Army a four-motor gear designed to drive one very large propeller with four Liberty engines. It passed a successful test on the stand, about 80 hours at full power, but the test consumed 15 Liberty engines, which perhaps explains why the transmission was never actually used.

In all large airships before the *Akron*, engines were hung outside in power cars. Streamlining of the cars and the necessity of clutches and brakes required shaft extensions several feet long. The *Shenandoah* drives included also reduction and reverse gears.

The *Akron* introduced an entirely different and very elaborate power plant. The eight engines are mounted inside the ship proper—an arrangement made possible by helium. From each engine a shaft more than 20 ft long extends laterally to a reduction bevel-gear head, which is supported on an outrigger and arranged to swivel so that the propeller may be pointed downward; and as the engines are reversible, the propeller may be used to exert thrust up or down, forward or backward. Two flexible couplings are used to allow for weaving of the ship structure, and a clutch and brake are provided at the inboard end. The gears are spiral bevels mounted in four bearings, and adjusted for reversible operation. The tilting device is a worm and quadrant in the drive-gear head, hand operated from the engine room.

Engineering Problems

The airplane designer, as an individual, has in many cases had the vision to wish for a separation of engine and propeller, but very generally has been appalled at the engineering problems involved, such as the question of weight when a propeller shaft weighs one or two pounds per inch; the matter of those 15-lb bearings and their housings; flexible couplings are unavoidable; few engineers have had experience with propeller gears; there are serious problems of whipping shafts and torsional vibration.

General recognition of the possibilities of gear and shaft transmissions would appreciably broaden the field of aircraft design. A considerable number of successful transmission systems have been built in the past several years, but experience in this field is built up slowly and cautiously, and the early designs, to be successful, must necessarily tend to be over-conservative. The author wishes to point out the most important problems and difficulties, to suggest methods of studying and meeting them, and above all to plead for further study and research, so that the airplane designer may be freed from the limitations of the fixed-unit power plant.

Stresses

Modern alloy steels are readily heat treated to an ultimate strength in shear of 100,000 lb per sq in. Presumably the fatigue strength bears the same relation to the ultimate in reverse shear as in reverse bending, and if torsional vibration is limited to an amplitude that just doubles the mean stress at peaks (greater amplitude would cause torque reversal, which is intolerable), then a design stress of 25,000 lb against nominal full-power torque should be permissible. But experience points to about half that figure as the safe limit for crankshafts; the discrepancy, of course, lies largely in the extremely complex nature of crankshaft stresses, which makes accurate analysis impossible. A simple drive shaft is free from such uncertainties; yet we have never dared go to a stress as high as that just suggested, for there is too much at stake in each new design. Eventually some one must take the chance, with his eyes open;

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

meanwhile the conservative course is to keep below 12,000 lb and take the weight penalty. There is much need for more research work, both in the laboratory and in service, on the fatigue strength of steel in torsion.

Torsional Vibration

In a simple engine-propeller system resonance occurs ordinarily only in single-noded vibration, and is substantially independent

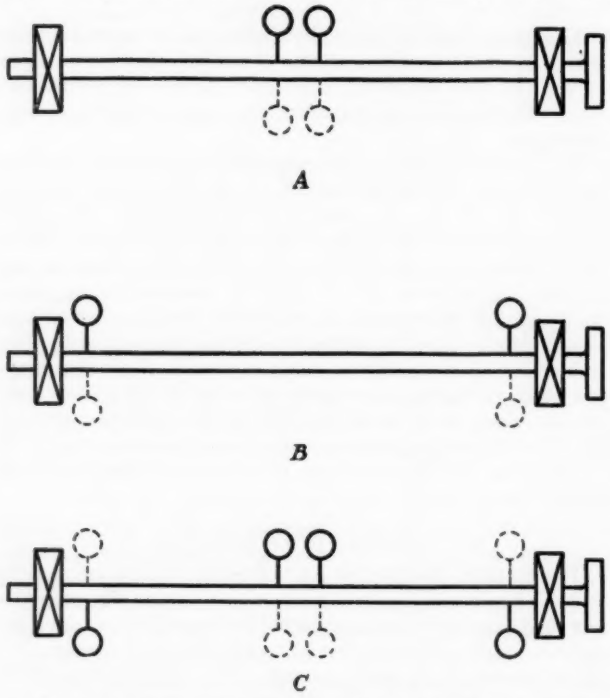


Fig. 1 Examples of Unbalance, With Corrections Shown by Dotted Lines

of the mass of the propeller. In other words, there is only one resonant frequency, and that one is inherent in the engine, although it of course may come in at several different engine speeds, excited in turn by different orders or harmonics of the torque curve. But as soon as a gear or extension shaft is interposed, the picture becomes more complex. In the first place, the natural frequency bears no relation to that of the simple power plant; an engine which is free from serious vibration throughout its operating range when direct connected to a propeller may run into dangerous periods when driving a shaft, and vice versa. Furthermore, a shaft always involves flexible couplings and sometimes gears, additional masses, so that vibration may be encountered in two or more nodes. In the *Akron* a 12-cylinder engine with at least eight measurable harmonics in its torque curve was coupled to a shaft system capable of vibrating in three or more nodes, so that there were at least 24 possible resonant speeds.

The five engines of the *Graf Zeppelin* are connected to their respective propellers through spring couplings and short extension shafts, almost exactly like those of the *Los Angeles* and other earlier ships. In tuning up the engine for the transatlantic flight, after many hours of successful operation, a slight change was made in the springs of the coupling. Shortly after starting on her trip, the *Graf* lost four engines in rapid order and was barely able to make her way back to a safe port. The apparently slight change in the coupling had brought the natural period directly into the operating range, and the crankshafts failed by fatigue.

The transmissions for the *Shenandoah* were built before the importance of torsional vibration was generally realized. They were designed for use with the 12-cylinder Liberty, and were entirely successful on the stand. But the engines finally selected for the ship were 6-cylinder motors, exceptionally rough, which set up vibration so severe and so sharply tuned that the shafts actually turned blue at the nodes. The difficulty was solved by the use of friction clutches set to slip just below the danger point.

A satisfactory analysis to predict dangerous torsional vibration is likely to be extremely difficult and involved; gears and couplings introduce considerable uncertainty at best. In many cases the best course is to build the transmission after only an approximate study and then explore its characteristics in actual running tests with the aid of a torsiograph. The characteristics of the system may be very materially altered, if necessary, by changing the diameter, i.e., the moment of inertia, of couplings, clutches, or gears; by changing the stiffness of shafts and couplings; or, as on the *Shenandoah*, by the use of a slipping clutch.

Speaking of torsional vibration, is it too much to ask some practical mathematician to develop an analysis without the use of differential equations? Put the problem of the simple pendulum before an engineer who is not familiar with the ritual of vibration, and he will regard it as a simple mass acted upon by a force. He will start it on a swing and will calculate directly how far it will go and how long it will take, and so derive its natural period and amplitude. His reasoning is straightforward and direct, and involves only purely physical concepts, unhampered by mathematical manipulations. But put the pendulum in the hands of a vibration specialist, and he catches it at some random point on its path and reasons backward to find where it must have been, how long ago, how it got there, how it got here, and what it will do next; he expresses all this in a mathematical code, eliminates the constants of integration, decodes the message, and discovers the natural frequency. The author does not question the value of the mathematical method, but the subject of vibration is so very important that its treatment should not be restricted, as it is now, to specialists. The designing engineer is likely to be a bit rusty in his code work.

Dynamic Balance of Long Shafts

A long extension shaft cannot be rigidly supported in an airplane, and if it is economically designed, it is likely to be operating above its critical whipping speed. And as the rigidity of the bearing supports is indefinite and variable, an unbalanced shaft may find resonant response, not only at the theoretical point, but at almost any speed, within itself, at one or both bearings, in a vertical, horizontal, or intermediate direction, unless it is perfectly balanced.

But balancing a long, slender shaft is a difficult task, involving factors that do not appear in the ordinary balancing problems. The usual methods and the familiar machines, based on reactions at the ends, do not apply; for the shaft can assume a different shape at each speed. The effect of flexibility is perhaps best illustrated by the cases of simple static unbalance illustrated in Fig. 1. Shaft *A* has two units of unbalance at the middle; shaft *B* has likewise two units, divided between the ends. If these shafts are rigid, as a turbine drum, or rigidly supported, as a crankshaft with intermediate bearings, or if the speed is low, so that distortion does not enter, then the end reactions of the two will be exactly the same, and a balancing machine would not distinguish between them. Shaft *C* would show perfect balance. But if the pieces are long and flexible, it is obvious that *A* and *B* would behave quite differently, and that *C* would be entirely out of balance because of distortion. For distortion balance not only must all forces be balanced, but all

internal couples as well; and from this standpoint the only possible complete corrections are those shown by the dotted lines. It will be noted that *C*, which from the ordinary standpoint is completely balanced, requires more correction than either *A* or *B*.

The longest single drive shaft in the *Akron* transmissions is some 13 ft long, 5 in. in diameter, with $^3/_{16}$-in. walls. Its critical speed in whip lies within the operating range. The first step toward balancing was to explore it inside and out with a dial indicator while turning it slowly by hand, measuring the amount and phase of eccentricity every 6 in. of length. Counterweights were calculated and applied to correct for unbalance due to eccentricity and variation in wall thickness; for this computation the shaft was considered as being made up of four short lengths, and each length was calculated for complete static and dynamic balance. These corrections left the shaft in very good approximate balance, so that the final corrections were small; but, more important, they left it substantially free from internal couples that would cause distortion at speed.

For final balance the shaft was mounted in three ball bearings on a long, level bench weighing several tons, with clamps so arranged that any bearing could be rigidly fixed or allowed to move horizontally; the mass of the bench prevented vertical movement. It was driven through a quillshaft by a variable-speed motor. Running at fairly high speed, two bearings were clamped and the vibration at the third or free bearing was reduced to a minimum by counterweights applied by systematic trial and error. This was done at each of the three bearings in turn; then starting at the first bearing, the course was repeated several times over. When finally released, the tubes were so closely balanced that no critical speed could be detected, whether running on two bearings or three, and the maximum vibration

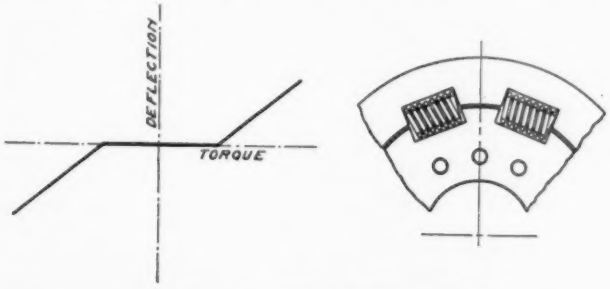


Fig. 2 Preloaded Spring Coupling

at any speed was less than 0.0015 in., or within the probable errors of the bearings and their mountings.

Flexible Couplings and Universal Joints

The influence of spring couplings on vibration has been mentioned. In many cases an elastic coupling is not required, but it is necessary to allow for angular misalignment due to weaving of the structure; probably the lightest coupling for this purpose is the internal-external gear type, sometimes known as Fast's coupling. It is sometimes objected that this type is noisy, for its flexibility is due entirely to loose fitting; but as the teeth cannot separate under load unless vibration is so severe as to cause torque reversal, the noise of the coupling might very well be regarded as a useful danger signal.

Among the elastic couplings there are several types available. In one form, Fig. 2, the drive is through compression springs preloaded in such a manner that no torsional deflection occurs until the torque reaches a certain predetermined value, the theory being that when vibration approaches a dangerous value the springs come into play and entirely change the elasticity of the system, thus destroying resonance. It can happen, however, that at certain other speeds a sudden accidental disturbance will momentarily bring the springs into play and throw the system into resonance. Once deflected, there is no appreciable change of stiffness with amplitude, so that resonant vibration once established can become dangerous. There is no friction.

The Falk or Bibby coupling, Fig. 3, depends on the deflection of flat spring elements. It has a considerable amount of free play or backlash, but its stiffness increases with increasing deflection. It may respond to a large range of frequencies, but

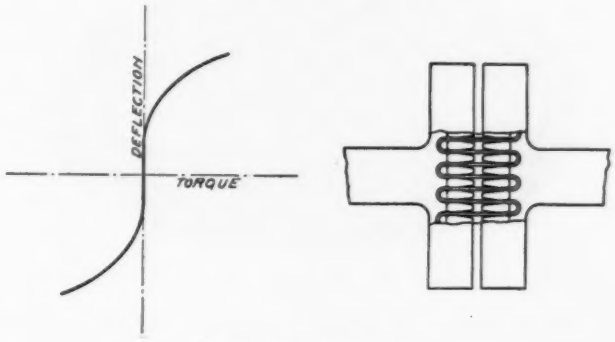


Fig. 3 Falk Coupling

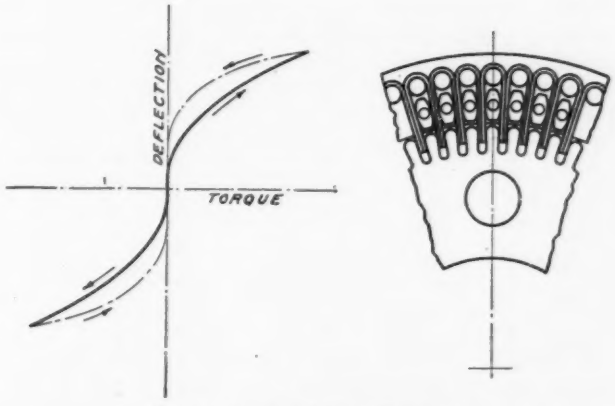


Fig. 4 Allison Spring Coupling

the amplitude at any given frequency is definitely limited. There is no friction and therefore no damping action.

The Allison spring coupling, Fig. 4, resembles the Falk in principle and in elastic characteristics, but interleaf-spring friction gives it a very considerable damping action, the hysteresis amounting to 10 to 15 per cent of each torque cycle. Consequently, although it responds to a wide frequency range, it possesses a powerful damping action. It has been used successfully in hundreds of geared transmissions and is an important element in the control of vibration in the *Akron*.

A very ingenious coupling is the Kirchbach, Fig. 5, also used on the *Akron;* it is of the familiar type that employs pins connected by links. The links are elliptical loops of flexible steel cable entirely embedded in rubber. Under tension, the loops tend to straighten out, putting the rubber in compression. Thus they combine the valuable damping feature of rubber with the strength and safety of steel.

Splines and Riveted Couplings

In view of the major importance of splines in all engine and power-plant design, it is really surprising that they have received so little attention. Except for well-placed emphasis on accuracy,

the design of splined joints has scarcely progressed beyond the rule of thumb. The author offers the following observations, although without complete proof except that the principles have been used in successful designs:

With a steady load in one direction, any reasonably well-made splined joint will distribute its load safely, because if one part is overloaded beyond the yield point, it will yield until the load is better distributed. But in aircraft power plants, with their inevitable vibration, failure by fatigue may occur at

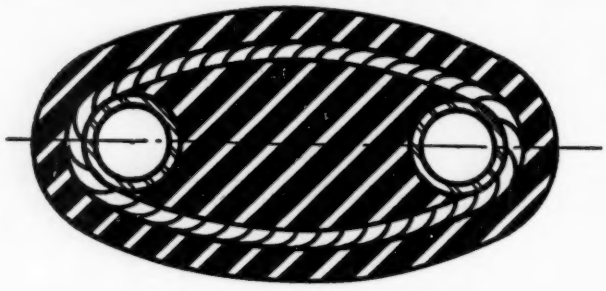


FIG. 5 KIRCHBACH COUPLING LINK

stresses far below the elastic limit. What is the fatigue strength of a perfect spline, of an ordinary good one, of a long spline, of a short one? These questions are only partially answered at present. Of two well-known geared engines, one drives its pinion successfully through a spline only $^1/_2$ diam in length; the other, with a spline length of nearly $1^1/_2$ diam, had a record of repeated spline failure; the difference, of course, was one of accuracy, affecting load distribution. It might well be pointed out that a perfect spline theoretically need be only $^1/_4$ diam long.

The importance of precision is obvious, but the influence of elasticity of the parts is not generally realized. Within the length of any joint the two members, shaft and hub, share the torque load in direct proportion to their relative stiffness. Now in the typical case of a slender shaft driving a heavy pinion or hub through a long spline, the hub is quite stiff and the shaft comparatively flexible; and it is quite obvious that the portion of the shaft within the hub cannot be carrying any load, for the shaft cannot carry torque without twisting. It follows that all the load is transferred to the hub at the point where the shaft enters it. Examination of splines in service will confirm this; they are overloaded at the point of entry, and the rest of the length does nothing.

Now suppose the hub and shaft are made to be equally stiff. Similar reasoning shows that they share the load equally. It follows that half of the load is transferred to the hub at the point where the shaft enters; throughout the length of the joint the load remains equally divided, and so there is no transfer; at the opposite end, the remaining (half) load is transferred. And so, by this simple expedient, the concentrated load on the splines is cut in half. A further step can be made by tapering the hub, whereby the load can be distributed in any manner desired; but of course such refinement is entirely out of place unless the splines are very accurately cut.

Clearly, the length of a spline as a rule has no influence on its capacity. Excessive length is a handicap; a long spline is likely to be cut with a slight twist or taper, so that it seems tight when in reality only a few points are bearing; short splines, conversely, are an aid to close inspection.

AER-55-18

Torsional Vibration of Aircraft Engine Crankshafts

By W. G. LUNDQUIST,[1] PATERSON, N. J.

Results of the study of torsional vibration of aircraft engine crankshafts are presented. Almost all engines have one or more easily detectable critical speeds for torsional crankshaft vibration. If one such critical speed should fall in the operating range of the engine, it is likely to cause trouble. Many apparently unrelated failures of engine parts can be traced to crankshaft vibration. It is therefore important that the vibration characteristics of every engine model be known so as to intelligently develop its design for maximum reliable power output.



THE subject of torsional vibration of crankshafts is certainly not new. Neither is the mathematical analysis of natural frequencies and critical speeds as much of a mystery to engineers as it was some years ago. Almost all phases of the subject have been discussed in the technical literature of recent years, and it therefore becomes difficult to present any new data which will be of interest to any one who has studied this source of trouble in internal-combustion engines. However, since aircraft engines appear to hold a popular interest and since they are by no means immune to the malady under consideration, a short presentation of some of the results of such vibration studies may be in order.

Almost all engines have one or more easily detectable critical speeds for torsional crankshaft vibration. If one such critical speed should fall in the operating range of the engine, it is very likely to cause trouble. There is a popular misapprehension to the effect that if the crankshaft of an engine does not break during operation, the engine must necessarily be free from serious crankshaft vibration. Such is not always the case. A long list of apparently unrelated engine-parts failures can be traced to crankshaft vibration, because such vibration besides overstressing the crankshaft also induces abnormal stresses in crankcases, connecting rods, valve gear, and accessory drives—particularly supercharger drives. It is therefore of great importance that the vibration characteristics of every engine model be known in order to intelligently develop its particular design for maximum reliable power output.

Present-day conventional aircraft engines can be classified roughly under two headings: first, in-line engines and, second, radial engines. Most of the engines under the first classification will be found to be liquid cooled, while the majority under the second classification are air cooled. However, as far as this study goes the method of cooling is of no great importance and will be given only little attention. The vibration characteristics of the engines under these classifications are as distinct as the classification to which they belong, and it is this point and the reason therefor that we wish to consider among other things.

Before taking up any specific phase of this subject, however, let us review briefly the underlying cause of torsional vibration in any engine in order to direct our attention intelligently and in order to give us a consistent method of attack in all cases.

Every crankshaft and connecting-rod system has at least one natural mode of vibration, the frequency of which can be determined. The calculation of natural vibration frequencies has been treated so thoroughly in technical literature of recent years that it need only be mentioned here. Any one not already familiar with the subject need only refer to Professor Timoshenko's books or to F. M. Lewis's paper of 1925, in order to get the necessary information for calculation of natural vibration frequencies. Only the single-noded mode of vibration will be considered in this discussion, since that is almost invariably the one which gives trouble. Since we have at our disposal, then, ready means for determining natural frequencies of crankshaft assemblies, we need only in addition to determine the engine speeds at which dangerous resonance may occur. Again we find ample literature to help us out, and we will only consider the main points.

As we all know, torsional crankshaft vibration is caused by torque variation. Every engine has a torque curve—sometimes unfortunately so. This torque curve is made up of the summation of the turning efforts of all cylinders combined in the proper phase relation as determined by the firing order. The fluctuations indicated by this curve may be large or small, depending upon the engine; but regardless of the size of the engine, the curve will repeat itself once every two revolutions of the crankshaft in the case of a conventional four-stroke-cycle engine. Accepting the harmonic method of analysis, we can take the turning-effort curve for each cylinder unit and resolve it into its harmonic components, the magnitude of which we are able to determine. We can then combine each successive component of each cylinder with the corresponding components of all the other cylinders of the engine in their proper phase relation and determine which harmonic of the single-cylinder torque curve will have a resultant effect in the engine. Selecting from this analysis the harmonic components which are cumulative, we can easily establish the location of our critical speeds. If we are still curious and wish to try to calculate the amplitude of vibration at a critical speed, it becomes necessary to know the magnitude of the damping forces in the engine, and in our present state of ignorance we can do little better than guess at the value of this variable. If we are going to guess at one of the stages of our calculation, we might just as well guess at the final result to start with, and thus save ourselves many hours of hard work. Furthermore, there are instruments available which will tell us without guesswork what the amplitudes during critical speeds are, and if our calculations do not agree with the instruments, we will have to believe the experimental data anyway, until we know more about

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the subject. When sufficient consistent experimental data have been obtained, it will no doubt be possible to estimate within very close limits what the damping factor of a particular engine type will be, and using this value, we will be able to predict before building any engine very closely what the vibration characteristics will be. Until such data are available, however, we will have to content ourselves with being able to predict the location of the critical speeds and will have to use our judgment, based

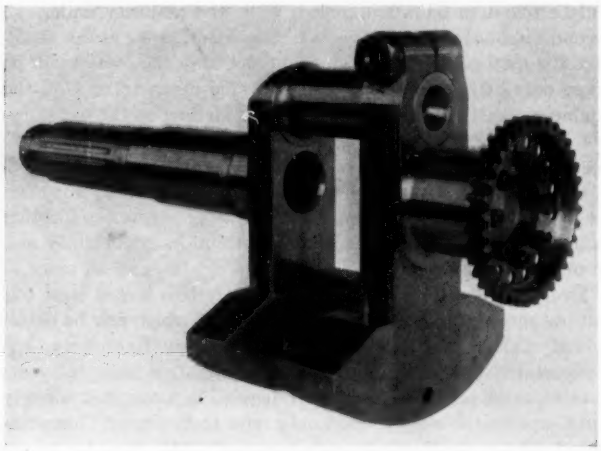


FIG. 1 SINGLE-THROW CRANKSHAFT

on experience in estimating the seriousness of any particular critical speed.

We will consider radial engines first, since they are the most orthodox in their behavior where torsional vibration is concerned. Strangely enough, radial engines show more violent vibration amplitudes during a critical speed than do line engines. Vibration torques reach almost unbelievable magnitudes in the case of some of the large engines. We will try to discover the reason for this as we proceed.

Most successful radial engines of today are single-row types having single-throw crankshafts. Following in general our proposed line of attack, we will first see how the natural frequencies work out.

Fig. 1 shows a conventional single-throw crankshaft as used in Wright engines.

The natural vibration frequencies of single-row radials with such shafts will usually be found to lie between 150 and 300 cycles per second with the small displacement engines on the high-frequency end of the list. The author has found that over the conventional range of displacements the natural frequency of similar engines varies inversely as the displacement raised to the 0.30 power quite closely, or

$$\frac{n_1}{n_2} = \left(\frac{D_2}{D_1}\right)^{0.30}$$

Accepting this formula as a rough approximation, we can make a fair guess at the range of natural frequencies which may be expected with present-day single-row radials. The main difference between engines of the same displacement will be the torsional stiffness of the crankshaft. If we assume certain extreme proportions for the dimensions of possible shafts, we can locate approximately the limiting curves as shown by Fig. 2. Curve *A–B* will pertain to engines having relatively stiff crankshafts (usually short nosed), while curve *C–D* will pertain to engines having relatively very flexible shafts (usually long-nosed). These curves are of interest since they facilitate the study of any particular family of engines—particularly so in the case where the design of a whole series of engines is contemplated. In such an instance the proportions of the basic design can be so determined as to preclude the coincidence of a critical speed with the operating range of any one of the models of the series.

Concerning two-row radials, few data are available. In general, however, two-row radials will be found to have higher natural frequencies than single-row engines of the same displacement. Two-row frequencies are indicated by the estimated curve *E–F* on Fig. 2. The author wishes to point out here that these curves are empirical and must be used only with considerable discretion.

After determining the natural-vibration frequency of the crankshaft assembly of an engine, the next step is to find what harmonics of the torque curve will be cumulative for a particular engine. In the case of radial engines, we discover that as we combine the successive harmonics of the torque curves of the cylinders in their firing order, all harmonics cancel out with the exception of those which are multiples of the number of cylinders. It is customary when making a harmonic analysis to assume that all connecting rods center on the crankpin. Such an assumption is not true, of course, when a master rod with articulated rods is used. Link-rod motion modifies the torque curve some-

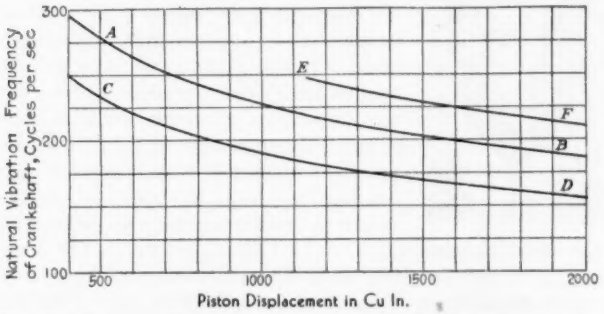


FIG. 2 NATURAL VIBRATION FREQUENCY OF CRANKSHAFTS VERSUS PISTON DISPLACEMENT, RADIAL ENGINES

what, the net result being the introduction of a second harmonic torque variation of small magnitude. This effect may be neglected when making an analysis, since the critical speed for a second harmonic is usually far beyond the normal range of the engine. From the preceding, then, we may make a tabulation of the engine-torque components which may give trouble. (See Table 1.)

TABLE 1

Engine types	Critical harmonics
5-cylinder radial, single-row	2, 5, 10, 15, 20, etc.
7-cylinder radial, single-row	2, 7, 14, 28, etc.
9-cylinder radial, single-row	2, 9, 18, 27, etc.
11-cylinder radial, single-row	2, 11, 22, 33, etc.
14-cylinder radial, two-row	2, 14, 28, etc.

Knowing which harmonics are effective, we locate the critical speeds as follows:

$$N = \frac{n(120)}{K}$$

where N = critical speed in rpm
n = natural vibration frequency of crankshaft (cycles per second)
K = number of harmonic torque component, as per Table 1, known as harmonic coefficient.

Let us assume that we have a whole series of single-row engines whose natural frequencies lie along curve *C–D*. We will further assume that the displacements of our models are as given in Table 2.

TABLE 2

	Cu in.	Rated speed (rpm)
5-cylinder model	500	2000–2400
7-cylinder model	700	2000–2400
1st 9-cylinder model	900	2000–2400
2d 9-cylinder model	1800	1800–2200

Referring now to our frequency curve and to Table 1, we can assemble the data for Table 3.

TABLE 3

Engine model	Natural frequency of crankshaft	Critical harmonics and speeds (rpm) Harmonic	Critical speed
5-cylinder	232	5	5570
		10	2320
		15	1855
7-cylinder	211	7	3600
		14	1800
1st 9-cylinder	196	9	2610
		18	1305
		9	2145
2d 9-cylinder	161	18	1072

We notice from this tabulation that the only criticals which fall in our proposed operating range are the 9th harmonic of the nine-cylinder engine and the 10th harmonic of the five-cylinder engine. Reference to a harmonic analysis will show that the magnitudes of these two harmonics are nearly the same, and both may give trouble. Torsiometer tests verify these deductions. In addition to the harmonics which fall directly in the operating range, other harmonics may also give trouble. If a large harmonic lies just below the rated operating range, it will probably give trouble at cruising speeds. Similarly, a critical speed above the rated rpm may give trouble at diving speeds in the case of pursuit ships or diving bombers.

The torsiograms of Fig. 3 were made by a Prescott torsiometer.

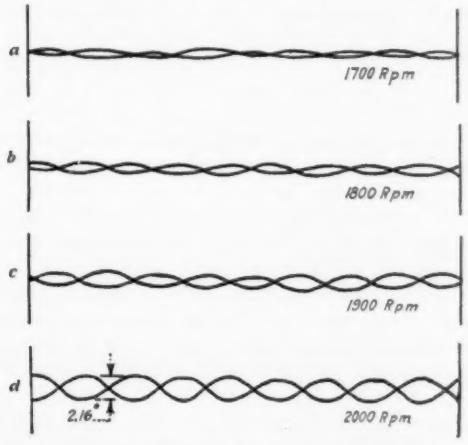


FIG. 3 TORSIOGRAMS MADE BY 9-CYLINDER SINGLE-ROW RADIAL ENGINE OF 1820 CU IN. DISPLACEMENT

(See paper, "Vibration Characteristics of Aircraft-Engine Crankshafts," by Ford L. Prescott, presented at the Fourth National Aeronautic Meeting of the A.S.M.E., paper AER-52-19.) They show the reaction of large nine-cylinder radial direct-drive engine of 1820 cu in. displacement to its critical harmonic (9th). The torque variation of this harmonic was approximately ± 9000 lb-in. The calculated critical speed for this engine was approximately 2200 rpm. The instrument was driven directly from the free end of the crankshaft. A vertical displacement of 1/8 in. represents 1 deg of torsional deflection at the crankshaft. The length of the card represents one revolution of the crankshaft. It will be noticed that two curves are recorded; these curves are two supplementary revolutions of the crankshaft, and represent a complete cycle.

From these torsiograms we can plot the curves of Fig. 4 showing the vibration torque and deflection of the crankshaft during resonant vibration. The curves represent conditions at 2000 rpm, as shown by corresponding torsiograms.

Notice from these curves the tremendous torque variation which apparently occurs during resonant vibration. Remembering that the disturbing harmonic in this case had an amplitude of 9000 lb-in., the ratio of torque variation at resonance to normal torque variation is 128000/9000, or 14.2 to 1. The damping factor (Δ) becomes 1/14.2, or 0.070.[2]

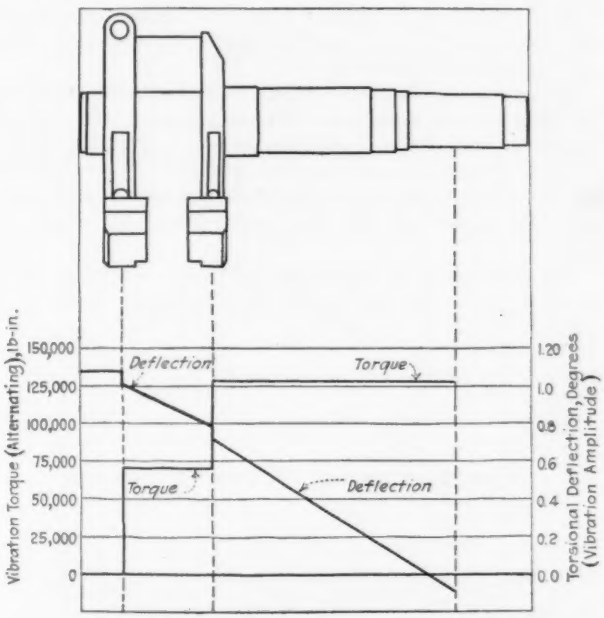


FIG. 4 TORQUE AND DEFLECTION CURVES FOR RESONANT SPEED

Major B. C. Carter states in his paper, "Dynamic Forces in Aircraft Engines," that the damping in radial engines appears to be of such an order that the torque variations in the shaft at synchronous speeds are from 5 to 10 times the normal gas-torque variations. The damping factor corresponding to this would vary between 0.1 and 0.2. It appears from the results presented herewith that the damping is less than what Major Carter suggests. The torsiogram from which our damping factor is calculated does not quite represent synchronous vibration either, being taken at a speed approximately 175 rpm below the critical speed of the engine.

Referring again to the torque curve, note the magnitude of the torque applied at the rear end of the crankpin. Mathematical analysis of the forces acting on a three-bearing crankshaft such as the one under consideration shows that approximately 38 per cent of engine torque is applied to the crankpin during normal operation. In the case of this engine the mean torque carried amounted to approximately 8300 lb-in. Due to resonant vibration we have an additional alternating torque of ± 70,000 lb-in. applied at the rear of the crankpin. The need for a good joint between the rear cheek and the crankpin is im-

[2] The resonance factor is:

$$\frac{1}{\sqrt{(1 - X^2)^2 + \Delta^2 X^2}}$$

where X is the ratio of the disturbing force to the natural frequency of the system and Δ is the damping factor. At resonance $X = 1$. The resonance factor becomes $1/\Delta$. Therefore Δ = 1 ÷ resonance factor.

mediately apparent. A well-designed clamp joint at this point will carry as much as 90,000 lb-in. torque.

It is difficult to say offhand just where the maximum actual stress occurs in a crankshaft during resonant vibration. The highest calculated stress will usually be found to occur in the section under the rear propeller cone. Oil holes and keyways, however, produce stress concentrations at other points of the shaft, and highest actual stress will usually be found in some section that is drilled or slotted. Fatigue cracks may therefore occur anywhere from the rear main journal to the propeller splines, but almost invariably do they start at some hole or sharp corner. Ample fillets, smooth holes with well-rounded edges, and careful machining operations are safety measures which must be adhered to faithfully.

We note in our fictitious analysis that the 10th harmonic of the torque curve would come into resonance in the assumed operating range of the five-cylinder engine, and we may expect trouble. This deduction is not idle speculation. Actual vibration of a five-cylinder engine crankshaft in resonance with a 10th harmonic torque variation has been recorded by torsiometer tests.

One more point should be mentioned here before we leave radial engines. We have noted that, neglecting link-rod action, the lowest harmonic of the torque curve which may cause trouble is that one whose coefficient corresponds to the number of cylinders of the engine. Furthermore, reference to a sample harmonic series will show us that the magnitude of the harmonic components decreases rapidly as the higher frequency harmonics are reached. Thus the 12th harmonic is, approximately, 25 per cent of the 6th, or again the 12th harmonic is only 45 per cent, approximately, of the 9th. If we now assume the damping factor to be constant, we know that the vibration amplitudes will be roughly proportional to the amplitude of the forcing torque. The best method of controlling vibration therefore is to use a large number of cylinders. For this reason two-row radials of 14 or more cylinders are particularly attractive. Such engines can usually be operated directly in a critical range without any observable damage.

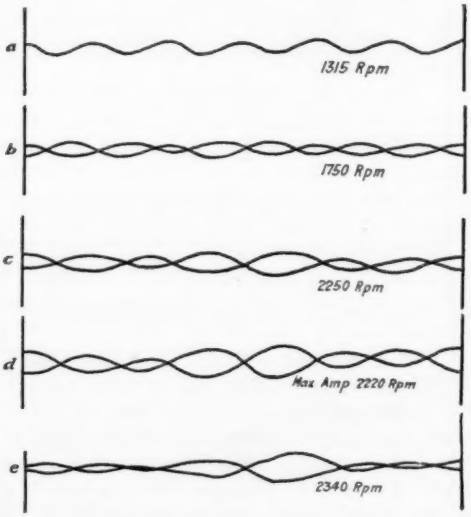


FIG. 5 TORSIOGRAMS MADE BY A 12-CYLINDER INVERTED-VEE ENGINE OF 1560 CU IN. DISPLACEMENT

The vibration characteristics of line engines are somewhat different from those of radial engines. The calculation of natural frequencies may be done in the same way as were those for single-throw shafts. It is when we attempt to determine which harmonics of the torque curve are effective that we come across some apparent discrepancies. Some actual examples will illustrate this.

The torsiogram of Fig. 5*a* is a transcript of a card made by a 12-cylinder inverted air-cooled vee engine of 1560 cu in. displacement. The card shows six cycles per revolution, or 12 in two revolutions. In other words, it represents the critical speeds for the 12th harmonic of the torque curve. If we make a harmonic analysis for such an engine as this, and combine the successive harmonics of the single-cylinder torques of the cylinders in their firing order, we discover as before that so far as resultant torque

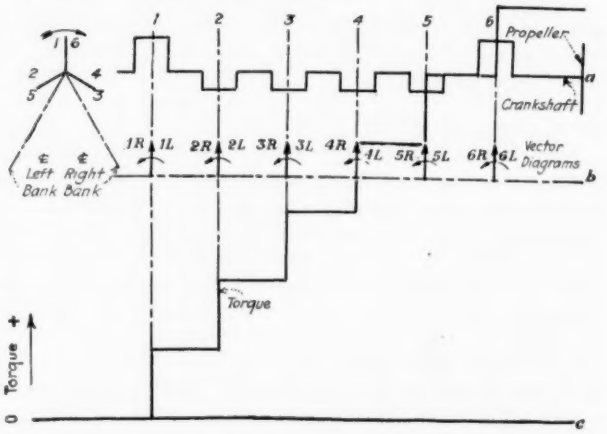


FIG. 6 SUMMATION OF 12TH HARMONIC TORQUE VARIATION OF 12-CYLINDER INVERTED-VEE ENGINE

on the whole crankshaft is concerned, all harmonics cancel out except those which are multiples of the number of cylinders. One writer has suggested that since in a line engine we always have pairs of crank throws whose torque curves are exactly 360 deg out of phase, we may combine the turning-effort curve of each pair and let our cycle of events be 360 deg of the crankshaft instead of 720 deg. Doing this automatically eliminates all half-order criticals, which is a mistake, as will be seen.

Referring to torsiograms of Fig. 5*b* and Fig. 5*c*, both made by the same engine that made Fig. 5*a*, we find that the 9th and also the 7th harmonic of the torque curve cause resonant vibration. Our previous analysis would indicate that these harmonics should cancel out. The reason for this apparent discrepancy lies in the displacement of the points of application of the successive torque components. A simple graphical representation will show this.

Fig. 6*a* shows a line diagram of a six-throw crankshaft as used in a 12-cylinder vee engine. Assume that we have an inverted engine as just described, with the banks designated as right and left when viewed from behind the engine, then cylinders 1 right and 1 left are to the rear. The conventional firing order for this engine is: 1R, 6L, 5R, 2L, 3R, 4L, 6R, 1L, 2R, 5L, 4R, 3L.

We can now draw vector diagrams showing the phase relation of the torque components of each cylinder, considering each of the harmonics of the torque curve in order. Fig. 6*b* shows such a vector diagram for the 12th harmonic (harmonic torque of No. 1 right assumed maximum). Fig. 6*c* shows the summation of applied torque plotted against the points of application along the crankshaft. Notice that the harmonic components of the cylinders add up directly from No. 1 to No. 6 throw. Figs. 7 to 17, inclusive, show similar diagrams for each of the other first eleven harmonics of the torque curve. Notice particularly that on some of these diagrams (Fig. 9 for instance) there is a resultant average torque applied to the crankshaft

between No. 1 and No. 6 throws, although the sum of the torque is zero at No. 6 throw. In a case such as this, therefore, where No. 6 throw is very near the node, we are probably safe in concluding that the crankshaft will respond to the average torque applied along its length rather than to the algebraic sum of the torque over the whole length. Our diagrams indicate that the 12th, 9th, 7th, 5th, and 3d harmonics will produce an average torque as described, while the other harmonics show none at all or at least only a very small average torque from No. 1 and No. 6 throw. Notice that the forcing torques of the 6th harmonic cancel out at every throw. This analysis agrees with the known results as indicated by tests on this type of engine. No torsiograms of a 5th or 3d harmonic have been recorded to the author's knowledge, but such vibration probably would be encountered were it possible to attain high enough speeds to come into resonance with these harmonics. By such an analysis as this we may study the effect of various crank arrangements, firing orders, and the effect of changing the angle between banks.

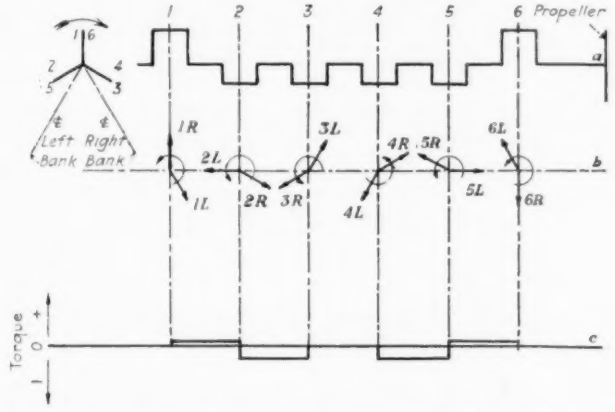


FIG. 7 SUMMATION OF 11TH HARMONIC TORQUE VARIATION OF 12-CYLINDER INVERTED-VEE ENGINE

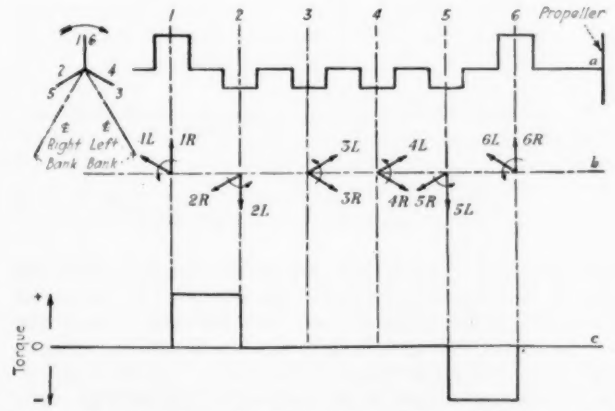


FIG. 8 SUMMATION OF 10TH HARMONIC TORQUE VARIATION OF 12-CYLINDER INVERTED-VEE ENGINE

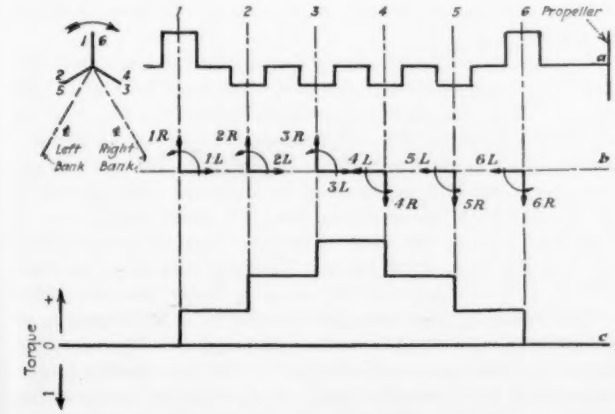


FIG. 9 SUMMATION OF 9TH HARMONIC TORQUE VARIATION OF 12-CYLINDER INVERTED-VEE ENGINE

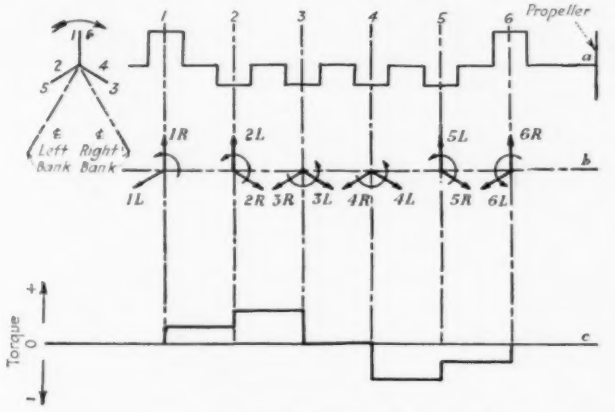


FIG. 10 SUMMATION OF 8TH HARMONIC TORQUE VARIATION OF 12-CYLINDER INVERTED-VEE ENGINE

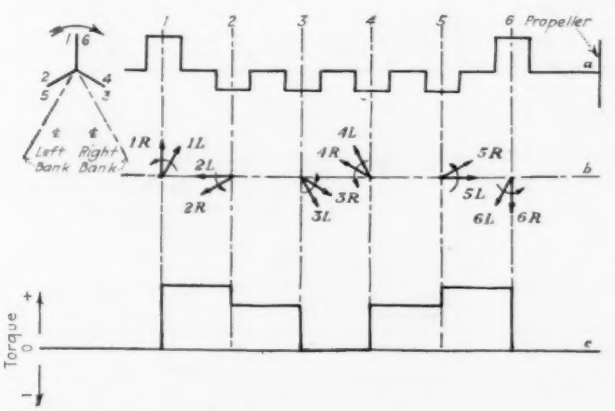


FIG. 11 SUMMATION OF 7TH HARMONIC TORQUE VARIATION OF 12-CYLINDER INVERTED-VEE ENGINE

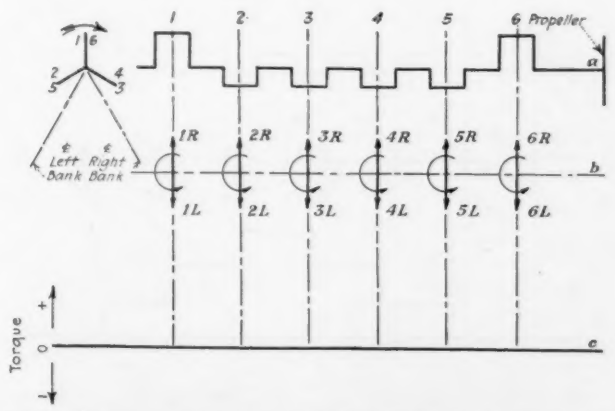


FIG. 12 SUMMATION OF 6TH HARMONIC TORQUE VARIATION OF 12-CYLINDER INVERTED-VEE ENGINE

Referring again to the torsiograms, notice that Fig. 5c and Fig. 5d show a peculiar form having large and small waves. Before we can estimate the crankshaft stresses we must determine which portion of the deflection recorded thus is to be attributed to vibration and which to normal asynchronous torque variation. Since the power impulses in a line engine are not applied to one point of the shaft but to successively different points, the free end of the shaft will not have a steady deflection due to mean torque, but will exhibit a more complicated motion.

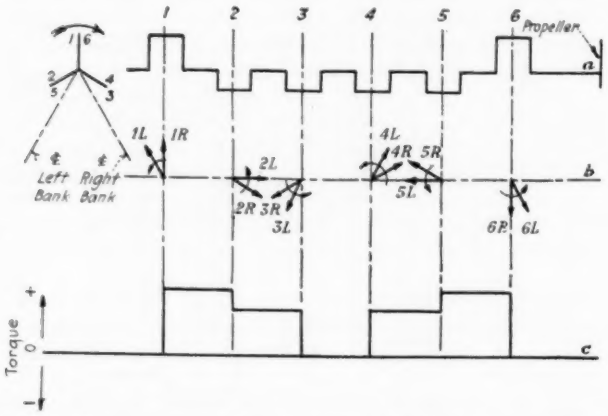


FIG. 13 SUMMATION OF 5TH HARMONIC TORQUE VARIATION OF 12-CYLINDER INVERTED-VEE ENGINE

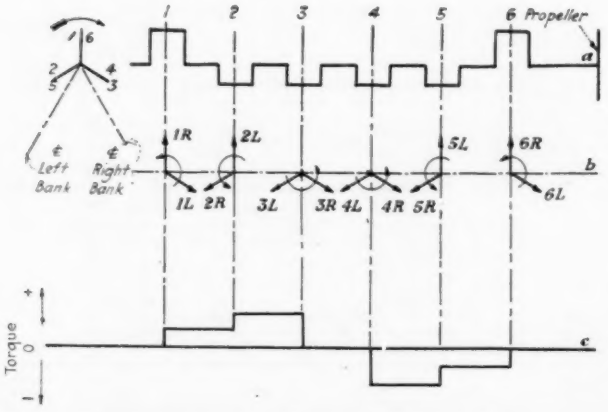


FIG. 14 SUMMATION OF 4TH HARMONIC TORQUE VARIATION OF 12-CYLINDER VEE ENGINE

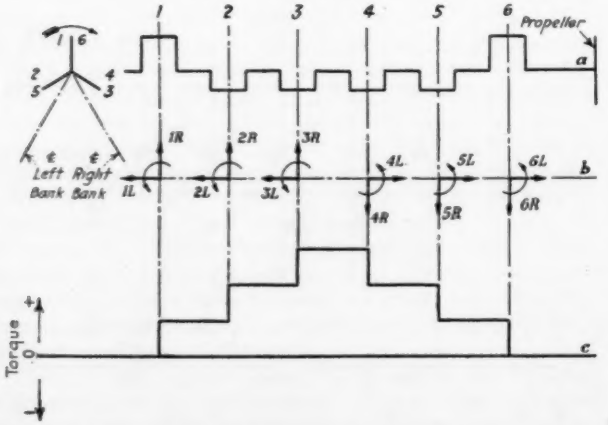


FIG. 15 SUMMATION OF 3D HARMONIC TORQUE VARIATION OF 12-CYLINDER VEE ENGINE

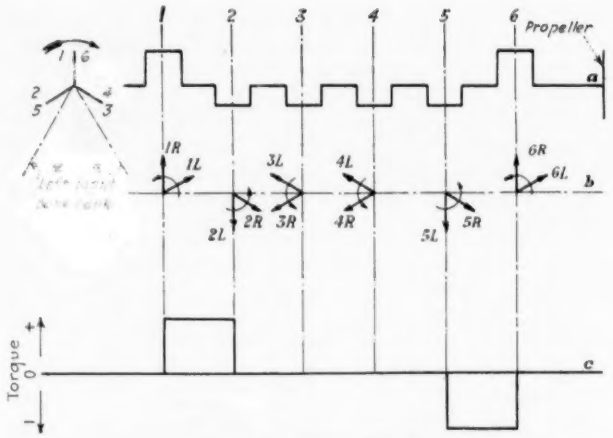


FIG. 16 SUMMATION OF 2D HARMONIC TORQUE VARIATION OF 12-CYLINDER VEE ENGINE

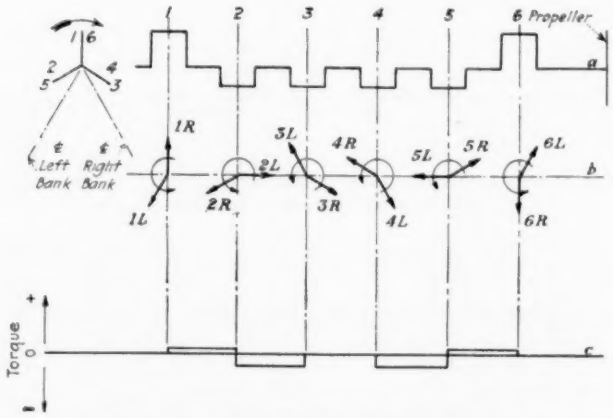


FIG. 17 SUMMATION OF 1ST HARMONIC TORQUE VARIATION OF 12-CYLINDER INVERTED-VEE ENGINE

During a critical speed there will be superimposed upon this asynchronous motion a harmonic vibration, and the torsiometer will record the resultant of these two vibrations. The easiest way to differentiate between these curves is to draw the enveloping lines of the higher frequency vibration. Fig. 18 is an example. Notice that the curve described by the envelope corresponds to the torsiogram which recorded the motion of the free end of the crankshaft at an asynchronous speed (see Fig. 18*d*).

Fig. 19 shows the elastic curve and the torque curve of a 12-cylinder engine crankshaft during resonant vibration, as shown by Fig. 5*d*.

Since we do not know exactly the resultant magnitude of the disturbing force of the 7th harmonic indicated by Fig. 5*d*, we are safer if we calculate our damping factor from the diagram of the 12th harmonic (Fig. 5*a*), because we know that all cylinder components add up directly (refer to Fig. 6*c*). The total torque variation of the 12th harmonic of the engine at 1315 rpm based on propeller-load characteristics from 600 brake horsepower at 2400 rpm was ± 620 lb-in. The total torque variation indicated by Fig. 5*a* is ± 9500 lb-in. The resonance factor becomes 9500 ÷ 620, or 15.3, and the damping factor becomes 0.065.

This would indicate that the damping in a line engine is of about the same order as that in a radial engine. The redeeming feature of a line engine in this respect is that the effective torque variation of the harmonic which usually falls in the operating range (the 7th) is small. Compare the torque summation of Fig. 6 and Fig. 11. For this reason it is not unusual to be able

to operate directly in the range of this harmonic without serious difficulty, providing that the accessories are driven from the propeller end of the engine or through some kind of flexible coupling.

Should the 12th harmonic of such an engine as described fall in the operating range, considerable trouble could be expected. Reference to Fig. 6 will remind us that the effective torque variation of this harmonic is large, and large vibration amplitudes could be expected. Crankshaft trouble, excessive bearing wear, and valve-gear failures would not be surprising. Fortunately, this harmonic usually lies in the idling range of this type of engine.

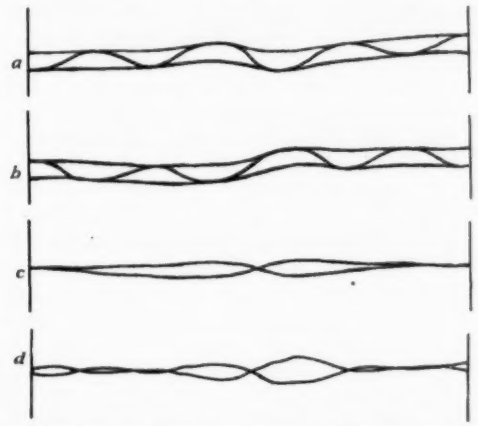


FIG. 18 METHOD OF DISTINGUISHING BETWEEN ASYNCHRONOUS AND RESONANT VIBRATION

(*a*, first revolution of Fig. 5*d*; *b*, second revolution of Fig. 5*d*; *c*, combination of enveloping curves of *a* and *b*; *d*, motion at asynchronous speed.)

In contrast with this we have previously noted that the major critical speed of a radial engine often falls in the operating range. That is why crankshaft vibration may be expected to give more trouble in a radial than in a line engine. Experience has taught us that this is true.

No discussion about torsional vibration would be complete without at least some mention of vibration dampers, and to be orthodox we will not deviate from this custom. Vibration dampers occupy more or less the position of poor relations to the aircraft-engine industry. We will welcome them with open arms if and when they come with real value in their hands, but until such a time we prefer not to be bothered with them. There are several objections to using conventional types of dampers in present-day conventional airplane engines. The three main objections are:

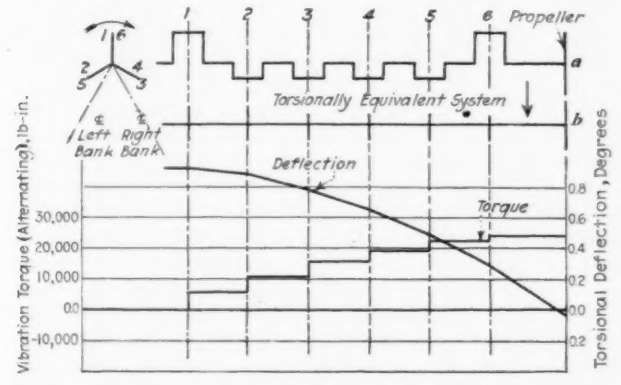


FIG. 19 TORQUE AND DEFLECTION CURVES FOR RESONANT SPEED, LINE ENGINE

1 They add weight
2 They are, in some forms, susceptible to misadjustment
3 They do not eliminate vibration; they only damp it.

Faced with these and other objections, we prefer to modify our engine designs to avoid resonance rather than to resort to dampers. However, we are open to persuasion, and perhaps some day we will make the addition of a vibration damper a further sales attraction, as do the automobile builders now.

AER-55-19a

I—The Problem of Elastic Stability

By L. H. DONNELL,[1] PASADENA, CALIF.

Increasing loading on a structure brings about "failure," either in the familiar way in which the material does not stand up at some point or by the way in which a part under compression reaches a state of instability and buckling occurs. The first of these modes of failure is a failure of material; the second is of a more structural character. The paper discusses preliminary stress calculations, independence of the two types of failure, characteristics of structures influencing each type of failure, relation of the characteristics to weight saving, secondary failures, experimental and analytical methods of studying buckling problems, comparison of different methods of analysis, the simple strut, column formulas, other stability problems, and possible methods of analysis of monocoque construction.

WHEN a structure is subjected to increasing loading of a type such that some part of it is compressed, there are two general ways in which "failure" can occur. The way most familiar to engineers is that in which at some point the material fails to stand up under the conditions of stress—combined perhaps with fatigue, extreme temperature, or other conditions—to which it is subjected. The other way in which failure may occur is that in which some part under compression reaches a state of instability, and buckling takes place.[2]

The first of these modes of failure is a failure of the material at some point or points. The mechanism of this failure takes place on a microscopic or perhaps even a molecular scale and is not well understood. Its laws are therefore studied almost entirely in a purely empirical manner. The second mode of failure is of a more structural character. Its mechanism is on a comparatively large scale, and its laws are quite well understood. It is therefore both practical and useful to study it theoretically as well as experimentally.

Preliminary Stress Calculations

Before we can determine whether, or when, failure of any type will occur at some point in a given structure we must first determine what the conditions are at all points, before it has failed. That is, we must determine how the loads or stresses are distributed throughout the structure, while it is still acting under the simple laws of elasticity. Sometimes the structure is so simple or familiar that this seems self-evident; sometimes the structure is so complex that this part of the problem involves complicated calculations or experiments. Engineers are likely to confuse this determination of the elastic stress distribution in the structure with study of the first type of failure, and to consider that it has nothing to do with buckling. This mistake arises chiefly from the fact that when we are studying the possibility of failure of the material, the determination of the stress distribution may be very difficult (as in some of the cases in the third part, AER-55-19c, of this paper), whereas after we know what the conditions are at each point the question whether failure will occur is usually simple or self-evident. On the other hand, the problem of determining whether buckling will occur or not under certain conditions of stress is in general much more difficult than the problem of determining the condition of stress. Practically all cases of buckling which have been studied theoretically up to the present have been cases in which the distribution of stress is very simple and familiar. But there are plenty of cases to be studied in which the stress distribution is far from obvious—as, for instance, a stressed-skin wing.

As the mechanism of failure of material is on a microscopic scale, this type of failure can start at any point in the structure when a certain stress is reached, even if on all sides of this point the stress is lower. Hence in studying this type of failure the maximum value of the stress is the thing to be found, and the stress distribution must be studied in great detail. On the other hand, buckling is more dependent on average conditions over considerable portions of the structure, so that it is not important to know the stress distribution in such detail. Nevertheless it should be emphasized that determination of stress distribution, theoretically or experimentally, is to be regarded as a preliminary survey of the situation to determine the conditions in all parts of the structure, preparatory to considering the possibilities of any type of failure.

Independence of Two Types of Failure

Having studied the structure as a purely elastic problem, we are ready to determine two things: the loading at which failure of material will start, and the loading at which buckling will start. We can consider these two questions entirely independently, and take the smaller of these two loadings as the answer to the problem. We need not worry about the possibility of a combination failure occurring at still lower loadings.[3] After one type of failure has started, then all conditions are changed, and subsequent events are quite likely to involve a combination with the other type of failure, but the conditions for the starting of each type can be considered to be independent.

Characteristics of Structure Influencing Each Type of Failure

The characteristics of a structure which influence one type of failure have little connection with the characteristics affecting the other type. As for the characteristics of the material used, failure of the first type depends on the strength of the material,

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[2] It is conceivable, of course, that a state of instability may exist in which there is so nice a state of balance that buckling does not, at least immediately, occur. But this is of no practical importance, and it must always be assumed that buckling will take place as soon as a state of instability is reached. Thus a theoretical study of this subject usually consists and need only consist in a study of the conditions for instability—the conditions under which buckling *may* take place.

Part I of a Symposium, "Survey of Problems of Thin-Walled Structures," contributed by the Aeronautic Division and presented at the Pacific Coast Aeronautics Meeting, Berkeley, Calif., June 9 and 10, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] It may be argued that an exception to this may occur in such a case as an eccentrically loaded strut. However, the effect of one type of failure on another would not lower the failure load greatly even here. Also, true instability is impossible in this case, and the problem may be considered as a simple elastic problem up to the moment of failure of material. This question is considered in detail later for the case of struts.

while failure of the second type depends only on the stiffness of the material (that is, the modulus of elasticity, and to a slight extent Poisson's ratio) and is in no way influenced by its strength. The geometrical shape of the structure is of importance in connection with the first type of failure in so far as it determines the maximum stress. The influence of the shape on the second type of failure depends on quite different and more complex characteristics, having to do principally with the ratio between the smallest dimension perpendicular to a compression and other dimensions. Hence it is structures in which, due to weight considerations or other reasons, thin plates or rods are used to carry compressive stresses, that are likely to suffer buckling before failure of the material.

Relation of the Characteristics to Weight-Saving

One of the chief problems of the aeronautical engineer is to design a structure not only capable of resisting a certain loading condition, but having the minimum possible weight. As far as the first type of failure is concerned, this requires the choice of a material having a low weight-strength ratio, while for the second type of failure it favors a material with not only a low weight-stiffness ratio, but also with a low absolute density, so that thicknesses may be as large as possible. It happens that the weight-strength and weight-stiffness ratios for the three materials chiefly used for aeronautical structures, wood, duralumin, and high-strength steels, are not very widely different, so that the lighter materials, allowing greater thicknesses, are evidently more suitable for parts subjected to compression. However, other considerations seem to be favoring the heavier materials, and this makes the buckling problem more and more important.

For maximum weight saving, considering the first type of failure, the detailed design should be such that all parts of the structure reach failure of the material at the same loading. If at the same time the structure can be made in such a stable way that buckling failure does not occur before this same loading is reached, then we may consider that we have reached 100 per cent efficiency in weight saving for the materials and general design employed; to further increase the resistance to buckling does no harm, but also no good in general. This is the ideal, the goal to be aimed for; in some cases it can be attained easily, and in all cases it could theoretically be attained by a sufficient complication of the structure, although this might be quite impractical from economic, aerodynamic, or other points of view. If it is impossible to keep buckling from occurring before material failure, then there is no simple criterion for measuring the maximum of weight saving, but obviously the practical designer needs all the information possible on the laws of buckling.

Secondary Failures

In some cases, when failure once starts, it is so complete that we are no longer interested in the structure. In other cases, the first failure is of a partial or local character, and the loading may be increased until other partial or complete failures occur. The study of these later failures is likely to be very much more difficult than study of the first failure, as the structures are no longer in a simple elastic condition when they take place, but may be in a combination of elastic, plastic, and buckled states. Some problems of this type are considered in the second part of this paper. Of course if the first failure is sufficiently minor or local in character (as, for instance, the local yielding of ductile material around small holes, due to stress concentration), its effect may be neglected entirely in considering the next failure. Another generalization that can be made regarding secondary failures is that if a structure or part of a structure is subjected to compression and fails first by yielding of the material, then a buckling failure is certain to occur at very little increase of the load, no matter what the shape of the structure (unless it is of extremely stable shape). This very important generalization will be discussed in detail later, under struts.

Methods of Studying Buckling Problems, Experimental and Analytical

Let us consider now the general methods of studying buckling or instability. In a complicated structure there are usually many different types of buckling which may occur in different parts of the structure, or even different types which may occur in the same part. Each would occur, if nothing else happened first, at a certain loading of the structure. Our problem is to find this critical loading for each of these types of buckling failure and take the lowest loading as our answer. This is the loading at which the structure will first fail (what happens afterward is a new and more difficult problem, as discussed before). This multiplicity of the different possible types of buckling adds much difficulty to the subject. Not only does it add to the work to have to consider so many different things (although of course with experience many can be eliminated by inspection), but the question of whether we have considered all the possibilities, or have neglected some of them, is a major difficulty.

This makes experience with buckling problems in general and experiments on the particular type of problem in question of very great importance. A mathematical analysis which takes all possible modes of buckling into consideration and leads us inevitably to the correct solution is practicable only in simple cases. Hence in complex cases we often must rely upon experiments to indicate to us what modes of buckling must be analyzed. Even then the analysis is likely to be difficult and to necessitate simplifying assumptions such as the neglect of eccentricities and nonuniformity of material. Hence there is a strong temptation to neglect analysis and rely wholly upon experiments. However, it is obvious that experiments on each particular detailed design are expensive and tell us little about possibilities of improvement, while series of experiments, no matter how extensive, can obviously cover only a limited field, and applications of their results outside this field are dangerous. It would seem that, to have a knowledge of our buckling problems really useful for design purposes, we must have some kind of rational analyses (even if somewhat approximate), using experiments as a guide to and a check upon these analyses and to get an idea as to the variations to be expected in practice due to unknown factors such as eccentricities. In cases where we have not been able to make satisfactory analyses, we may be able to use the results of analyses of simpler but similar problems, or the results of very approximate analysis, as a basis for formulating the results of experiments; such formulations should be much more applicable outside the range of the experiments than completely empirical formulas.

Methods of Analysis, the Elastic-Line Method

There are two general methods for analyzing the conditions under which a structure or part of a structure will become unstable. The first method, which might be termed a "direct" method, consists in setting up a general expression for the lateral deflection (that is, deflection perpendicular to the compression[4]) of the body, in the same manner as deflections are calculated in any elasticity problem—that is, by assuming that there is deflection and applying the laws of equilibrium and Hooke's law to elements of the body. This gives an expression for the lateral deflection at any point of the body in the form of a differential equation involving the external loading (or the stresses produced by it) as a constant factor. Solutions of this differential equa-

[4] In some cases deflections in other directions also take place during buckling and must be considered. This is discussed later under the energy method.

tion represent possible deflected states in which the structure will be in equilibrium. If the problem is one in which instability is possible, it will be found that the only solutions satisfying the differential equation and the boundary conditions are as follows: One solution will be that in which the deflection is everywhere zero and the external loading may have any value; the physical meaning of this is obvious. In addition to this there will be other solutions, in each of which the external loading must have a certain definite value, and the deflections must vary in a certain way over the body, but may have any absolute magnitude;[5] each of these solutions represents a possible type of buckling, and the value which the external loading must have in it is the loading at which this buckling would take place if nothing else happened first. The smallest of these values of the loading is the loading at which buckling will first take place.

A variation of this method of analysis, perhaps more satisfying to practical minds, is obtained by assuming a small initial curvature (or eccentricity or bending moment), and then setting up an expression for the lateral deflection as before. Again, numerous solutions can be found, corresponding to the various possible types of buckling. The deflection given by one of these solutions for any point of the body is no longer zero up to the buckling loading and any value at this loading, as implied in the foregoing solutions, but is a definite function of the loading, of the approximate form:

$$d = \frac{e}{\frac{P_e}{P} - 1} \qquad [1]$$

where d is the lateral deflection at some point due to a loading P, e is a constant proportional to the initial curvature, and P_e is the critical buckling loading corresponding to the solution. Fig. 1 shows the deflection plotted against the load for various values of e. It will be seen that with smaller and smaller initial curvatures the law of the deflection approaches more and more nearly to that implied in the foregoing solution with no initial curvature.

A New Method

Another and less mathematical way of applying this principle of analysis, not previously published, is used later in discussing simple struts. In this an initial curvature is again assumed. Due to this initial curvature, the compressive load produces bending moments along the strut, which cause an additional curvature. This additional curvature produces additional bending moments, which cause another increase of curvature, and so on. This gives the final shape of the bar under any load as a superposition of an infinite series of deflections, starting with the initial deflection. If the loading is below the critical loading, this infinite series converges and gives a finite deflection. When the loading reaches the critical value, the series no longer converges. This method eliminates differential equations, and moreover leads directly to the correct shape of the buckling deflection, so that it is no longer necessary to choose between various possible types of buckling. If the assumed initial deflection happens to be of the same shape as the correct final buckling deflection, then all the increases in deflection will also be of this same shape, and the infinite series becomes a simple geometric series whose sum is well known. If the initial deflection does not happen to be of the same shape as the final buckling deflection, it is an interesting fact that the additional deflections, calculated from it and from each other, become of very nearly the final shape after a comparatively few steps. Hence, for any assumed shape of initial curvature, the first few steps may be calculated by elementary analytical or graphical methods, after which the remainder of the work is the mere summation of a geometric series. This method gives a graphical explanation of the otherwise very puzzling phenomenon of an initial curvature of one shape leading to a final buckling of a quite different shape. It is also very useful for studying the effects of various types of initial curvature and for obtaining simple ratios between the magnitudes of these various types which will produce the same final result. This method has not been applied to other cases than that of a simple strut, but other applications are quite possible.

The Energy Method

The other general method of analyzing the conditions for instability is the "energy" method. This begins with setting up an expression for the total energy change which takes place during a buckling deflection. (The direction or directions of this deflection will be discussed later.) This energy change is the algebraic sum of the work done by the external forces (or by the stresses produced by the external forces) during buckling, and the elastic energy stored up in the body in deflecting it (or the work done against the stresses produced by the deflection). If this total energy change is zero—that is, the work done by the external forces equals the elastic energy stored up—then the deflection can evidently take place spontaneously—that is, by buckling. Hence the equation obtained by setting the total energy equal to zero must be satisfied by the true values for the buckling deflections and loading. This equation is an integral equation for the deflections, involving the loading as a constant factor. It will be found that if we take *any* expression whatever for the deflections and substitute it into this equation, we can solve for a value of the loading which will make the equation satisfied. Obviously all such solutions cannot represent true buckling conditions, even if we limit ourselves to those which satisfy the boundary conditions. The correct solutions are those which represent states of deflection and loading under which the body can be in equilibrium. The only way that we could make it in equilibrium while bent in a wrong shape is by applying additional external forces to it. Hence we can consider wrong solutions as correct solutions for a similar structure which is subjected to additional constraints. Since additional constraints can only increase the stability and therefore the buckling loading, it is evident that the correct solution is that one which leads to the minimum value for the buckling loading.

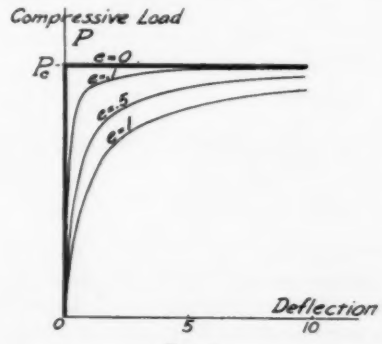


Fig. 1

The physical significance of this can best be understood by imagining that we could plot on paper a quantity describing shapes of the deflections against the value for the loading which we would obtain by substituting the deflection expressions into

[5] In setting up the differential equation, we assume that deflections are small and neglect quantities of a second order of smallness, as in other elasticity problems. If a more exact analysis is made, it is found that conditions change slightly when the deflections become very large. This is of no importance in most applications.

the energy equation. We would obtain a curve such as suggested in Fig. 2*a*. The point at which the curve is horizontal corresponds to the true shape of the deflection and the true buckling loading. Now the fact that the curve is horizontal at this point means that if we make a small error, such as *a*, in the shape of the deflection, we would only make a much smaller error *b* in the value for the loading. The practical significance of this is that if we take an expression for the deflections which satisfies the boundary conditions and is of the same general form as the true buckling deflections and substitute it into the foregoing energy equation, and solve for the value of the loading which makes the equation satisfied, we obtain in general a very good approximation to the true buckling load. Hence if we know in advance the general shape which the buckling deflections will take, we can readily obtain by this means an approximate value for the loading at which the buckling will start. Frequently this is sufficient for our purposes.

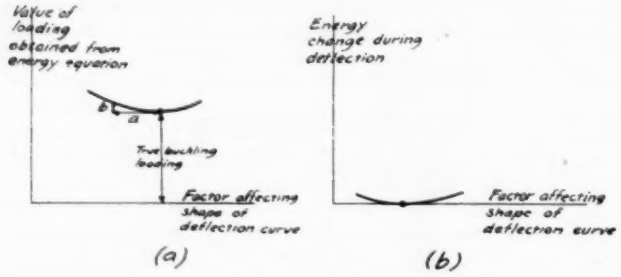


FIG. 2

More often, however, we must have some means of determining, more closely at least, what the correct shape of the deflection is. For this purpose we consider that this correct shape is produced by mixing various arbitrarily chosen shapes (each of which satisfies the boundary conditions) in the proper proportion. It is well known that if we take a sufficient number of curves, such as ordinary sine and cosine curves of various frequencies, and combine them in the proper proportion—that is, multiply each by the proper factor and superpose—we can duplicate any desired curve. Having chosen the curves to be combined, arbitrarily (but with an eye to suitability), all that remains to be determined is the values of the factors which give the proportion of each in the mixture. In our case we make use of the fact that the correct solution is the one which leads to the minimum value for the loading, and determine the factors so as to make this value a minimum. We do this in the usual mathematical manner by solving for the loading in the energy equation and equating to zero the derivatives of the resulting expression with respect to each of the factors. This gives us as many equations as factors, so that it is only a matter of algebra to solve for the factors, put them back in the expression for the loading, and solve for it, or to use an equivalent determinantal solution.

This is the way in which the energy method has commonly been used, but it has the serious disadvantage that the expression for the loading obtained from the energy equation is the quotient of two expressions, each of which contains many terms (in all but the simplest problems). This makes the work of differentiation, and the "mere matter of algebra" of solving the resulting equations simultaneously, very difficult. Now it can be shown that the derivative, with respect to each factor, of the expression for total energy change during deflection is also zero, as suggested by Fig. 2*b*. The physical meaning of this is that if we have a structure in a buckled condition (which means in a condition of equilibrium) and we change the deflection curve slightly by applying additional external forces, the work done by the external forces and the elastic work stored in the structure will both start to increase at the same rate, so that the total energy change remains zero at first. Whether this represents a minimum, a maximum, or a point of inflection for the energy change is of no importance to us. After substitution of the arbitrary expressions for the deflection, the expression for the total energy, in even the most complex problem, reduces to a summation of squares or products of the unknown factors, with constant coefficients (determined by the physical properties of the structure). The partial differentiation of this expression with respect to each factor is therefore easy, and the equations obtained by equating these derivatives to zero are simple linear equations in the factors.

COMPARISON OF DIFFERENT METHODS OF ANALYSIS

Simple problems in buckling can be studied by any of the foregoing methods, by one about as easily as by another.[6] But for very complicated problems the energy method seems to be the only one practical. In many problems, such as the buckling of curved sheets, not only lateral deflections but deflections in all three directions must be considered. To study such problems by the elastic-line method involves simultaneous solution of three differential equations. Moreover, if the structure has discontinuities, as a sheet with stiffeners, the number of differential equations to be solved simultaneously to obtain a complete solution would be still greater. If the energy method is used, on the other hand, such complications merely increase the number of terms in the expression for the energy and in general the number of factors to be solved for. No matter how complex the problem, the mathematical operations remain of the same simple character. However, in general, the work involved, particularly the work of solving the final system of linear simultaneous equations, increases enormously with the number of unknown factors. This is the only real limitation of the energy method; if some easy method of solving such simultaneous equations were available, the analysis of almost any problem would be practicable.

To keep the number of unknown factors, and therefore of simultaneous equations, as small as possible, we must choose the arbitrary deflection shapes so that as few of them as possible need be combined to get a good approximation to the true buckling shape. Buckling deflections are in general of a wave form, so that trigonometric series, sine or cosine, are usually most suitable for this purpose. As is well known, such series can express any shape whatever, and mathematical operations with them are generally simple. Often the actual buckling shape is sinusoidal, so that one term of the series gives an exact solution, and it is seldom necessary to use more than a few terms to get a satisfactory degree of accuracy—especially since, with the energy method, an approximation to the true deflection shape leads in general to a much better approximation for the buckling loading, as discussed before.

THE SIMPLE STRUT

The simplest problem in buckling is that of a simple uniform strut under longitudinal compression;[7] this case will be considered in some detail, as many important principles applying to buckling problems in general can be illustrated by it. This was the first buckling problem to be solved, the solution being given by Euler in 1744. Most engineers did not realize at that time

[6] Both the elastic line and the energy methods are exact statements of the same physical problem. Fundamentally, therefore, they must amount to the same thing, and one form could be changed into the other by the proper mathematical manipulation. The energy method is sometimes mistakenly called an "approximate" method. Actually, if carried out completely, it would lead to an exact solution; however, it lends itself conveniently to approximate solution of problems for which an exact solution is impracticable.

[7] See Wagner, N.A.C.A. Tech. Mem. No. 500.

that Euler's formula gave only the load at which buckling would start, if nothing else happened first. Structural materials at that time were chiefly wood and masonry, and the columns made from these were usually so thick in proportion to the length that failure of the material occurred before buckling. Naturally Euler's formula did not give the correct value for the failing load, and so it fell into disrepute among engineers. In more recent years materials of higher strength have come into use, cross-sections have been reduced accordingly, and buckling failure often occurs before material failure, so that Euler's formula is frequently applicable.

Suppose that a hinged end strut has an initial curvature in the form of a half sine wave, given by the expression $y = d_i \sin \frac{\pi x}{L}$, as shown in the first sketch of Fig. 3(a). Due to the compressive load P, there are bending moments along the strut with the magnitude $Pd_i \sin \frac{\pi x}{L}$, and it can be shown by the usual elementary means that these bending moments cause an additional curvature given by $y = \left(\frac{PL^2}{\pi^2 EI}\right) d_i \sin \frac{\pi x}{L}$, shown by the second sketch in Fig. 3(a). This additional curvature causes the additional bending moments $P\left(\frac{PL^2}{\pi^2 EI}\right) d_i \sin \frac{\pi x}{L}$, which produce the additional deflections $y = \left(\frac{PL^2}{\pi^2 EI}\right)^2 d_i \sin \frac{\pi x}{L}$, shown by the third sketch. Continuing this indefinitely, we see that the total

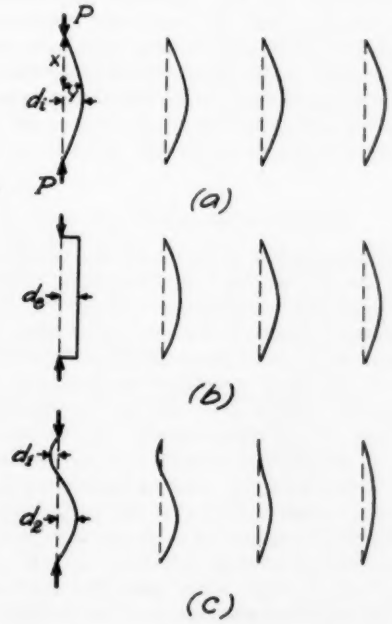


Fig. 3

movement of the center of the bar under the load P is given by the ordinary geometric series

$$d = \left(\frac{PL^2}{\pi^2 EI}\right) d_i + \left(\frac{PL^2}{\pi^2 EI}\right)^2 d_i + \left(\frac{PL^2}{\pi^2 EI}\right)^3 d_i \ldots\ldots$$

$$= \frac{d_i}{\frac{\pi^2 EI}{PL^2} - 1} \ldots\ldots [2]$$

If the load P is zero, this gives the initial deflection; the deflection becomes greater as P is increased, and finally becomes infinite when

$$P = P_e = \frac{\pi^2 EI}{L^2}$$

which is therefore the value of the critical buckling load P_e.

Substituting this in [2], we obtain Equation [1], e being equal to d_i in this case. In plotting this equation in Fig. 1, elastic conditions were assumed. Actually, even if the stress is well below the yield point when buckling begins, the combination of direct and bending stress, as the strut deflects, will eventually reach the yield point. The material on the concave side of the deflected strut will then yield and the stiffness of the strut will fall off rapidly, as indicated in Fig. 4(a). The load which an initially curved strut will take before complete failure, P'_e, will therefore be slightly less than the critical load, P_e, for straight

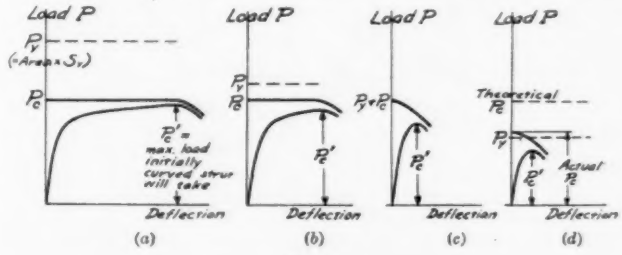


Fig. 4

struts. If the strut is nearly at the yield point when buckling begins, then obviously yielding on the concave side will come at a much smaller deflection, as indicated in Fig. 4(b), and the failing load for the initially curved strut will be still less. Fig. 4(c) and (d) show the conditions when the load at which the material yields is equal to and smaller than the critical load. It is assumed that the material has a sharp yield point; if the yield point is not sharp, analysis is much more difficult, but the general character of the results will be similar.

In the case just considered the initial curvature was the same as the final buckling curvature. If we start with a hinged end strut having the load eccentrically applied at the two ends, as shown in the first sketch of Fig. 3(b), then, due to this eccentricity, the load will cause uniform bending moments along the strut; these will produce a uniform curvature, as shown by the second sketch. This curvature to the arc of a circle is already not far from a sinusoidal curvature, and if we calculate the next increment of curvature, we obtain a shape practically indistinguishable from a sine curve, so that this and all the following increments can be considered to be sinusoidal. Similar results are obtained if we start with other shapes of initial curvature such as shown in Fig. 3(c), the increments of deflection becoming practically sinusoidal in a very few steps. To be sure, if the initial curvature is such that the tendency to buckle one way is exactly as great as the tendency to buckle the other way—for example, if we had symmetry about the center in Fig. 3(c)—then theoretically this symmetry will be maintained in all later steps. However if, by error, we give the slightest preponderance to one side in drawing one of the steps, we will find that this preponderance is greatly magnified in the next step, and very quickly the increments of deflection are all on that side. This is exactly the way it would happen in practice if we should have a state of instability in which there is so fine a state of balance that buckling does not at first occur. The slightest outside disturbance tending in the direction of a true buckling deflection is immediately magnified, and soon this deflection is complete.

For any given initial curvature we can calculate the successive

increments of deflection by elementary methods, until these increments become of constant shape. Comparing these with results obtained with sinusoidal initial curvature, we can easily find an equivalent sinusoidal initial curvature which gives the same results as the given initial curvature, at least for all but the first few steps. Using this in [2], we obtain a formula for the total deflection under any load; this will not be very accurate for small loads, as for such loads the first few increments of deflection are a large part of the total, but it will be a very good approximation for loads near the critical. The case of a strut with initial eccentricity of loading d_e, Fig. 3(*b*), is found to be equivalent to that of a strut with sinusoidal initial curvature such that $d_i = 1.25\ d_e$; and for the case shown in Fig. 3(*c*), where the initial curvature consists of two geometrically similar sine curves, it is found that $d_i = d_2 - d_1$.

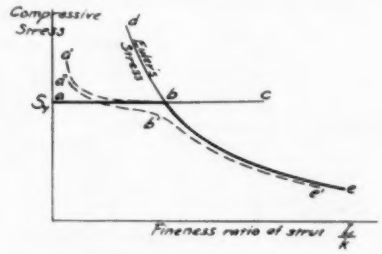


FIG. 5

By analogy or by repeating these methods we find that the critical compressive loads for struts with any end conditions are given by the general formula

$$P_c = \frac{n\pi^2 EI}{L^2}$$

or, dividing through by the area, the critical compressive stress is

$$S_c = n\pi^2 E \left(\frac{k}{L}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

where k is the smallest radius of gyration of the cross-section. The constant n depends on the end conditions, and has the values 1, 4, and $^1/_4$ for struts with both ends hinged, both ends fixed, and one end fixed and the other free, respectively. In practice, struts are usually fastened at both ends, with some frictional or elastic resistance to rotation. For such cases the value of n must evidently lie somewhere between the limits 1 and 4. It can be determined analytically if the end conditions are known, but is usually found experimentally.

Fig. 5 shows the compressive stress which produces failure, for a simple strut with given end conditions, plotted against the "fineness ratio" of the strut, L/k (the length divided by the smallest radius of gyration of the cross-section). The horizontal line *abc* shows the stress at which yielding of material would occur if nothing else happened first. The curve *dbe*, plotted from Euler's formula [3], shows the stress at which buckling would occur if nothing else happened first. Obviously, the heavy bent line *abe* gives the stress at which some kind of failure will first occur. As we increase the fineness of the strut, at first it will fail by yielding of the material (the line *ab*), then from a certain point on it will fail by buckling (the line *be*). If the material does not have a sharp yield point, but shows a rounding of the corner between the elastic and yield lines in its stress-strain curve, then there will be a corresponding rounding of the corner between the lines *ab* and *be*.

The line *a'be* shows the stresses at which complete or ultimate failure occurs, as determined by experiments and analysis.[8] It will be seen that it differs appreciably from the line *abe* only at very low values of the fineness ratio—that is, for extremely short, stocky columns. As we are not likely to find fineness ratios of this order in aeronautical structures, the simple law given by the line *abe* is sufficient for the aeronautical engineer. The physical meaning of this is that if a compressed strut fails first by yielding of the material, then complete failure (by buckling) will certainly occur at very slight increase of load. This principle applies also to other types of structures under compression, as discussed previously.

Column Formulas

Many attempts have been made to construct empirical formulas which would formulate experimental results on struts of various fineness ratios. From the foregoing discussion it is apparent that such a formula should represent the curve *abe* or *a'be*. As most of the formulas were made on the basis of incomplete experiments and without recognition of the exact problem involved, they are in general not very successful in this. Of course, actual struts always have a certain amount of initial curvature or eccentricity, and their behavior is as indicated by the line *a''b'e'*, all failing stresses being below those for corresponding straight struts by an amount depending on the ratio of the initial eccentricity to k.[9] Since this amount of initial curvature is likely to be small in well-made structures, and is in any case entirely undeterminable in advance, the most practical principle seems to be to let the factor of safety take care of this uncertainty (allowance for undeterminable and therefore unanalyzable factors being the legitimate use for a factor of safety) and to base design on the simple law represented by *abe*, which gives the limiting failure load for good construction. There would be no particular advantage in having a single formula to express this law. All the designer need do is to take the Euler stress or the yield point stress—whichever happens to be the smaller—as the failing stress.

Effect of Shear Deflection

Euler's formula was derived by considering the bending deflection of the strut due to the compressive load. The load also produces transverse shear in the strut, and, strictly, the deflection due to this should be added to the bending deflection in calculating the critical load. As in beam problems, if the struts are of rectangular or ordinary structural cross-sections, the shear stresses are small compared to the bending stresses, and the shear deflections are negligible compared to the bending deflections. But in aeronautical construction, to save weight, the parts of the strut taking the shear are commonly lightened up until they are stressed as highly as the parts taking bending stresses. The deflections due to shear will then be of the same order of magnitude as bending deflections, and the critical load for the strut will be much lower than that given by Euler's formula. This important question has been analyzed by Timoshenko and others for the case of a latticed column.[10]

Strut With Elastic Support

Practically all the more complex cases of buckling can be considered to be related to the case of a simple strut elastically supported along its length. This case is therefore worth brief consideration, although it is seldom found in simple form in practice. It was analyzed by Engesser and others for the case of uniformly

[8] See Engesser, V.D.I., 1898, p. 927; von Kármán, *Forschungsarbeiten*, 1910, no. 81.

[9] See Ros u. Brunner, Proc. Int. Cong. App. Mechanics, 1926; Westergaard and Osgood, Trans. A.S.M.E., 1928.

[10] See Timoshenko, "Strength of Materials," part II, p. 592.

distributed support, and by Zimmermann for the case of concentrated supports.[11] It is found that in general such a strut will buckle in a number of waves or half-waves. The resistance of such a strut to buckling is due partly to the stiffness of the strut and partly to the stiffness of the elastic support. Obviously the stiffness of the strut tends to keep the number of waves as small as possible, while the stiffness of the elastic support gives less resistance to buckling if the number of waves is large. The actual shape of buckling must be a compromise between these two tendencies, and the number of waves increases with the ratio between stiffness of the support and stiffness of the strut. As the number of half-waves is never fractional, the relation between the critical stress and the physical conditions is discontinuous as the number of half-waves changes from one number to the next. Fig. 6 shows the theoretical buckling stresses for struts of various lengths, the other physical properties and the elastic support per unit length remaining constant. Curve 1 shows the stress required to produce buckling in the form of one half-wave, curve 2 the stress required to produce buckling in the form of two half-waves, and so on. Obviously, the strut will fail first at the stresses given by the irregular heavy line. It will be noticed that as soon as the number of half-waves exceeds two, the buckling stress is practically independent of the length. This means that when the number of half-waves exceeds two, the buckling stress is also practically independent of the end conditions—that is, it makes no difference whether the ends are hinged or fixed—as a change in the end conditions can be considered as equivalent to a change in the effective length.

The photographs (see Fig. 9) show a model constructed to demonstrate the buckling of a strut with various degrees of elastic support. A vertical wire, representing the strut, is elastically supported by being attached to the ends of a number

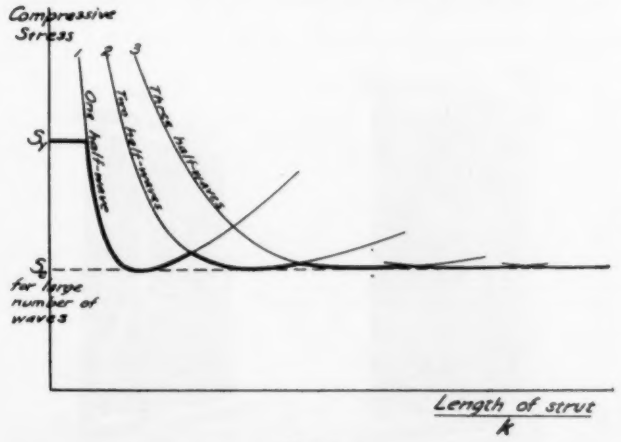


FIG. 6

of cantilever horizontal wires, as shown. By turning the knob at the top, the clamping point of the horizontal cantilevers can be moved, thereby changing their effective length and therefore their stiffness. If the strut is buckled, and the knob is then turned, the shape of the strut changes from one half-wave (as at A), to two, three, and four half-waves (as at B), and so on, the changes taking place quite suddenly.

FLAT PANELS

The stability of flat sheets of rectangular outline was analyzed by Bryan for the case of a plate hinged on all four sides and compressed on two sides, and by Timoshenko for other cases.[12] Plates compressed on two ends can be considered roughly as a series of struts side by side, with their lengths, L, in the direction of the compression. These struts are elastically supported by virtue of the fastening of the sheet along the sides, which causes buckling to be resisted by the stiffness of the sheet against lateral bending (with additional effect from resistance to twisting). The stiffness of this elastic support obviously varies inversely with the width perpendicular to the compression, w. A general formula for the uniformly distributed stress on the ends which will cause the plate to buckle is:

$$S_c = KE\left(\frac{t}{w}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

where t is the thickness of the sheet, and K depends on the ratio of length to width, L/w. Fig. 7 shows values of K for various

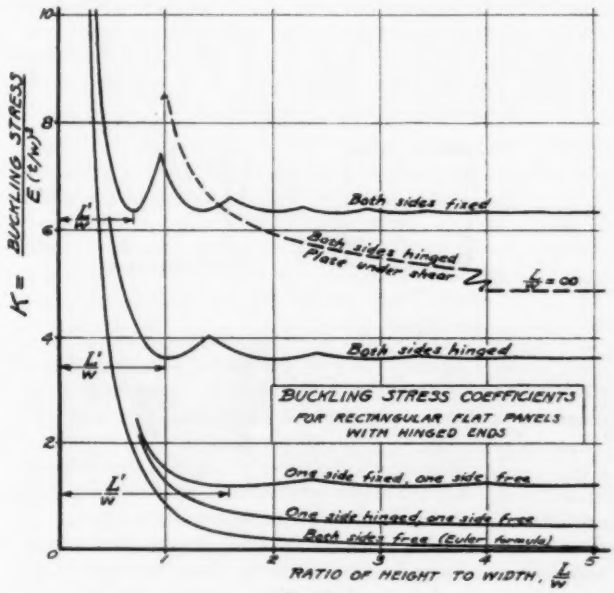


FIG. 7

edge conditions. It will be seen that in all cases, when L/w exceeds 2, K becomes practically constant, and has the following values for the cases given: ends hinged, sides hinged, 6.3; ends hinged, sides fixed, 3.6; ends hinged, one side fixed, one side free, 1.2; ends hinged, one side hinged, one side free, 0.4. (Poisson's ratio is taken as 0.3.) For the other extreme, when L/w is very small, the elastic support is also very small and the critical stress approaches that given by Euler's formula, with $k^2 = t^2/12\,(1 - \mu^2) = t^2/10.9$:

$$S_c = \frac{n\pi^2}{10.9} E\left(\frac{t}{L}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

The plate buckles in a half-wave or a quarter-wave in the w direction. In the direction of L it in general divides up into a number of half-waves. The lengths of these half-waves, L', will be approximately as given in Fig. 7, if the number of waves is large. It is evident that cross-stiffeners, running in the direction of w, would have to be spaced considerably closer than L' to do much good. If the spacing of such stiffeners is equal to or a multiple of L', they will be completely wasted, as the nodes of the buckling will be at the stiffeners. Longitudinal stiffeners will be much more effective; and if longitudinal stiffeners are

[11] See Timoshenko, Handbuch der Phys. u. Tech. Mechanik, vol. IV, pp. 106, 108; Klemperer, Trans. A.S.M.E., 1933.

[12] See Timoshenko, "Strength of Materials," part II, p. 604; Sezawa, Report of Aer. Res. Inst., Tokio Univ., no. 69; Mathar, N.A.C.A. Tech. Mem. No. 592.

used, the cross-stiffeners will be much more effective, through supplying additional elastic support to the longitudinals.

The critical stress for a flat rectangular plate acted on by uniformly distributed shear stresses on all four sides, and hinged on all edges, is also given by [4] and Fig. 7, S_c being taken as the shear stress. In this case buckling is caused by the compressive stress at 45 deg to the sides; the tensile stress at right angles to this helps to prevent buckling. The plate buckles in one half-wave in the direction of the shortest side, and usually in many waves in the direction of the longest side, the nodes being inclined.[13]

The foregoing are all theoretical results. Experiments indicate that they are accurate for very flat sheets; however, sheets thin enough to buckle before yielding are likely to have initial eccentricities of the same order of magnitude as the thickness, so that this factor usually plays a much more important part than it does in the case of struts. No actual figures can be given for the influence of eccentricity, as little work has been done on the subject; and in any case, as in all other types of buckling problems, the amount of eccentricity to be expected depends entirely on the carefulness of the construction.



FIG. 8

As in the case of struts, the edge conditions are in practice likely to be somewhere between the hinged and clamped conditions. However, there are many cases where pure hinging can be assumed, in spite of the lack of mechanical provision for it. Thus the four sides of a square box strut or the two sides of a compressed angle section will obviously buckle as shown in Fig. 8, in such a way that the angles between sides at the corners remain unchanged, and there will be no moments at the edges. However, if the adjoining sides are unequal, or any other condition makes them tend to buckle differently, then there will be interference between them, and the edge conditions will be between the hinged and the fixed.[14]

The general buckling of a sheet having different stiffnesses in the two directions, as a corrugated sheet or a sheet with many stiffeners, has been investigated by Reissner.[15] In such sheets the stiffness of the elemental struts into which we may consider the sheet to be divided are much greater than for a corresponding plain sheet, while the elastic support, due to transverse stiffness, is less. Hence the simple Euler's formula applies up to much greater values of L/w than for plain sheets.

CYLINDERS

The buckling of a cylinder under uniform axial compression can be considered equivalent to that of a group of struts running in the axial direction, under an elastic support due to the curvature. When a curved sheet buckles away from or toward the center of curvature, it must stretch or compress circumferentially. This produces circumferential stresses which resist the buckling like an elastic support. This problem was analyzed by Southwell and Timoshenko[16] by assuming that the buckling would be symmetrical about the center, with waves of definite length in the axial direction. This gives the critical stress as:

$$S_c = 0.6\, E\, \frac{t}{R} \qquad [6]$$

where R is the radius and Poisson's ratio is taken as 0.3. Experiments indicate that thin cylinders actually buckle in a definite number of waves about the circumference, at a stress much less than that given by [6]. The length of the waves in the circumferential direction is usually a small part of the circumference, which seems to indicate that where the stress varies along the circumference, as in cases of pure bending of a cylinder or combinations of axial compression and bending, buckling can start locally when the maximum compressive stress reaches about the value previously given for uniform compression. Numerous attempts have been made to obtain a theoretical explanation for the observed behavior of cylinders in compression and bending, but there seems to be still much to be explained.

Buckling of a cylinder under torsion is discussed by the author in a paper which will soon be published as a report of the N.A.C.A. A theoretical solution is given which is in reasonable agreement with experiments. Cylinders under torsion buckle in spiral waves. In the circumferential direction there are a number of waves, the number varying from a large number for very short cylinders to two for long cylinders. In the latter case the cross-section distorts to an ellipse.

The stiffness of flat panels is greatly improved by using corrugated sheet instead of flat sheet. Both theory and experiments indicate, however, that there is no such advantage in using corrugated sheet in making cylinders or curved panels. This is because the resistance to circumferential expansion or contraction is much lower with corrugated than with plain sheet construction.

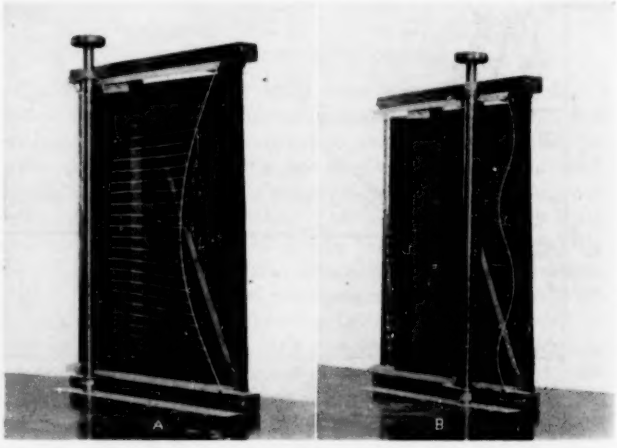


FIG. 9

The beneficial effect of curvature depends entirely on this circumferential resistance. With corrugated construction the decreased benefit from curvature just about balances the benefits from the increase in longitudinal stiffness. A construction composed of plain sheet with closely spaced stiffeners, on the other hand, has about the same resistance to circumferential contraction and expansion as plain sheet, and therefore gives the full benefit of the increase in longitudinal stiffness.

A practical advantage of corrugated construction, however,

[13] See Southwell and Skan, Proc. Int. Cong. for Applied Mechanics, 1924, p. 326; Bergmann and Reissner, *Z. für Flugtechnik u. Mot.*, vol. 18, p. 475, vol. 20, p. 18; Bollenrath, N.A.C.A. Tech. Mem. No. 601; Seydel, N.A.C.A. Tech. Mem. No. 602.

[14] Some problems involving the application of plate formulas are considered by Trayer and March, N.A.C.A. Report No. 382.

[15] See Reissner, *Z. für. Flugtechnik u. Mot.*, vol. 20, p. 475, and vol. 21, p. 306.

[16] See Timoshenko, Handbuch der Phys. u. Tech. Mechanik, vol. IV, p. 143; N.A.C.A. Tech. Notes Nos. 357 and 370; Aero. Res. Comm., Rep. and Mem. No. 1185; Sanden and Tölke, vol. 3, p. 24.

especially when the curvature is not great, is that eccentricity is not nearly so much of a problem. Eccentricities of the order of the thickness of the sheet greatly affect the resistance of plain sheet, but have little effect with corrugated sheet, as they are small compared to the effective thickness. This applies also to flat panels. Eccentricities in general are of considerable importance in connection with the buckling of curved sheet, but probably not as much so as in the case of flat sheet, as the curving of the sheet tends to remove eccentricities.

Curved Panels

Curved panels, supported or fixed along the sides and compressed in the direction of straight elements, can be considered equivalent to a system of struts side by side, with an elastic support due partly to the curvature (that is, to resistance to circumferential tension and compression as in the case of complete cylinders), and partly to the support of the sheet along the sides (that is, to resistance to lateral bending and twisting). The effect of the side support is usually small compared to the effect of curvature, unless the angle subtended by the panel is small. Hence for most purposes the critical stress can be considered to be the same as for a complete cylinder; in any case this assumption will be on the safe side.

Other Stability Problems

Many other types of stability problems have been investigated, such as the lateral stability of beams and the stability of tubes and rings under external pressure,[17] but these are seldom of interest to the aeronautical engineer. The greatest need at present seems to be for more information on the subject of cylinders of round or other cross-section, particularly of stiffened construction.

Stiffened Construction

The obvious purpose of longitudinals and bulkheads, as far as resistance to buckling is concerned, is to increase the average stiffness of the shell in the longitudinal and circumferential directions. This stiffness could also be obtained by using thicker sheet, and this has the advantage that longitudinal and circumferential stiffness is supplied simultaneously by a single element, the sheet; whereas a system of stiffeners must be a double system, each part of which is useful for one direction only. However, the greater depth possible with stiffeners more than makes up for the disadvantage of this necessary duplication, and experience indicates that the necessary stiffness can usually be obtained with less weight with stiffeners than with sheet; where extreme lightness is necessary, the stressed sheet must be eliminated entirely, as in dirigible construction. An analytical study of this whole question would be of great value. Another great practical advantage of the use of stiffeners lies in the reduction of the importance of eccentricity, due to the greater effective depth of stiffened construction, as discussed before in connection with corrugated sheet.

Possible Methods of Analysis of Monocoque Construction

The resistance to general buckling of a stiffened shell can obviously be studied approximately by considering it to be equivalent to a homogeneous shell having the average stiffness in each direction produced by sheet and stiffeners together. One question which arises in applying this method of analysis is that of the influence of the sheet on the effective moment of inertia of the stiffener. This question must be answered by use of the methods described in the third part of this paper. If it is desired to similarly study the ultimate resistance of the shell after the sheet has buckled, then the influence of the sheet must be studied by the methods in the second part of this paper (AER-55-19b). Experimental work on the resistance of shells is being carried out in many places. At the California Institute both experimental and theoretical work is in progress. Two machines have been constructed, for small and for large specimens, capable of testing cylinders, or actual wing or fuselage sections, in uniform or varying bending, in torsion or compression, or in any combination of these loadings.

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[17] See Timoshenko, "Strength of Materials," part II, pp. 609 and 598.

AER-55-19b

II—Strength of Thin Metal Structures Beyond the Stability Limit

By E. E. SECHLER,[1] PASADENA, CALIF.

When thin sheet metal was first used for aircraft, the design was in accordance with the usual laws of elasticity, and when buckling occurred in any part it was considered as failure. Frequently it was observed, however, that the buckling disappeared after the load was removed, without permanent deformation. This led to the study of thin sheet metal when used above the stability limit. The paper considers two methods of designing structures for thin sheet metal—one is to endeavor to make the buckling load correspond to a stress equal to the elastic limit of the material, and the other is to allow the structure to go into the buckled state at a stress below the elastic limit and then to determine the allowable load on the structure before permanent deformation would take place. It may be possible that a photoelastic study of buckled sheets would lead the way to a solution.

WHEN thin sheet metal was first used as an aircraft material, the structures to be made of this material were designed in accordance with the usual laws of elasticity, and the structure was considered as having failed when the buckling load had been reached in any part of the structure. It was noticed, however, that frequently after the buckling load had been reached, and even exceeded, on such a structure, the removal of the load caused the buckles to disappear, and no permanent deformation remained in the material. Further investigations showed that the load could be increased in some cases quite far beyond the stability limit before the material suffered any permanent deformation upon the removal of the load. This led to the study of thin sheet metal when used above the stability limit.

The foregoing discussion can be applied to any structure that has less resistance to one type of stress than to another. Take, for example, a fuselage or wing section made up of a number of light tubes or other strut members some of which are subjected to tension stresses and others to compressive stresses. If the tubes are of the same size, which may be necessary for design or construction reasons, those loaded in compression may fail by elastic buckling as columns before any failure occurs in the tension members. These buckled members will then cease to carry their proportion of any additional load, and this additional load must be taken up by either the tension members or by other compression members which are not stressed up to their critical value.

The aforementioned action, i.e., the elastic buckling of a part of the structure and the consequent redistribution of the load, may take place several times before any part of the structure is stressed beyond the elastic limit. If the structure is made up of a finite number of members, it is possible, although it may be laborious, to calculate with reasonable accuracy the loads at which the first buckling occurs and to determine the member undergoing this buckling. This is done by the usual methods of structural design, taking into consideration the critical column values of the members under compressive loads. The buckled member can then be replaced by a constant force, since a buckled strut offers very little additional resistance to further deflections, and the structure be recalculated for additional loading with the new distribution of stresses. This is continued until one of the members is stressed beyond the elastic limit or the structure becomes unstable due to members dropping out of action. This type of structural design follows the ordinary methods of determinant and indeterminant structures and will not be discussed further in this article. The limiting case of the foregoing problem is that of the thin sheet which has comparatively no resistance to bending and compressive stresses in relation to its resistance to tensile stresses, and it is this case which will be considered in more detail.

There are two possible methods of designing structures employing thin sheet material. The first is to endeavor to make the buckling load correspond to a stress equal to the elastic limit of the material. For this method of design the normal laws of buckling of thin sheets would be used, and buckling would be a true failure, since the deformations would not disappear upon removal of the load. The other method is to allow the structure to go into the buckled state at a stress below the elastic limit of the material and then to determine the allowable load on the structure before permanent deformation would take place.

An inspection of the first method shows that it may lead to sheet sizes and stiffener spacings which are not at all desirable from the standpoint of economical construction. For example, consider a sheet of 0.020-in. Dural, 9 in. long and 3 in. wide, supported on four sides and loaded with a uniform shear. The buckling stress is given by[2]

$$p_e = k \frac{\pi^2 E t^2}{12(1-\mu^2)b^2}$$

and for $a/b = 3$, $k = 6.1$, and for $E = 10^7$, $t = 0.020$, $b = 3$, and $\mu = 0.3$, $p_e = 2450$ lb per sq in., which is obviously much below the elastic limit of the material, and if the aircraft were to be designed using this method, the structure would be very heavy because the material would not be used to the best advantage.

On the other hand, if a reasonable value for the allowable stress in shear is taken, say, $p_s = 15{,}000$ lb per sq in., the value of b must be reduced to 1.2 in. to prevent buckling before this stress is reached. This gives a panel of 9 in. by 1.2 in., which means numerous stiffeners and expensive manufacture. However, if the original panel is allowed to buckle at the critical value, it can be loaded in the buckled state until the elastic limit of the material is reached in some point. This allows an economical construction and still permits an efficient utilization of the strength of the material.

The simplest illustration of the buckling of a thin sheet under load and the subsequent redistribution of the load to stresses of

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Part II of a Symposium, "Survey of Problems of Thin-Walled Structures," contributed by the Aeronautic Division and presented at the Pacific Coast Aeronautics Meeting, Berkeley, Calif., June 9 and 10, 1932, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Timoshenko and Lessells, "Applied Elasticity."

another type is given in the case of a plain supported sheet under a shearing load (see Fig. 1). Considering an element in the sheet, we find that the shearing stress is resisted by two mutually perpendicular stresses, one tensile and one compressive, at an angle of 45 deg to the original shear. Since, for very thin sheets, the resistance to compressive stresses is practically zero, the sheet will buckle into waves running perpendicular to the compressive stress as soon as a very small shear load is applied. Additional load cannot then be carried in compression, but is all carried by the corrugated sheet in the form of tensile stresses parallel to the buckles (see Fig. 2). The magnitude of these tensile stresses is equal to twice the value of the shearing stress on the sheet,

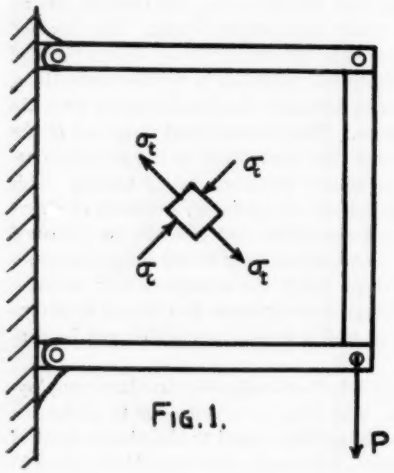


FIG. 1.

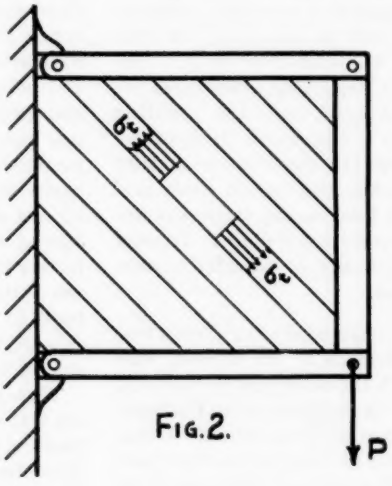


FIG. 2.

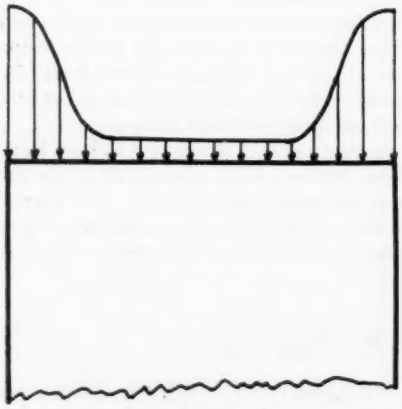


FIG. 3.

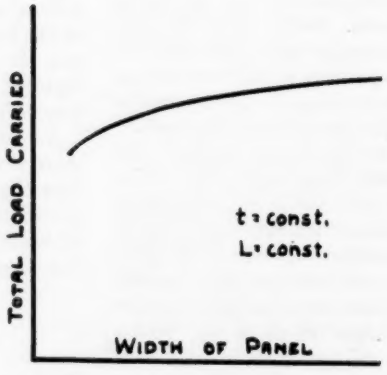


FIG. 4.

$S_t = 2S_s$, and, for a sheet supported by infinitely rigid members, the direction of the tensile stress is at 45 deg to the original shear.

The foregoing problem has been treated in great detail by Herbert Wagner, of Danzig, Germany, and his work is available to the designer in a number of translations in English.[3] In his work he first considers the case of a deep beam with a thin sheet web subjected to a shearing load, taking into account the stresses introduced in the uprights and the flanges by the diagonal tension field set up in the web. At very low values of the shearing stress the sheet goes into the buckled state, forming a tension field of stress in the web. The whole shear load is then transmitted as tension diagonally across the web of the beam. The first calculation is made assuming that this tension acts at 45 deg to the original shear load. These tensile stresses tend to produce bending in the beam flanges and to load the uprights between the flanges in compression. The deflection of these two members alters the distribution of stress in the tension field and the beam is then recalculated w th the altered distribution. This method can then be carried to any degree of accuracy desired. However, Wagner shows that by making certain approximations at the beginning of the analysis, the accuracy is usually sufficient after the first calculation.

The foregoing treatment is then expanded to the consideration of a beam with uprights which are not vertical, and it is found that for any beam there is an optimum angle at which the uprights should be placed to give the maximum strength-weight ratio. He also considers the effect of the compression loads on the uprights, taking into consideration their critical value as columns and the effect of the deflection of these uprights on the stress distribution as a whole. The case of a beam with taper is discussed, and the effect of this taper on the stress distribution in the web, uprights, and flanges. The whole discussion is very complete and is illustrated by examples which show the practical applications of the method. None of the formulas or results will be given here, as it would be impossible to give formulas without going into the assumptions and conditions under which they apply.

A few preliminary tests have been made on thin sheet subjected to shearing stresses, and the results of these tests were given in an article by Joseph S. Newell late in 1930.[4] The tests were made on both plane and stiffened sheet, and a comparison was made of the strength-weight ratios of the panel. It was found in these experiments that for the shear panel used, 24 in. by 36 in., the stiffened sheet gave a slightly higher strength-weight ratio for sheets of from 0.040 to 0.050 in. in thickness, but for other thicknesses the strength-weight ratio of the unstiffened sheet was better. This seemed true regardless of the number of stiffeners used. These experimental tests are being continued on thin sheet in shear, and more data may be available to the designer in the near future.

The investigation of curved panels under shear loads has not yet reached the stage where any reliable results are available to the designer. A few tests are mentioned by Wagner in Tech. Mem. No. 490, and in his article on flat sheets in shear he states that he will treat curved sheets in a later report.

The next problem to be considered is that of thin sheet used in compression. Obviously, if a thin sheet is unsupported and

[3] Herbert Wagner, N.A.C.A. Tech. Mem. No. 490; N.A.C.A. Tech. Mem. Nos. 604, 605, 606; Trans. A.S.M.E., 1931, paper AER-53-18, "Sheet Metal Airplane Construction."

[4] Joseph S. Newell, *Airway Age*, November and December, 1930.

loaded with a compression in its plane, the failure by buckling will come almost immediately. However, if the edges of the sheet are supported in some manner, then a different action takes place. The edges being restrained against motion in one or more directions has a tendency to delay the buckling phenomena and the sheet will support a load in compression. The value of this load can be determined by the well-known methods of Bryan and Timoshenko. This classical method assumes a uniform stress distribution up to the time the sheet buckles, and the solution is carried out using the boundary conditions corresponding to the edge restraint.

The values given by the foregoing method show that, except for extreme values of breadth to length of the sheet, the buckling takes place at a stress which is far below the elastic limit of the material. The question then arises if the sheet could not be used to sustain a load beyond the critical buckling load. This problem is being investigated at the present time by several different groups. Experimental work has been done at the National Advisory Committee, Stanford University, Massachusetts Institute of Technology, and the California Institute of Technology. Some of this work has been published, and more investigations by these groups are now under way. The N.A.C.A. has published its work in a late Technical Report,[5] and J. S. Newell has combined some of this work with that done at Stanford and M.I.T. in the two articles of *Airway Age*.[4] A preliminary semi-theoretical treatment of plane sheets loaded in compression suggested by Th. von Kármán at C.I.T. appeared recently,[6] and experimental work is now being carried out as a check on the theoretical values.

The important result of all of these experiments is that the strength of a supported sheet (in all of the foregoing cases simple support was used) is only slightly affected by variations in length and breadth of the sheet. For very narrow sheets the strength increases as the width increases, and the failure is due to the fact that the elastic limit of the material is reached before buckling takes place. For wider sheets, however, the sheet tends to buckle, and when this buckling takes place, a redistribution of the load occurs. The edges of the sheet being supported continue to carry additional load, but that part of the sheet at some distance from the supported edges buckles and carries little or no more additional load in its buckled state. The stress distribution would then be something like that indicated in Fig. 3.

In the work of the N.A.C.A. the experimental values of the allowable load on a series of sheets of different widths, thicknesses, and materials were taken and an empirical equation was given for the allowable load on any sheet. This equation was of the form

$$P = K_1 t^2 - K_2 t^3$$

where K_1 and K_2 are constants depending only upon the material used and not upon any dimension of the specimen. Deviations from this value were not great for sheets larger than 6 in. wide (all specimens considered were 24 in. long), while below this width the strength fell off rather rapidly. The experimental values obtained for the wider sheets were lower than would be normally expected, and this was explained by the fact that a comparatively light loading bar was used which had a tendency to deflect for the wider specimens.

The work at C.I.T. on plane unstiffened sheets was based primarily on an assumed stress distribution after buckling. The assumption was made that an "effective width" of the sheet near the stiffener was carrying the full load and that over this effective width the stress distribution was uniform. There were then two possible methods of considering this effective width. One was to consider it as a strip, supported on three sides, with the fourth side free, and the other was to consider that the tangent at the inside edge of this strip was horizontal, in which case the two strips could be considered together, disregarding all of the material in the center. It is obvious from an inspection of the buckling of such a panel that the correct answer must lie between these two methods. Both methods give, for the equation of allowable load, an equation of the form

$$P = C \sqrt{E\sigma_y}\, t^2$$

where E is the Young's modulus, σ_y the elastic limit, and t the thickness of the sheet; C is a function containing Poisson's ratio of the material, and the form of this function depends upon which of the aforementioned assumptions is used. This function is dimensionless, which leaves the equation dimensionally correct. It is believed that an experimental value of this constant C can be determined to be used as a design value.

The experimental work at C.I.T. has given results which are in the main similar to those obtained in other places. Great care was taken, however, to make the loading bar used in these tests rigid, and as a result the dropping off of the allowable load at the larger widths of sheet was not experienced. The curve of maximum load plotted against width of sheet seemed to indicate an asymptotic approach to a maximum, as would be expected from a stress distribution similar to Fig. 3. This curve is of the type shown in Fig. 4. It is hoped that a simple correction can be obtained from these curves to take account of the variation of P_{max} with the width of the sheet.

For plane panels with stiffeners, Newell suggests that the strength of the stiffeners tested as columns be added to the strength of the sheet, as given by the empirical formula:

$$P = K_1 t^2 - K_2 t^3$$

The comparison between this value and that found experimentally is given in the articles mentioned.

A slightly different line of attack is being followed at C.I.T., where the effective width of sheet is considered as acting in conjunction with the stiffener section to resist the compressive load. No experimental tests have as yet been made on stiffened sheet at C.I.T.

Mr. Newell's article also considers the case of curved sheet with and without stiffeners, and he gives a series of curves showing the relationship of curvature and number of stiffeners to the maximum allowable compression load. For stiffened, curved panels, the results seem to indicate that more research is necessary before anything of value can be given to the designer. Dr. Donnell, one of the present authors, is making a study of this problem for cylindrical sections, and tests will also be made on individual panels.

One problem which has not as yet been considered is that of thin sheet panels loaded in combined shear and compression. The experimental apparatus for this problem would be rather complicated, and it is also probable that with a complete solution of the two problems of pure shear and pure compression, these solutions could be combined in such a manner as to offer a solution to the problem of combined stress.

As the problem of thin sheet in the buckled state is essentially a problem of the stress distribution after buckling, it may be possible that a photoelastic study of buckled sheets would lead the way to a complete solution. Some photoelastic work has been done in the case of shear-stressed plates, and it may be possible to apply this method of study to the problem of the thin plate under compressive stresses. Some preliminary investigations of this method will probably be made soon at C.I.T.

[5] Louis Schuman and Goldie Back, N.A.C.A. Report No. 356.

[6] "The Strength of Thin Plates in Compression," by Th. von Kármán, E. E. Sechler, and L. H. Donnell, Trans. A.S.M.E., 1932, paper APM-54-5.

AER-55-19c

III—Analysis of Some Typical Thin-Walled Structures

By TH. VON KÁRMÁN,[1] PASADENA, CALIF.

The structural skeleton of an airplane of the orthodox type can be considered as a truss, and the structural analysis uses, in general, the familiar methods. Also the structural analysis of the spar-rib type of wing system can be done by the theory of elastically supported continuous beams. The structural analysis of thin-walled structures is a much more complicated problem. There are different methods of attack. A survey of typical cases shows that there is a possibility of producing usable information. Experiments are being carried out on the stability and ultimate failure of cylinders in compression, bending, and torsion.



THE structural skeleton of an airplane of the orthodox type can be considered as a truss, and the structural analysis uses, in general, methods familiar in structural engineering. Also the structural analysis of a spar-rib type of wing system can be done by application of the theory of elastically supported continuous beams. The structural analysis of thin-walled structures is a much more complex problem, because the elastic behavior of the sheet as a structural element is more complicated than the behavior of the members of a truss. There are different ways to attack the problem:

(*a*) We can consider the elastic behavior of a flat or curved sheet under a given load system and consider the heavier elements as supports. This kind of analysis may be successful in the case that the stiffness of the supports is very large compared to the stiffness of the sheet, and so the elastic deformation of the supports, i.e., of the bulkheads or stiffeners, can be neglected in the calculation of the stress distribution or stability of the sheet. In these cases the stress analysis of the sheet gives valuable information on the local strength of the sheet itself and helps us to prevent local failure, but apparently we do not get information as to the resultant strength of the whole system.

(*b*) There is an academic possibility to consider the structure as an elastic system, composed by beams and plates, and to calculate stress distribution and stability, applying the equations of plates or shells to the sheets and the differential or partial differential equations of beams to the stiffness. Unfortunately, this kind of computation affords a very elaborate arsenal of mathematical methods and is handicapped by many uncertainties in the assumptions which have to be made as to the connections between the elements. Hence, except for some simple fundamental cases, this possibility will be of little value for the designing engineer.

(*c*) The third and most promising procedure is to consider some typical combinations of sheets and stiffness as "beams with compound sections" and to establish simplified methods of computation for these combinations—using some fundamental results of the elastic theory of plates and shells and completing them by experimental research for the cases which are too complex for the analysis. The structural skeleton of an airplane will be built up in this case chiefly by beams with thin and wide flanges or thin webs, and the main problem is to find out which portion of the load can be carried by the thin-walled parts of the section.

The simplest case of a combined section is that of a reinforced sheet, as shown in Fig. 1. Let us consider a bending load. The elementary theory of bending assumes normal stresses proportional to the distance from the neutral axis, the neutral axis being located at the center of gravity of the combined section. According to this elementary theory, the stress would be uniform along the whole width of the sheet. However, this can only be assumed, if the width of the sheet is small compared to its length. Two effects diminish the stresses in the outer parts of the wide flange; the first is due to the distortion of the sheet in its own plane and occurs whether the sheet is used in tension or in compression, and the second effect, due to the buckling of the sheet, occurs only in compression.

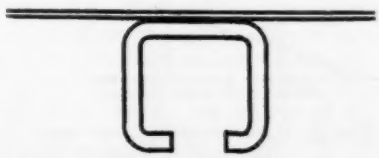


FIG. 1 TYPICAL COMBINED SECTION

In order to calculate the stress distribution in the first case (sheet in tension or in compression below the stability limit), we imagine the flange to be cut off along the section *a-b-c* in Fig. 2*a* before the beam is bent. The bending of the beam obviously changes the length of the line elements along *a-b*, so that in order to reattach the sheet, the latter must be stretched or compressed by shear forces applied on the cut section. We have to consider the problem of two-dimensional stress distribution in a strip subjected to shearing forces applied along one edge. The author has previously studied this problem by the use of Airy's method.[2] If the shearing forces or the bending moment can be developed in a Fourier series, Airy's method immediately gives the stress distribution along the width of the flange. The stress distribution is shown schematically in Fig. 2*a*, both in the main part of the beam and in the flanges. The stresses in the beam are composed of the linearly distributed bending stresses and uniformly distributed compression or tension stresses.

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[1] Director, Guggenheim Aeronautical Laboratory and Graduate School of Aeronautics, California Institute of Technology. Mem. A.S.M.E. Dr.-Ing. von Kármán received his M.E. at Budapest in 1902 and Ph.D. at Göttingen in 1908; honorary degree of Doctor of Engineering, University of Berlin, 1929. He was Privat-Docent, Göttingen, 1910–1913; Professor of Mechanics and Aerodynamics, Director of the Aerodynamical Institute, University of Aachen, 1913; member of Gesellschaft der Wissenschaften zu Göttingen, 1925; foreign member of the Royal Academy of Sciences, Turin, 1928.

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[2] Th. von Kármán, in Föppe, "Beiträge zur technischen Mechanik und technischen Physik, Springer, 1924. W. Metzer and A. B. Miller, *Luftfahrtforschung*, vol. 4, no. 1.

The resultant of the latter stresses is equal and opposite to the resultant of the stresses in the flanges. If the stress distribution in the flanges is known, an effective width can be calculated—i.e., the width of the sheet which, uniformly stressed by the maximum stress occurring in the adjacent part of the beam, gives the same resultant as the effective stresses.

We obtain the maximum of the effective width if we assume the sheet indefinitely extended. In this case the effective width depends chiefly on the rate of variation in the bending moment—i.e., on the span or on the distance between the zero points of the bending moment or on the length in the case of a column in compression. For a continuous beam under uniform load, we obtain the effective width w to be equal to:

$$w = 0.11s \quad \ldots\ldots [1]$$

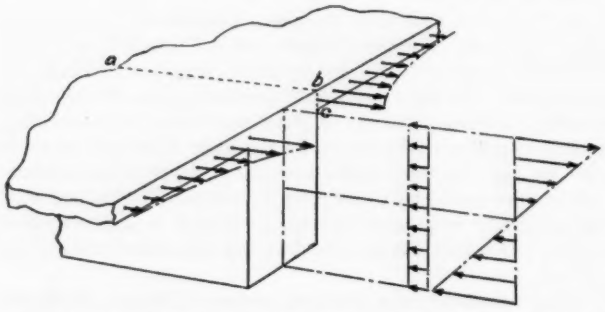


FIG. 2a STRESS DISTRIBUTION IN T-BEAM WITH WIDE FLANGES

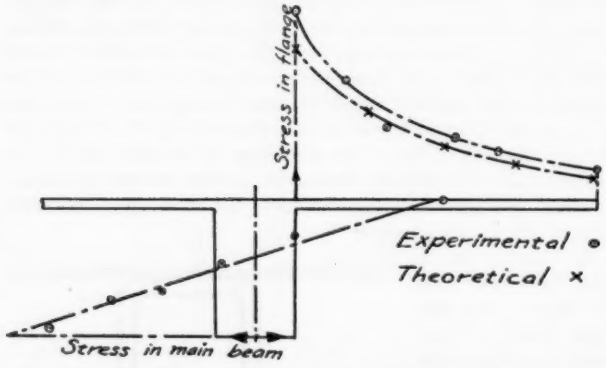


FIG. 2b COMPARISON OF COMPUTED AND MEASURED STRESS DISTRIBUTION IN T-BEAMS WITH WIDE FLANGES

where s = span, and for a beam on two supports with free ends:

$$w = 0.165s \quad \ldots\ldots [2]$$

In the general case the effective width is expressed by the Fourier coefficients of the bending moment considered as a function of the length coordinate of the beam. Fig. 2b shows the comparison between computed and measured stresses in the wide and thin flange of a T-beam.

If the sheet is used on the compression side, as is customary in aeronautical structures, the possibility of buckling must be considered. The stability limit can be calculated by equations deduced by G. T. Bryan, Th. Reissner, and S. Timoshenko. However, in most cases the sheet is used beyond the stability limit, the latter being very low in the case of thin sheets. The mechanism of stress distribution and the question of ultimate strength of thin sheets are discussed in the second part of this paper (AER-55-19b), and also the method suggested by the author to calculate approximately the ultimate strength is indicated.

Using this method, we obtain the following formula as an estimate for the effective width of the buckled sheet:

$$w = \frac{\pi}{\sqrt{12(1-\mu^2)}}\sqrt{\frac{E}{\sigma}}\,t \quad \ldots\ldots [3]$$

where t denotes the thickness of the sheet, E Young's modulus, μ Poisson's ratio, and σ the value $P/\delta w$, if P is the force carried by the sheet. It appears that the effective width depends on the load and decreases with increasing value of σ, until the latter reaches the yielding point σ_y. Putting $\sigma = \sigma_y$, the value of the effective width will be

$$w = \frac{\pi}{\sqrt{12(1-\mu^2)}}\sqrt{\frac{E}{\sigma_y}}\,t \quad \ldots\ldots [4]$$

Substituting for dural $E \cong 10.5 \times 10^6$ lb per sq in., $\sigma_y \cong 40{,}000$ lb per sq in., we obtain $w \cong 16t$. For steel we find $w \cong 22t$.

Shipbuilders are facing a similar problem, in the computation of the contribution of the hull to the effective section of bulkheads. They use practical rules suggesting an effective width equal to 20–25t.

In the cases just considered, it was assumed that the beam and the sheet were straight. Considering a cylindrical sheet with a circumferential stiffness, such as occurs in the case of a cylindrical monocoque fuselage reinforced by bulkheads and loaded in the plane of one of the bulkheads, a third effect diminishing the effective width has to be mentioned. It is obvious that the stress in the curved sheet produces a displacement in the radial direction (see Figs. 3a and 3b), and this displacement diminishes the circumferential strain and the circumferential stress, both in the case of tension (Fig. 3a) and in that of compression (Fig. 3b). The deformation of the sheet and the stress distribution resulting from this deformation can be calculated,[3] and we obtain for the maximum value of the effective width:

$$w = 0.54\sqrt{tR} \quad \ldots\ldots [5]$$

where R denotes the radius of curvature of the sheet.

A special problem related to that of the reinforced sheets is the stability and ultimate strength of sheets with isolated corrugations. The ordinary corrugated sheet can be considered as a homogeneous plate with increased flexural stiffness in the sections perpendicular to the corrugations. However, isolated corrugations are often used in order to increase the stiffness of a sheet; in these cases, the question of the average stiffness of the whole



FIG. 3a COMBINED SECTION IN BENDING, FLANGE IN TENSION



FIG. 3b COMBINED SECTION IN BENDING, FLANGE IN COMPRESSION

[3] An analogous problem occurs in the calculation of the flexural rigidity of curved pipes. Dr. Th. von Kármán, *Z.V.D.I.*, 1911, vol. 55. The present formula (5) has not been previously published. W. Hovgaard, *Journal of Math. and Physics of the M.I.T.*, also Proc. of the Third International Congress for Applied Mechanics, vol. 2, p. 331.

sheet is raised by the designer. This problem is obviously similar to that of the sheet reinforced by beams.

The curvature effect mentioned occurs also in the bending of straight reinforced sheets, but it is a second-order effect, because in this case the curvature $1/R$ is itself proportional to the bending moment M divided by the flexural rigidity yE, and so the variation on the stress distribution due to the curvature is of the order (M/yE).[2] Consequently, for slightly bent beams or in the case of buckling of columns this effect can be neglected.

The foregoing considerations show that the elastic behavior of a reinforced sheet is a rather complex problem, and systematic experimental research work is needed in order to determine the limits within which the theoretically deduced formulas can be applied and to determine the resultant influence of the different effects outlined.

Another use to which the flat sheet is often put is to serve as web of a deep beam (Fig. 4) on the side wall of a box. In this case also we have to distinguish between structures in which the stability limit is considered as an upper limit for the stresses and those in which the sheet is used as a "tension field" in the buckled state. In structures of the first kind the sheet usually has a considerable thickness and contains cutouts to reduce weight (Fig. 5). The strip with holes affords an interesting example of two-dimensional stress distribution. Using Airy's method, we are able to calculate the maximum stresses at the circumference of the holes and the influence of the holes on the rigidity of the whole construction. The infinite plate with a hole under uniform tension and shear has been calculated by many authors. The chief result is that the maximum stress at the circumference of the hole is twice the uniform stress applied to the sheet. This rule can be applied to strips of finite width, if the diameter of the hole is small in comparison to the width. At the author's suggestion, M. Hirota[4] recently calculated the stress distribution in the case that the diameter of the holes is from the same order as the width of the sheet. For the ratio 1:2 between diameter and width, he finds that the maximum stress is about eight times as high as

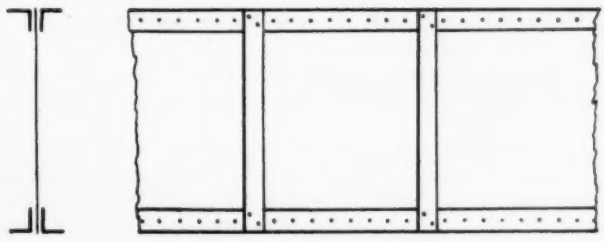


FIG. 4 TYPICAL THIN WEBBED BEAM

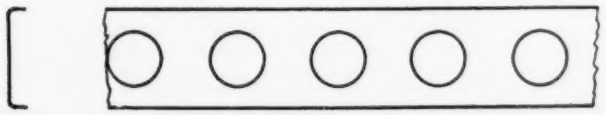


FIG. 5 WEB WITH HOLES

that which would occur without holes. It should also be mentioned that a beam whose web contains holes has considerably smaller flexural rigidity than would correspond to the average value of the inertia moment.

If the web or wall is enclosed between two rigid longitudinal elements and loaded by shearing forces, the stability limit is given by a simple formula, first given by Southwell and Skan; the influence of the "bracings" which generally are used in order to stiffen the plate both in the longitudinal and in the transversal direction has been worked out by H. Wagner and C. Schmieden.

[4] Thesis at the University of Aachen.

If the wall is used beyond the stability limit, we obtain "the tension-field type" of beam, i.e., the sheet can be replaced by a system of tension elements or strings, as shown in the second part of the paper. The most important progress in this field, due to Dr. Wagner's suggestions, is the fact that it is possible to develop structural elements lighter than trussed structures whose stress distribution and elastic behavior can be computed with nearly the same accuracy as is the case with simple trusses. Dr. J. Mathar, assistant professor at the Aachen University (Germany), made

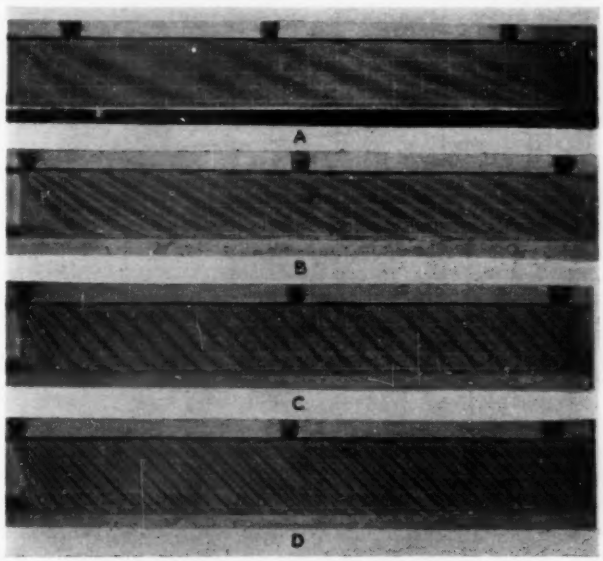


FIG. 6 ACTION OF WEB UNDER SHEAR
(A, p = 0.0025 kg per cm; B, p = 0.04 kg per cm; C, p = 0.01 kg per cm; D, p = 0.08 kg per cm; p denotes the shearing force per unit length.)

some experimental investigations on the elastic behavior of such high beams with tension field webs.[5] He made first some observations on a paper model, which show the development of the tension field, especially the rate of increase of the waves as a function of the shearing stress. Some of his photographic records are reproduced in Fig. 6. Then he constructed a model of a high beam, using dural sheet of 0.2 mm thickness as web between two longitudinal duralumin flanges of about 6 ft length. The sheet was stiffened by transverse braces; the number of the latter elements was varied from one to seven. If the beam is loaded by a concentrated force Q applied at the free end, the stresses in the longitudinal elements are found in first approximation to be equal to

$$\sigma = \frac{Qx}{h} \pm \frac{Q}{2} \cot \varphi \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

In this equation x represents the distance of the section considered from the load, h the height of the beam, and φ the inclination between the longitudinal elements and the direction of the tension elements or waves developed by the buckling of the sheet (approximately 40 deg). The tension stress in the sheet is:

$$\sigma = \frac{2Q}{th \sin 2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

in which t = thickness of the sheet.

The compression load in the upright elements (transversal bracings) is, according to this elementary theory:

[5] Jahrbuch der wissenschaftlichen Gesellschaft für Luftfahrt (W.G.L.), 1929, p. 205.

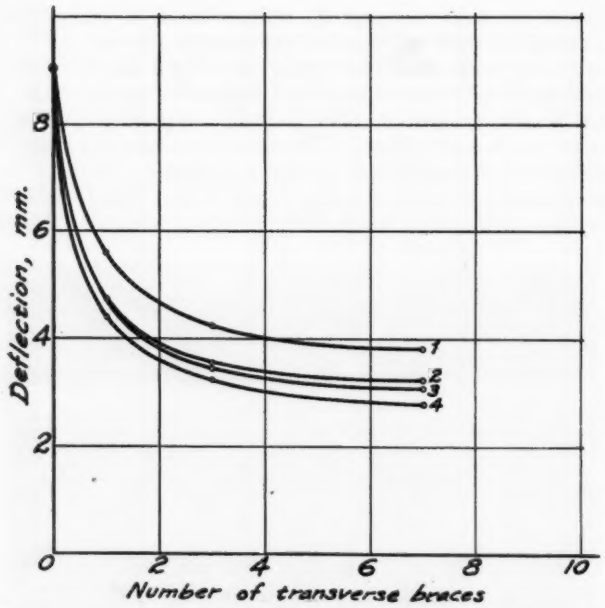


FIG. 7 INFLUENCE OF NUMBER OF TRANSVERSE BRACES ON THE STIFFNESS OF WEB IN SHEAR

$$P = \frac{Q}{Ch} \tan \varphi \quad \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

where C is the distance between the transverse braces. The stresses in all these elements have been measured by extensometer. Table 1 gives the comparison between computed and measured stresses in one of the panels (at a distance of about one-third of the length from the supported end of the beam). The load is equal to 60 kg.

Fig. 7 gives an interesting comparison between the values of the deflection at the free end of the beam as function of the number of the transverse braces, corresponding to four different types of connection between the flanges, the transverse braces, and the sheet:

Curve 1. Transverse braces pin-jointed to the flanges. No connection between sheet and transverse braces.

Curve 2. Rigid connection between flanges and transverse braces; sheet as in case 1.

Curve 3. Flanges and transverse braces, as in case 1; sheet fastened to the transverse braces.

Curve 4. Longitudinal and transverse braces, as in case 2; sheet as in case 3.

TABLE 1

Stress	Measured, kg per sq cm	Computed, kg per sq cm
In the upper flange	136	120
In the lower flange	147	156
Maximum tension in the sheet	172	180
In the transverse brace	18	22.5

J. Mathar also investigated the elastic behavior of a fuselage of rectangular section under different static and dynamic load conditions. An actual fuselage of the Dornier type was used for these experiments.[6] Mathar found that the effective rigidity of the fuselage in bending is fairly well approximated if the following elements are assumed to carry stresses: the walls and stiffness in tension, the stiffness alone in compression, and half of the side-walls.

This short survey of some typical cases shows that, in spite of some gaps in our present knowledge, the calculation of structural elements composed of beams and flat sheets can be attacked with some hope of obtaining usable information. The situation is worse as far as curved sheets and especially curved sheets in connection with stiffeners are concerned. A method similar to that sketched for calculation of the effective width of a flat sheet can probably be applied to curved sheets.

At the present time experimental investigations are being carried out at several places on the stability and ultimate failure of cylinders in compression, bending, and torsion and on the ultimate strength of reinforced curved sheets under different load conditions. It may be hoped that in the following few years the problem will be solved in sufficient generality to give a safe foundation for the design of monocoque structures.

[6] Jahrbuch der W.G.L., 1928, p. 105.

AER-55-20

Possibilities of Cabin-Type Autogiros

By A. E. LARSEN,[1] PHILADELPHIA, PA.

The development of the Autogiro has virtually been made on the open-cockpit type, one to three place, weighing from 1000 to 3000 lb, and with an engine of 80 to 400 hp. The question has often arisen if it could be adapted to passenger transport. The author asserts that the cabin-type Autogiro is entirely possible. Three cabin Autogiros have been built, markedly different as to type, size, power, and general construction. The trial flights in this country suffered because of lack of mechanical starting units, not then available. Detail studies have been made of various sizes of cabin-type Autogiros, and the author gives an idea of the possibilities in this development.



THE progress of Autogiro development has virtually been made on the open-cockpit type. These machines have been one- to three-place and the weight from 1000 to 3000 lb with power plants from 80 to 400 hp. Questions often arise as to the possibilities of the cabin Autogiro and if it could be developed for transport work. This paper will definitely assert that the cabin-type Autogiro holds tremendously interesting possibilities, with promise of increased performance and still greater utility than has been displayed by the present Autogiro. Mention must be made of those cabin models of Autogiros that have been built:

1 The LePere cabin Autogiro, model C-18, made in France in 1929 by the Weyman LePere Company for the Cierva Autogiro Company, Ltd., of England, as an intended entrant in the Daniel Guggenheim Safe Aircraft Competition.

2 The Kellett convertible-cabin Autogiro K-3, now available in this country.

3 The Cierva Autogiro C-24, which received widespread publicity last Summer.

These three machines are markedly different as to type, size, power, and general construction.

The LePere combined the novel Autogiro features and metal construction. It was a four-place cabin type, with a geared seven-cylinder Wright J-6 engine; gross weight, 3060 lb; useful load, 1100 lb. The trial flights in this country were marked by experimental difficulties owing chiefly to lack of mechanical starting units not then available. Starting was effected by the deflector-tail principle, and though several rotors were fitted and numerous flights completed, the possibilities were not fully developed.

The Kellett Autogiro Company, of Philadelphia, devised a convertible cabin, capable of rapid fitting and removal on their side-by-side K-3. This arrangement follows the lines of the rather successful enclosures on a number of British sport planes and makes an admirable conversion without serious impairment of vision. The main characteristics: Gross weight, 2300 lb; useful load, 374 lb; rotor diameter, 40½ ft; 210-hp Kinner C-5 engine; maximum speed, 110 mph; cruising speed, 95 mph; initial rate of climb, 1000 fpm. It is doubtless a forerunner of a type that will find wide application among private owners.

England has given considerable thought to various sizes and types of cabin Autogiros, single-engine and multi-engine, pusher and tractor, and from numerous design projects the Cierva C-24 Autogiro was constructed along lines of maximum aerodynamic cleanliness. This little two-place machine, with a four-cylinder inverted aircooled D.H. Gypsy Mark III engine, developing 115 hp, attained 118 mph in trials. Its all-round performance compared favorably with the D.H. Puss Moth, considered a good example of modern English airplane design. This comparison, for practically equivalent weight and power, is as follows: Top speed, C-24, 118 mph; Puss Moth, 126 mph. Rate of climb, C-24, 900 fpm; Puss Moth, 900 fpm. The possibilities of cabin-type Autogiros are indicated most favorably. Following are comparisons of parasite drags of wind-tunnel models at 100 mph, with the fixed wing drag deducted: C-19 Mark III (the English type shown frequently in this country), biplane tail, 200 lb. C-19 Mark IV (the current production model in England), monoplane tail, 170 lb. C-24 (cabin), 100 lb.

Numerous studies have been made of the possibility of cabin-type Autogiros in this country, owing to the potential buying market developed by demonstrations in which particularly the PCA-2 three-place model was flown on numerous estates along the Eastern Central seaboard. Literally hundreds of estate owners will not lay out landing fields, but all would consider a fairly uniform type of Autogiro, of 300 to 400 hp, four- or five-place cabin, with provisions for luggage, and capable of two or three hour flights. The consistency of these specifications has resulted in the undertaking of such a project by one of the Autogiro licensees, Pitcairn Aircraft, Inc.

Other detail studies have been made, the computed specifications and performance of which follow. The first study, of a machine with gross weight of 7500 lb, had these specifications: Rotor diameter, 70 ft; chord, 24.6 in.; parasite drag, 425 lb at 100 mph; engine hp, 575 at 1950 rpm; propeller efficiency, 0.80; thrust hp, 460. The computed performances follow:

	Light	Loaded
Weight, lb	5,000	7,500
High speed, mph	137	128
Rate of climb, fpm	1,337	673
Ceiling, ft	23,700	13,300
Minimum speed, mph	24.0	35.4
Steepest angle of climb, deg	15.2	7.17
Speed of vertical descent, fps	16.6	19.8

A 15,000-lb three-engine Autogiro, with these specifications, was studied in comparison with a well-known 16-passenger tri-

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

motored transport plane: Rotor diameter, 100 ft; rotor chord, 47.6 in.; parasite drag, 650 lb at 100 mph; engine hp, 1300; propeller efficiency, 0.80; thrust hp, 1040. The computed performances follow:

	Light	Loaded
Weight, lb	10,000	15,000
High speed, mph	160	150
Rate of climb, fpm	1,510	820
Ceiling, ft	24,000	15,400
Minimum speed, mph	25.0	32.0
Steepest angle of climb, deg	16.0	7.25
Speed of vertical descent, fps	16.4	20.0

The third study was made on a 6500-lb machine, designed for nine passengers, for the sake of comparison with a well-known single-engine transport plane. The specifications are: Rotor diameter, 64 ft; rotor chord, 29 in.; parasite drag, 364.5 lb at 100 mph; engine hp, 575 to 1950 rpm; propeller efficiency, 78 at high speed loaded and 74 light; thrust hp, 448 at high speed loaded and 424 light. The computed performances follow:

	Light	Loaded
Weight, lb	4,000	6,500
High speed, mph	148	135.5
Rate of climb, fpm	1,600	820
Ceiling, ft	26,100	16,200
Minimum speed, mph	25.0	29.0
Steepest angle of climb, deg	18.0	8.5
Speed of vertical descent, fps	15.8	19.4

Following these studies, Autogiro cabin applications were considered in conjunction with a number of interesting types of planes, as for example the Stout Skycar, provided with 25 per cent increase in power.

The following study was undertaken to obtain an idea of the performance that might be expected from a twin-engined Autogiro of pilot-and-four-passenger capacity. With this data, the effect on performance of variation in parasite drag, the addition of controllable-pitch propellers whose low-pitch position would be at a design V/ND corresponding to the speed of maximum rate of climb, and of "boosted" or supercharged engines, both with and without controllable-pitch propellers, were investigated. It is believed the variation in performance due to the influence of these various items may be used to draw general conclusions as to their effect on any Autogiro.

The propellers referred to as "standard" are designed by the nomographic chart for propellers in the Air Corps "Handbook for Airplane Designers," 1925 edition. The design speeds in determining these diameters by this chart are taken as 10 per cent less than the actual "expected" high speeds found by Mr. Cierva as advisable in Autogiro propellers and corrected by a factor of 1.053, as it was found in propeller tests on the PAA-1 Autogiros that the best propeller was 5.3 per cent greater in diameter than one whose diameter was determined as above. However, the design V/ND for which the maximum efficiency was determined was at the high speed, for the large diameter, so all estimated propeller efficiencies are thereby conservative in this regard.

For the unsupercharged engines, the brake-horsepower corresponding to different engine rpm was obtained from power curves submitted by the engine manufacturer. The supercharged or "boosted" engine was considered as giving 50 per cent more than normal horsepower at the same rpm. Simply as a supercharged engine, this would correspond to one with a 10,000-ft rating. As it has been demonstrated as practical to run engines with both high compression and highly geared induction systems at full throttle for a minute to a minute and a half at sea level, it is considered reasonable to compute the performance with the power of the engine boosted 50 per cent as being a performance of which the machine is capable for take-off over obstructions or for short bursts of speed.

TABLE 1

Machine*	Parasite drag, lb at 100 mph	Parasite drag in slipstream, %	Prop. diam, ft.	Design V/ND	Prop. efficiency at design V/ND, %	Gross weight, fully loaded, lb	High speed at sea level, mph	Rate of climb at sea level, ft per min	Max. path angle of climb, sea level, deg	Absolute ceiling, ft	Service ceiling, ft	Performance at 5000 ft: High speed, mph	Performance at 5000 ft: Rate of climb, ft per min	Performance at 5000 ft: Max. path angle of climb, deg	Performance at 10,000 ft: High speed, mph	Performance at 10,000 ft: Rate of climb, ft per min	Performance at 10,000 ft: Max. path angle of climb, deg	Performance at 15,000 ft: High speed, mph	Performance at 15,000 ft: Rate of climb, ft per min	Performance at 15,000 ft: Max. path angle of climb, deg
2, 150 lp	201.964	40.9	7.5	0.782	78.0	3600	133.2	675	5.5	12,200	10,440	124.5	400	3.25	107.8	120	1.0	...	...	...
2, 150 hp	389.536	59.9	7.5	0.782	78.0	3600	101.8	350	3.2	7,400	5,300	91.5	112	1.0	...	...	...	...	...	...
2, 210 lp	203.964	41.5	7.9	0.828	79.0	4250	152.0	862	6.75	14,300	12,700	141.0	564	4.4	125	260	2.1	...	...	...
2, 210 hp	403.036	60.3	7.9	0.828	79.0	4250	114.5	495	4.5	9,300	7,450	106.0	230	2.1	...	...	...	...	...	...
8, 210 lp, 2d. prop., eng. sup.	203.964	41.5	7.9	0.828	79.0	4400	...	1515	12.1	18,500	17,825	...	1210	9.3	...	860	6.3	...	425	2.8
2, 210 lp, cont. prop., eng. std.	203.964	41.5	7.9	0.343	60.5	4310	...	1250	13.95	15,400	14,175	...	845	9.45	...	440	4.8	...	33	0.35
2, 210 lp, std. prop., and eng.	203.964	43.6	8.7	0.804	78.4	4270	142.0	705	5.75	12,800	11,000	132.0	425	3.55	124	155	1.2	...	...	...
2, 210 lp, std. prop., eng. sup.	203.964	43.6	8.7	0.804	78.4	4420	166.0	1570	11.55	20,000	18,700	170.0	1180	8.25	162.5	780	5.25	148.0	385	2.6
2, 210 lp, cont. prop., eng. sup.	203.964	43.6	8.7	0.363	62.4	4500	...	2240	21.5	21,150	19,900	...	1555	15.2	...	985	9.85	...	515	5.15

* Abbreviations: lp, low parasite; hp, high parasite; std., standard; cont., controllable; prop., propeller; eng., engine; sup., supercharged.

The power-available curves for different combinations, at different altitudes, were determined in the usual manner.

The performances of these machines were considered for the more efficient three-bladed rotor, based upon performance reports of the latest C-19 Mark III-A received from the Cierva Autogiro Company, Ltd. The parasite drag of this machine is 165 lb at 100 mph, and with the curve of power available furnished in this report, the rotor fixed-wing (L/D) ratio was computed from the climb and glide tests.

The climb tests give very optimistic results, showing a maximum fixed-wing rotor L/D of over 16. The glide data give more conservative results, a projection of the general curve showing a fixed-wing rotor (L/D) maximum of 13.6, which is considered reasonable and possibly on the conservative side. This latter set of data, in comparison with the characteristics of a four-blade rotor fixed-wing combination, is used in determining the rotor fixed-wing drag that is used in determining the "power required."

The parasite drags, corrected for slipstream velocity, are obtained for each modification of machine considered. The weight of each modification, fully loaded, the probable value of Ω/V at different speeds, the corresponding rotor fixed wing (L/D), and the rotor fixed-wing drag for each speed are found. The rotor wing drag and parasite drag are added, and the horsepower required determined from these total drags.

From these power curves the performance data are obtained, covering rate of climb, ceiling, speeds at altitude, engine rpm at altitude, and maximum path angle of climb. All performance figures are for the machine fully loaded, with pilot and four passengers, with four hours' gas, the specified amount of oil and 150 lb baggage. No attempt was made to estimate the rotor rpm exactly, the figures used in determining Ω/V corresponding to different speeds being obtained from analysis.

The main characteristics of each of the combinations, with their performances, are given in Table 1.

The conclusions from these performance estimates are as follows:

If the twin-engine machine can be made as clean aerodynamically as the "low parasite," it will have a very creditable performance. To obtain this, it is recommended that extensive wind-tunnel tests be carried out on the fixed-wing parasite parts assembly, that modifications be made during these tests, and that the final machine constructed be the best compromise of aerodynamic cleanness consistent with practical construction.

In the general case, aerodynamic cleanness plays a more important part in influencing performance than either weight or power, especially with the more efficient three-blade rotor. Consequently, all future designs of Autogiros, besides being made as light as possible with ample reserve power, should have all parasite drag reduced to a minimum. In this regard, anti-drag engine cowling, fillets at all intersecting surfaces, streamlining of all parasite parts, and elimination of gaps between surfaces should be considered in all design.

The addition of a controllable-pitch propeller to any design improves the take-off performance an average of 44 per cent in initial rate of climb, 96 per cent in maximum path angle of climb at sea level, and 6.8 per cent in absolute ceiling.

The addition of a 10,000-ft supercharger (which can be used full-on for short periods below 10,000 ft) to a machine with a sea-level propeller increases the initial rate of climb 76 per cent, the maximum path angle of climb at sea level by 79 per cent, and the absolute ceiling by 29.5 per cent.

The addition of a 10,000-ft supercharger, with a 10,000-ft propeller, increases the performance for an unsupercharged machine with standard sea-level propeller 82 per cent for initial rate, 71 per cent for maximum path angle of climb at sea level, 150 per cent for maximum path angle of climb at 10,000 ft, 30 per cent for high-speed at 10,000 ft, and 40 per cent for absolute ceiling.

Conversely, a 10,000-ft propeller on a machine, either unsupercharged or one with a supercharger not functioning, decreases the performance 22 per cent in initial rate, 7 per cent in high speed, 17 per cent in maximum path angle of climb at sea level, and 12 per cent in absolute ceiling.

It is believed the most satisfactory method of increasing engine performance would be a high-compression engine with a highly geared induction system, similar to that on the PCA-2 No. 5, which could be operated at full throttle for short periods below 5000 ft or 8000 ft, rather than an independent supercharger which would require one additional control and would take up more space in the engine compartment than the recommended arrangement.

The ideal combination would be a machine of low parasite drag, with high-compression engine and highly geared induction system, fitted with a controllable-pitch propeller.

The next phase of this design study investigated was that of the ability to overcome yawing moments at all speeds by the use of rudder alone, with one engine operating at full throttle. This investigation showed that it would be advisable to tilt the engines approximately 3 deg each, with the thrust lines converging aft in order to reduce the yawing moments due to the engines and to enable the slipstream to strike the vertical surfaces at a greater incidence. It was further shown that, in order to completely overcome the yawing moment due to one engine, it would be necessary to use a slotted-airfoil arrangement for the vertical surfaces, operating in conjunction with the rudder as a flap.

AER-55-21

Principle of Similitude as Applied to Research on Thin-Sheet Structures

By JOHN E. YOUNGER,[1] BERKELEY, CALIF.

Any program of research on airplane structures must give consideration to the design and stress-analysis requirements of the Army, the Navy, and the Aeronautics Branch of the Department of Commerce. The requirement that the breaking strength of the structure or of its component parts must be the basis of strength calculations places a difficult problem before research men. Hooke's law of proportionality of stress to strain forms the basis of the mathematical theory of elasticity, and above the proportional limit of the material (which for aircraft metals may be taken as the elastic limit), formulas based on this law do not apply. Formulas are derived from the standpoint of the mathematical theory of elasticity which fit the data quite accurately for stresses within the elastic limit. For stresses above the elastic limit an empirical formula will be necessary. In so far as the practical use of the data is concerned, since stresses above the proportional limit are involved, a derived formula may not be as desirable as the experimental curve. The value of the experimental curve depends very greatly upon the proper selection of the coordinates and the parameters.



IN ANY program of research on airplane structures, consideration must be given to the design and stress-analysis requirements of the Army, the Navy, and the Aeronautics Branch of the Department of Commerce. The requirement that the breaking strength of the structure or of its component parts must be the basis of strength calculations places a difficult problem before research men. Of course, Hooke's law of proportionality of stress to strain forms the basis of the mathematical theory of elasticity, and above the proportional limit of the material (which for aircraft metals may be taken as the elastic limit), formulas based upon this law do not apply.

The author wishes to avoid discussion of the desirability of discarding the ultimate-strength requirement as the basis of design for that of a lower stress requirement. Both sides of the question have excellent arguments in their favor. The fact is that the ultimate-strength requirement is the "law," and right or wrong, research men must take this into consideration if their results are to be of practical use.

Most theoretical formulas pertaining to the strength of a structure or the division of loads between members of the structure, as for example in the theory of least work, are based upon Hooke's law. Since we must determine, however, the breaking strength of the airplane or parts of the airplane, it is necessary to assume an extension of this law above the elastic limit or else use other means of developing our design criteria and formulas.

There are two principal ways in which design formulas for stresses above the elastic limit may be developed. The first is to assume the breaking strength to be below the proportional limit of the material, derive theoretical formulas by the application of the mathematical theory of elasticity, then assume the validity of these formulas above the elastic limit, or to assume that the basic terms in these theoretical formulas may be used in connection with experimental coefficients determined from model experiments. We incidentally note that very few engineering formulas are the direct result of theoretical calculations. They are nearly all tempered by experimental coefficients which have been very aptly called "coefficents of ignorance." An outstanding example of the application of a theoretical formula by means of experimental coefficients is the application of the so-called "precise three-moment equation," with the use of the chart of Fig. 13 in the Aeronautics Department of Commerce Handbook, Bulletin 7-A.

The second method of predicting the strength of a structure for stresses above the elastic limit is by the application of model testing through the principle of similitude. Coefficients from experimental results are obtained from model tests and applied to the full-scale condition through the use of empirically derived formulas. Model testing, in general, has formed the basis for strength calculations of airplanes, inasmuch as any experiment on the strength of a structure is a model test, whether the results be applied to the full-scale structure through the use of coefficients or through the use of empirically derived formulas.

In the application of the theory of dimensions in connection with the principle of similitude, it is necessary that we use good judgment in determining what physical conditions connected with our problem determine the strength of the structure. In other words, we cannot approach the problem blindly. We must assume the dimensions or other factors which may affect the result which we hope to obtain. Now, if we assume these incorrectly, then our result will obviously be incorrect. If we assume too few conditions, our results obviously may not be complete.

As a very simple case to introduce the subject, let us assume that we have been incapable of deriving the simple modulus-of-rupture formula,

$$f = \frac{Mc}{I} \qquad [1]$$

in which M is the bending moment, c is the distance from the neutral axis to the outermost fiber, and I is the moment of inertia of the cross-sectional area of the beam. As stated previously, let us assume that we know nothing of this equation, but that it is necessary for us, from an engineering standpoint, to know the

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

stress which will be introduced in a certain large beam before the beam is constructed. In order that we may apply our theory more readily, let us assume this beam to be rectangular in cross-section and of very large dimensions. For example, consider the length to be 1000 in., the depth 20 in., and the breadth 10 in. We are to apply a load of 1000 lb at the outer end. We must know the maximum fiber stress in the beam.

Under the assumption that we have no knowledge of the theory, we shall attempt to find this fiber stress by an experiment on a model of this beam. Let the scale of the model be one-tenth of the full-scale prototype. Let the scale = 10 = N; therefore we have the model 100 in. long, 2 in. deep, and 1 in. wide. We apply a load W to the model.

Now let us observe again the quantities we wish to find. We use the experiment on this model to find the stress from which we can determine the stress on the full-scale beam; hence we go from model to full scale. We first determine in so far as possible the dimensions and quantities which may influence the stress. Let us assume, then, that the fiber stress is a function of these dimensions or quantities. We of course know that the stress will be influenced by the moment of inertia, but we recall that the moment of inertia is a product of the derivation of the modulus-of-rupture formula; hence if we know nothing about that theory, as assumed, we would not be inclined to assume that the moment of inertia entered into the problem, and therefore we shall omit it in our first analysis. We note here that we assume that there is only one dimension, l, but if the model is geometrically similar to the full-scale prototype, then the one dimension expresses this relationship thoroughly. Let us first solve the problem on the basis of this assumption of geometrical similarity. We assume that the fiber stress depends upon the size of the beam and the weight W. Thus we write f as a function of l and W as

$$f = F(l, W) \quad [2]$$

We assume the form of the function to be

$$f = Kl^a W^b \quad [3]$$

which, it will be noted, is a function of a general form, since the value of a or b may be any quantity whatsoever. Writing this equation in dimensional form, we have

$$\frac{ML}{L^2T^2} = L^a \frac{M^b L^b}{T^{2b}} \quad [4]$$

Now equating the exponents of M, we have

$$1 = b \quad [5]$$

Equating the exponents of T, we find that $-2 = -2b$, $b = 1$, as previously noted. It is therefore possible to find only two unknowns, since two of our equations are identical. Equating the exponents of L, we have

$$-1 = a + b \quad [6]$$

or since $b = 1$, we have $a = -2$. Therefore, we have upon substitution in Equation [3]

$$f = Kl^{-2}W = K\frac{W}{l^2} \quad [7]$$

Therefore, we find that the fiber stress in each of the two beams is proportional to the ratio of applied load W and inversely proportional to the square of any dimension. We find therefore that (letting subscript f refer to full scale and m refer to the model)

$$\frac{f_m}{f_f} = \frac{K\left(\frac{W}{l^2}\right)_m}{K\left(\frac{W}{l^2}\right)_f} = \frac{W_m}{W_f}\left(\frac{l_f}{l_m}\right)^2 \quad [8]$$

Now we note that the ratio of the dimension f of the full-scale beam to the dimension of the model beam is the scale, which we designated as N. Therefore, we have, since $N = 10$

$$f_f = f_m \frac{W_f}{W_m}\left(\frac{1}{N}\right)^2 \quad [9]$$

It is not the formula here which is important, but the method of approaching the formula which is of primary value. Now let us apply this to the example. Let us assume that the weight applied to the model beam is 100 lb. Therefore, we have

$$f_f = f_m \frac{1000}{100}\left(\frac{1}{10}\right)^2$$

$$f_f = 0.1\, f_m \quad [10]$$

We note that f_m, the fiber stress in the model, is determined by experimental apparatus such as a strain gage. Having known this value, we can then determine what the fiber stress will be in the full-scale beam. Now let us assume that we have performed an experiment on the model and have obtained the desired result. We will, however, now revert back to our theoretically derived formula to determine f_m, and then let us determine f_f from the same formula, and see whether or not the relationship of Equation [10] actually holds. We find that

$$f_f = \frac{M_c}{I} = \frac{1000 \times 1000 \times 10}{\frac{10 \times (20)^3}{12}} = 1500 \text{ lb per sq in.}$$

Now from Equation [10] we find

$$f_f = 0.1\,\frac{100 \times 100 \times 1}{\frac{1 \times 2^3}{12}} = 1500 \text{ lb. per sq in.} \quad [11]$$

which checks the calculated result. Now it is to be noticed in this respect that whereas we use the simple cantilever beam for the derivation of our principle of similitude, the derived formula, Equation [10], will apply to more complicated types of beams. Yet for a model experiment in so far as the principle of similitude is concerned, it would be just as simple to determine the stress in this type of beam as it would be to determine the stress in a cantilever beam.

From Equation [7] we note, since the bending moment in the beam is $W \times l$, that

$$f = K\frac{M}{l^3} = K\frac{Ml}{l^4} \quad [12]$$

Since the moment of inertia may be expressed as a proportion of l^4 and c as a proportion of l, we may write Equation [12]

$$f = K'\frac{Mc}{I} \quad [13]$$

which is the modulus of rupture formula as we use it. Of course, we would not expect to derive a modulus of rupture formula from this standpoint. We are merely pointing out here that our derived formula gives a modulus of rupture in a form which is readily applicable.

As another example, let us consider the prediction of the

strength of a strut. The application of the mathematical theory of elasticity gives us, of course, Euler's formula,

$$\frac{P}{A} = \frac{C\pi^2 E}{\left(\frac{l}{\rho}\right)^2} \quad [14]$$

which states that the unit stress is proportional to $E\ (l/\rho)^{-2}$; that is,

$$\frac{P}{A} = KE\left(\frac{l}{\rho}\right)^{-2} \quad [15]$$

This formula of course applies only for stresses less than that of elastic limit, and accordingly has a very limited application. It does, however, indicate the coordinates for a graphical representation of the experimental data from which empirical formulas for stresses above the elastic limit may be determined, as, for example, Johnson's parabolic formula.

In attempting to represent experimental data graphically, for the purpose of determining an empirical formula, it is sometimes difficult to determine the coordinates to be used. In such cases, the principle of homogeneity of fundamental dimensions, length, mass, and time in an equation is of great assistance. It may be noted that most of the stress formulas applicable to the determination of the ultimate strength of aircraft structures may be arrived at by this simple process. For example, in considering the strength of struts, we assume that the strength is a function of the length, the modulus of elasticity, and the disposition of the cross-sectional area, which may be specified by the radius of gyration ρ, thus:

$$\frac{P}{A} = F(E, l, \rho) \quad [16]$$

Let us now assume a general form of this function, thus

$$\frac{P}{A} = KE^a l^b \rho^c \quad [17]$$

Writing the equation in the dimensional form, we have

$$\frac{ML}{L^2T^2} = \frac{M^aL^a}{L^{2a}T^{2a}} L^bL^c \quad [18]$$

Equation the exponents of M, we have $a = 1$. Equating the exponents of T, we find again that $a = 1$. Equating the exponents of L, we find $-1 = -a + b + c$, or $b = -c$. Thus Equation [17] becomes

$$\frac{P}{A} = KEl^b\rho^{-b} = KE\left(\frac{l}{\rho}\right)^b \quad [19]$$

Thus we find that P/A is a function of (l/ρ). This function may be determined from experimental data, in the form of an empirically derived formula.

Now for the purpose of studying the applications of the principle of similitude in determining the deflection of a structure, let us consider the simple example of bending of a cantilever beam. We are to find the deflection at the end of the beam. Let us assume that we have been unable to calculate this deflection from a theoretical standpoint. It becomes necessary, then, that we perform experiments on the model in order that we can determine the deflection of the full-scale structure. We apply the principle of similitude in order that we may determine the relationship between the quantities involved in the model and those same quantities which may affect this deflection. We assume that the deflection is a function of the variables as noted in Equation [20]:

$$y = F(l, E, W) \quad [20]$$

in which l is the length of the beam, E is the modulus of elasticity, and W is the applied load. We note that in Equation [20] we use one term involving the dimension of length; i.e., we do not incorporate the breadth and the thickness of the beam in the function, inasmuch as for geometrically similar structures these would be expressed as a proportion of l. Hence we are adding no new term in the function. We assume, therefore, that the equation may be written in the following form:

$$y = Kl^aE^bW^c \quad [21]$$

Writing the equation in the dimensional form, we have

$$L = L^a \frac{M^b}{L^bT^{2b}} \frac{M^cL^c}{T^{2c}} \quad [22]$$

Equating the exponents of the dimension L, we find that

$$1 = a - b + c \quad [23]$$

Equating the exponents of M, we find that

$$o = b + c \quad [24]$$

Equating the exponents of T, we find that

$$o = -2b - 2c \quad [25]$$

which is the same as Equation [24]. Let us assume that b and c are known quantities, since it is possible to evaluate only one of the unknown quantities a, b, or c. We therefore have

$$a = 1 + b - c = 1 - 2c \quad [26]$$

Thus we find in Equation [21] that

$$y = Kl^{1-x}E^{-c}W^c \quad [27]$$

Rewriting this equation, we have that

$$y = Kl\left(\frac{W}{El^2}\right)^c$$

which becomes

$$y = KlF\left(\frac{W}{El^2}\right) \quad [28]$$

Now writing the equation for the comparison of the deflection of the model to that of the full scale, we have

$$\frac{y_m}{y_f} = \frac{Kl_mF_m\left(\frac{W}{El^2}\right)}{Kl_fF_f\left(\frac{W}{El^2}\right)} \quad [29]$$

We note in Equation [29] that if $F_m = F_f$, then we can make a direct comparison of deflections. We note that the functions F_m and F_f must be equal because they are of an unknown form; that is, the comparison of the model to that of the full scale requires that these two functions be equal. They can be made equal provided

$$\frac{W_m}{E_ml_m^2} = \frac{W_f}{E_fl_f^2} \quad [30]$$

These terms can be made equal by selecting the proper experimental conditions. For example, the modulus of elasticity of

the model may be made the same as the modulus of elasticity of the full-scale prototype. In this case we find that

$$\frac{y_m}{y_f} = \frac{l_m}{l_f} = \frac{l_m}{Nl_m} = \frac{1}{N}$$

In other words, the deflection is proportional to the scale

$$y_f = y_m N \quad [31]$$

We should note that in general this relationship need not apply to the deflection of any structure. It is the application of the principle of similitude that we are attempting to illustrate. The derived formula has no particular value to us. A point which should be brought out is that for a particular problem we may apply the principle of similitude to determine the relationships which are desired.

A question has arisen as to what the effect would be of the comparison of the results obtained on the model to the full-scale prototype if, for example, the modulus of elasticity of the model is different from the modulus of elasticity of the full scale. If the moduli of elasticity are constant in both cases, a direct comparison can be made; for example, as noted in Equation [30]. However, if the modulus of elasticity of one material is not constant, but varies with the stress, this fact has to be taken into consideration in developing the dimensional equation.

Suppose that the stresses in the full-scale structure are to be above the elastic limit. In general, then, if the stresses in the model are not above the elastic limit, no direct application can be made. It appears in this case that the procedure to follow is to so determine the stresses in the model that the same ultimate strength is developed as is expected to be developed in the full-scale prototype. It would mean, then, that in general the elastic limit would be reached at approximately the same per cent of applied load on the model as on the full scale. In other words, it is clear that the experiment on the model should be similar to the experiment on the full-scale prototype in so far as approaching the elastic limit is concerned. In general, however, we note that unless the modulus of elasticity is involved it may not be necessary to give this case consideration. In all problems of deflection, however, this feature must be very carefully considered.

Let us consider another example of the application of the principle of similitude. The problem of vibration in aircraft structures is very important. However, it is quite often very difficult to determine the frequency of vibration in certain types of structures theoretically. We ask ourselves, then, the question as to whether it would not be desirable to construct a model from which we may determine the frequencies and other characteristics necessary. Then, if we find that the model has a certain frequency, we ask what would be the frequency of the full-scale prototype. Let us investigate this phase of the problem. Let us assume that we are considering the vibration of a cantilever wing structure, and to make it still more conventional, let us assume this cantilever wing structure to be uniform and homogeneous throughout its length. The quantities which may affect its vibration we write in Equation [32]:

$$f = F(l, \rho, E) \quad [32]$$

in which f is the frequency and ρ is the mass density per unit volume. We therefore write the equation:

$$f = Kl^a \rho^b E^c \quad [33]$$

Writing this equation in dimensional form, we have

$$\frac{1}{T} = L^a \frac{M^b}{L^{3b}} \frac{M^c}{L^c T^{2c}} \quad [34]$$

Equating the exponents of T, we find that

$$-1 = -2c, \quad c = \frac{1}{2} \quad [35]$$

Equating the exponents of M, we find that

$$0 = b + c, \quad b = -\frac{1}{2} \quad [36]$$

Equating the exponents of L, we find that

$$0 = a - 3b - c \quad [37]$$

Solving these equations, we find that

$$a = -1, \quad b = -\frac{1}{2}, \quad c = \frac{1}{2} \quad [38]$$

Thus we find that

$$f = Kl^{-1}\rho^{-\frac{1}{2}}E^{\frac{1}{2}} \quad [39]$$

or

$$f = \frac{K}{l}\sqrt{\frac{E}{\rho}} \quad [40]$$

Writing the equation of ratio of frequency of the model to that of the full-scale prototype, we find that

$$\frac{f_m}{f_f} = \frac{\frac{K}{l_m}\sqrt{\frac{E_m}{\rho_m}}}{\frac{K}{l_f}\sqrt{\frac{E_f}{\rho_f}}} \quad [41]$$

We note that if we make the modulus of elasticity and the density of the model the same as that of the full-scale prototype:

$$\frac{f_m}{f_f} = \frac{l_f}{l_m} = \frac{Nl_m}{l_m} = N \quad [42]$$

Thus

$$f_m = Nf_f \quad [43]$$

That is, the frequency is inversely proportional to the scale of the model. For example, if the model is $^1/_{10}$th scale, its frequency of vibration will be ten times the frequency of vibration of the full-scale prototype.

Let us apply the principle of similitude to beams subjected to combined transverse and axial loads, from the standpoint of determining the ultimate strength of the beams. This implies stress considerably above the elastic limit; hence our theory based upon Hooke's law does not apply.

We assume, in this case, that we are to compare the strength of, for example, steel or duralumin tubes. We are to perform a series of experiments and from these experiments we are to obtain data which may be applied to the design of tubes in aircraft structures. We assume that the axial stress is a function of the following terms:

$$\frac{P}{A} = F(l, r, t, M) \quad [44]$$

Assume l to be the length, r the radius of gyration, t the thickness of the wall of the tube, M the applied bending moment. Writing the equation in the general form, we have

$$\frac{P}{A} = Kl^a r^b t^c M^d \quad [45]$$

Writing the dimensional equation, we have

$$\frac{M}{LT^2} = L^a L^b L^b \frac{M^d L^{3d}}{T^{2d}} \quad [46]$$

Equating the exponents of M, we find that

$$d = 1 \quad [47]$$

Equating the exponents of T, we find again that $d = 1$. It is therefore apparent that we can find only two unknowns in this case. Equating the exponents of L, we find that

$$-1 = a + b + c + 2d \quad [48]$$

Let us assume that c and a are the known values. We have, therefore

$$b = -3 - (a + c) \quad [49]$$

We note, then, that

$$\frac{P}{A} = K l^a r^{-3-(a+c)} t^c M \quad [50]$$

Rewriting this equation, we find that

$$\frac{P}{A} = K \frac{M}{r^3} \left(\frac{l}{r}\right)^a \left(\frac{l}{r}\right)^c \quad [51]$$

It will be noticed that the arrangement of Equation [51] is not directly obvious. We must use our judgment, based upon past experience with structures, in this arrangement. We note, then, that in this equation the P/A of the material is proportional to the direct bending stress as specified by KM/r^3, and is a function of the ratio of the length to the radius of gyration or to the radius of the tube. Therefore, we find in plotting the data we may use P/A as the ordinate, as in Fig. 1, and the data will be applicable to the full-scale prototype, provided we use the same values of the ratio of length to radius of gyration and also provided that on the curve AB the values taken have a constant ratio of thickness to radius. We note, then, that the scheme of plotting the data as indicated in Fig. 1 is not accurate for tubes unless the ratio of the thickness to the radius is the same for the experiment as it is for the full-scale prototype. It may be noticed in this connection that the variation of the function t/r may be determined from a carefully planned set of experiments, and an empirical formula derived therefrom may be used in connection with the chart.

Let us take the same example and assume that P/A is also a function of additional quantities, such as, for example, the modulus of elasticity E, and the ultimate fiber stress in bending f_{ub}.

$$\frac{P}{A} = F(M, r, t, l, f_{ub}, E) \quad [52]$$

Let us apply the theory of dimensions and see what kind of a story it tells us this time. We find that

$$\frac{P}{A} = KM^a r^b t^c l^d f_{ub}{}^e E^f \quad [53]$$

We have, therefore, from the dimensional equation

$$\frac{M}{LT^2} = \frac{M^a L^{3a}}{T^{2a}} L^b L^c L^d \frac{M^e}{L^e T^{2e}} \frac{M^f}{L^f T^{2f}} \quad [54]$$

Equating the exponents of M, we find that

$$1 = a + e + f \quad [55]$$

Equating the exponents of T, we find that

$$-2 = -2a - 2e - 2f \quad [56]$$

which is the same as Equation [55]. Equating the exponents of L, we find that

$$-1 = 2a + b + c + a - e - f \quad [57]$$

From Equation [55] we find that

$$a = 1 - e - f \quad [58]$$

It is necessary to assume that e, f, and the other remaining exponents are known. We therefore have

$$\frac{P}{A} = KM^{1-e-f} r^{-1-2+2e+2f-c-d+e+f} t^c l^d f_{ub}{}^e E^f \quad [59]$$

Now, we have to use certain judgment in pairing off these terms. For example, we note that the ultimate bending stress will have some connection with bending moment. It is therefore reasonable that they should be paired together. It is apparent that the ratio of the length to the radius of gyration will influence the

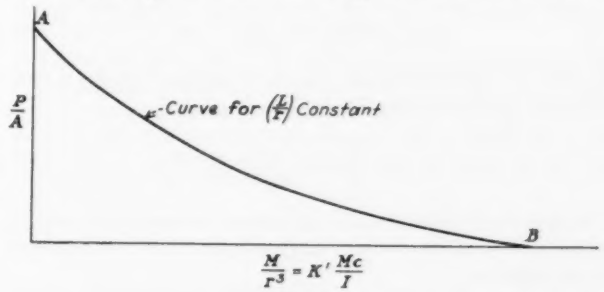


FIG. 1

structure, so it is logical to pair these terms; and in some manner possibly the modulus of elasticity may effect some other term, so that we could pair that with this other term. Therefore, we obtain

$$\frac{P}{A} = K \frac{M}{M^{(e+f)}} \frac{r^{3e} r^{3f}}{r^3 r^c r^d} t^c l^d f_{ub}{}^e E^f$$

$$= K \frac{M}{r^3} \left(\frac{t}{r}\right)^c \left(\frac{l}{r}\right)^d \left(\frac{f_{ub} r^3}{M}\right)^e \left(\frac{E r^3}{M}\right)^f$$

$$= \frac{M}{r^3} F\left(\frac{t}{r}, \frac{l}{r}, \frac{f_{ub} r^3}{M}, \frac{E r^3}{M}\right) \quad [60]$$

Let us consider some of the equations which we have previously derived, from the standpoint of differently assumed variables. For example, let us assume that the stress in bending is a function of

$$f = F(W, l, I) \quad [61]$$

We find from this that

$$f = \frac{W}{l^2} F\left(\frac{I}{l^4}\right) \quad [62]$$

Let us assume that the deformation of a structure is a function of the variables indicated in Equation [63].

$$y = F(P, l, E, I) \quad [63]$$

We find, then, from the theory of dimensions that

$$y = lF\left(\frac{P}{El^2}, \frac{I}{l^4}\right) \quad [64]$$

Let us now consider a torque equation. We assume

$$S = F(q, l, J) \quad [65]$$

in which q is the applied torque, l is the length of the loaded structure, S is the unit shear, and J is the moment of inertia of the cross-section. We find, therefore, that

$$S = \frac{q}{l^3} F\left(\frac{J}{l^4}\right) \quad [66]$$

The torsional deformation let us indicate as

$$\theta = F(q, l, J, E_s) \quad [67]$$

in which E_s is the modulus of elasticity in shear. We, therefore, have

$$\theta = lF\left(\frac{J}{l^4}, \frac{q}{E_s l^3}\right) \quad [68]$$

Now, if we consider the weights of the structure in a similar equation, we have to introduce the new terms, the density of the material and the gravity constant. We have, then,

$$f = F(P, l, I, \rho, g) \quad [69]$$

in which ρ is the density per unit of volume and g is the gravity constant. We find from this that

$$f = \frac{W}{l^3} F\left(\frac{I}{l^4}, \frac{P}{l^4 \rho g}\right) \quad [70]$$

This is the formula which would have to be used in suspension-bridge structural experiments inasmuch as it involves the weight of the structure.

Considering the deflection of the structure in terms of the additional dimensions ρ and g, we have that

$$y = Kl F\left(\frac{I}{l^4}, \frac{P}{El^2}, \frac{P}{l^3 \rho \rho}\right) \quad [71]$$

Writing similar equations for the conditions of torsion, we have

$$S = K \frac{q}{l^3} F\left(\frac{J}{l^4}, \frac{q}{l^4 \rho g}\right) \quad [72]$$

and

$$\theta = Kl F\left(\frac{J}{l^4}, \frac{q}{E_s l^3}, \frac{q}{l^4 \rho q}\right) \quad [73]$$

Likewise in determing the critical loading of a strut, we may add some additional terms, as in the following equations:

$$P = F(l, f_u, E, I, \rho, g) \quad [74]$$

in which P is the strength of the strut, L is the length, f_u is the ultimate strength in bending (the modulus of rupture), E is the modulus of elasticity, I is the moment of inertia in the cross-section, and g is the gravity constant. We find from the theory of dimensions that

$$P = K \frac{EI}{L^2} F\left(\frac{f_u}{E}, \frac{I}{l^4}, \frac{E}{l \rho g}\right) \quad [75]$$

We note in this last equation that the question which was previously mentioned has been answered. That question was as to how the apparent change in the modulus of elasticity at the elastic limit is going to affect the experimental results. We notice in this case that for direct comparison, at least for one term, the ratio of f_b to E of the model must be the same as this ratio for the full scale; that is

$$\left(\frac{f_u}{E}\right)_f = \left(\frac{f_u}{E}\right)_m \quad [76]$$

Again, if we have the two structures geometrically similar and in the same material, the terms I/l^4 and f_u/E will be constants and the factor $E/l\rho g$ may be expressed as the ratio $\frac{P}{l^3 \rho g} \Big/ \frac{P}{El^2}$. It follows that for this last quantity to be constant under the conditions just stated, the loading must be proportional to the weight of the structure and to the square of the scale ratio, since the modulus of elasticity is a constant for the given material. If all the terms within the parenthesis of Equation [75] are thus made constant, there remains

$$P = K \frac{EI}{l^2} \quad [77]$$

This term K is known as the critical load coefficient, and under these implied conditions reduces to the following:

$$K = \frac{Pl^2}{EI} \quad [78]$$

It is observed that in order to make the third term in the parenthesis of Equation [75] constant, it is necessary to make the loads proportional to the square of the dimensions as well as proportional to the weight of the structure. If the model is to be of the same material as the structure, as previously assumed, it must be loaded at certain intervals to bring the effective weight up to the required value. Unless this is done, another material must be chosen such that the terms f_u/E and $E/l\rho g$ both may be made the same in the model as in the structure. Likewise, most of our standard strength of materials formulas may be determined.

Structures of thin sheets of metal, namely, "monocoque" or "stressed skin," present many difficult problems in stress analysis, many of which, even for low stresses well within the range of Hooke's law and subject to mathematical analyses based upon the theory of elasticity, have not been solved. It appears that in this field of research work use may be made of the principle of similitude. Let us consider, for example, the strength of thin-walled tubes to resist torsion or bending. In these cases, we assume that the strength is a function of the modulus of elasticity, the radius of curvature of the metal sheets, and the thickness of the sheets, thus:

$$S = F(E, r, t) \quad [79]$$

Assuming

$$S = KE^a r^b t^c \quad [80]$$

Writing the dimensional equation, we have

$$\frac{M}{LT^2} = \frac{M_a}{L^a T^{2a}} L^b L^c \quad [81]$$

from which we find $a = 1$ and $c = -b$. Thus:

$$S = KE\left(\frac{r}{t}\right)^c = KEF\left(\frac{r}{t}\right) \quad [82]$$

We therefore represent the experimental data, torsion or bending, as allowable fiber stress as a function of r/t.

Among the research problems on thin-sheet-metal construction on which extensive data were obtained this year at the University of California were:

1 Allowable fiber stress in thin duralumin tubes subjected to bending, by F. F. Imperial and R. F. Bergstrom.

2 Allowable fiber stress in thin-walled duralumin tubes subjected to torsion, by A. E. Arslan and W. P. Berggren.

The tubes below 3 in. in diameter were taken from the stock of the Aluminum Company of America. The tubes 3 in. to 10 in. in diameter were from flat sheet and riveted sides of the tube; hence we may assume that the results are practically independent of the riveted joints. In all tests, wooden blocks were fitted in

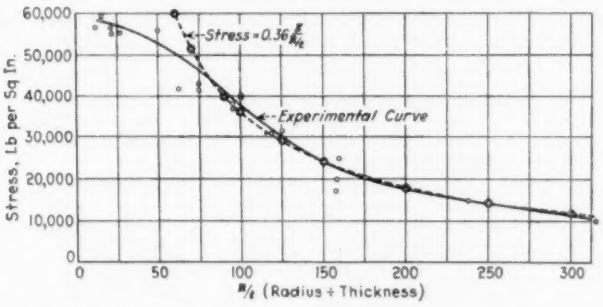


FIG. 2 BUCKLING STRESS OF THIN-WALLED DURALUMIN TUBES IN BENDING

the ends of the tubes, which were 36 in. in length, to which the loads were applied.

The results of the experiments are represented in Figs. 2 and 3. Mr. Bergstrom and Mr. Berggren succeeded in deriving formulas from the standpoint of the mathematical theory of

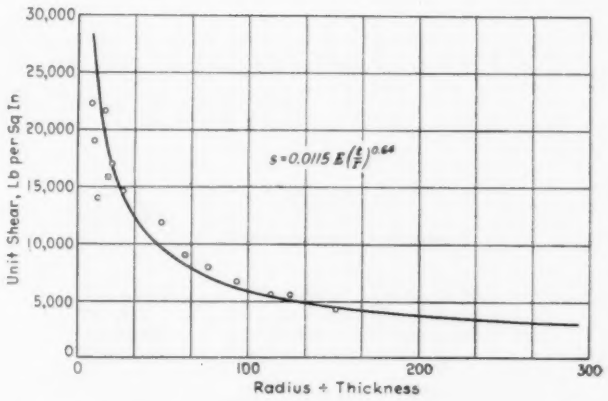


FIG. 3 BUCKLING STRESS IN TORSION OF DURALUMIN TUBES

elasticity which fit the data quite accurately for stresses within the elastic limit (see Figs. 2 and 3). For stresses above the elastic limit an empirical formula will be necessary. The derivation of formulas for Figs. 2 and 3 will be published in the near future by the authors of the experimental data.

We may note, in conclusion, that in so far as the practical use of the data is concerned, since stresses above the proportional limit are involved, a derived formula may not be as desirable as the experimental curve. The value of the experimental curve depends very greatly upon the proper selection of the coordinates and the parameters.

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Duluth — Public Library
Minneapolis — University of Minnesota
Minneapolis Public Library (Engineering and Circulating Libraries)
St. Paul — James Jerome Hill Reference Library

Mississippi

Columbus — Mississippi Agri. & Mech. College

Missouri

Columbia — University of Missouri
Kansas City — Public Library
Rolla — Missouri School of Mines and Metallurgy
St. Louis — Engineers Club of St. Louis
Public Library
Washington University
Mercantile Library

Montana

Bozeman — Montana State College

Nebraska

Lincoln — University of Nebraska
Omaha — Public Library

Nevada

Reno	University of Nevada Library

New Hampshire

Durham	University of New Hampshire

New Jersey

Bayonne	Free Public Library
Camden	Free Public Library
Elizabeth	Free Public Library
Hoboken	Stevens Institute of Technology
Jersey City	Free Public Library
Newark	Newark College of Engineering
	Free Public Library
New Brunswick	Rutgers University
Paterson	Free Public Library
Princeton	Princeton University
Trenton	Free Public Library

New York

Albany	New York State Library
Brooklyn	Polytechnic Institute
	Pratt Institute
	Brooklyn Public Library
Buffalo	The Grosvenor Library
	Engineering Society of Buffalo
	Buffalo Public Library
Ithaca	Cornell University
Jamaica, L. I.	Queens Borough Public Library
New York	Engineering Societies Library
	Public Library
	College of the City of New York
	Cooper Union
	Columbia University
	N. Y. University Library
Potsdam	Clarkson College of Technology
Rochester	Rochester Engineering Society
Schenectady	Union College
Syracuse	Syracuse University
	Public Library
Troy	Rensselaer Polytechnic Institute
Utica	Public Library
Yonkers	Public Library

North Carolina

Chapel Hill	University of North Carolina (Engineering Library)
Raleigh	North Carolina State College

North Dakota

Fargo	North Dakota State College
Grand Forks	University of North Dakota

Ohio

Ada	Ohio Northern University
Akron	Public Library
	University of Akron
Canton	Public Library
Cincinnati	University of Cincinnati
	Public Library
	Engineers Club of Cincinnati
Cleveland	Public Library
	Case School of Applied Science
	Cleveland Engineering Society
Columbus	State of Ohio Library
	Public Library
	Ohio State University
Dayton	Engineers Club of Dayton
Toledo	Public Library
	University of Toledo
Youngstown	Public Library

Oklahoma

Norman	Oklahoma University
Oklahoma City	Public Library
Stillwater	Oklahoma Agricultural and Mechanical College
Tulsa	Public Library

Oregon

Corvallis	Oregon State Agricultural College
Portland	Portland Library Association

Pennsylvania

Allentown	Free Library
Bethlehem	Lehigh University
Easton	Public Library
	Lafayette College
Erie	Public Library
Lewisburg	Bucknell University
Philadelphia	Engineers Club
	Drexel Institute
	University of Pennsylvania
	Franklin Institute
Pittsburgh	University of Pittsburgh
	Engineers' Society of Western Pennsylvania
	Carnegie Institute of Technology
	Carnegie Library (Schenley Park)
	Carnegie Free Library of Allegheny
Reading	Public Library
Scranton	Public Library
State College	Pennsylvania State College
Swarthmore	Swarthmore College
Villanova	Villanova College
Wilkes-Barre	Public Library

Rhode Island

Kingston	Rhode Island State College
Providence	Brown University
	Providence Engineering Society
	Public Library

South Carolina

Clemson College	Library, Clemson College

Tennessee

Kingsport	Public Library
Knoxville	University of Tennessee
Memphis	Goodwin Institute
Nashville	Vanderbilt University

Texas

Austin	University of Texas
College Station	Agricultural & Mechanical College of Texas
Dallas	Public Library
El Paso	Public Library
Fort Worth	Carnegie Public Library
Houston	Rice Institute
	Public Library
Lubbock	Texas Technological College (School of Engineering)
San Antonio	Carnegie Library

Utah

Salt Lake City	University of Utah
	Public Library

Vermont

Burlington	University of Vermont

Virginia

Blacksburg	Virginia Polytechnic Institute
Charlottesville	University of Virginia
Norfolk	Carnegie Public Library
Richmond	Virginia State Library

Washington

Pullman	State College of Washington
Seattle	Public Library
	Engineers Club
	University of Washington
Spokane	Public Library
Tacoma	Public Library

West Virginia

Morgantown	West Virginia University

Wisconsin

Madison	Library, University of Wisconsin
Milwaukee	Public Library
	Board of Industrial Education, Vocational School Library
	Marquette University

Wyoming

Laramie	Wyoming University

Progress in Applied Mechanics

Theory of Elasticity and Stress Analysis[1]

DURING the last year several problems in applied elasticity, having practical interest, have been discussed. Among the papers presented at the last two meetings of the Applied Mechanics Division of the A.S.M.E. was one by J. N. Goodier, on "Compression of Rectangular Blocks" (1).[2] This paper discussed local stresses produced by concentrated forces, and investigated also the accuracy of the ordinary bending theory when the forces applied at the ends do not follow a linear law. Other papers, by A. G. Solakian and G. B. Karelitz (2) and M. M. Frocht (3), dealt with the application of photoelasticity in the solution of design problems.

During the last National Meeting of the Applied Mechanics Division (New Haven, June, 1932) several papers on stress analysis were presented. One by Dr. A. M. Wahl (4) gave an approximate solution of the problem of the deflection of a semicircular plate under uniform load. The theory was checked by experiments on models, and found application in the calculation of the deflection of semicircular diaphragms in steam turbines.

A paper by Dr. M. A. Sadowsky (5) gave some interesting solutions of local stresses at the surface of contact between two bodies in compression.

A. N. Dinnik (6) presented a number of solutions of the problem of lateral buckling of columns of variable cross-section. Some of his results are of practical interest for airplane construction and for bridges.

A paper by H. Hencky (7) on the elastic behavior of vulcanized rubber gave a theory for materials with large deformations which is in good agreement with experiments.

A paper by Stewart Way (8) discussed the problem of the deflection of long rectangular plates uniformly loaded. It also gave some examples of the application of theoretical results to the design of ships.

J. N. Goodier (9) discussed the concentration of stresses around spherical and cylindrical inclusions and flaws. His paper may prove of interest in discussing some results of fatigue tests of weld metal having spherical cavities. Often fatigue cracks are originated at these cavities.

Papers by W. O. Richmond (10) and T. H. A. Brahtz (11) contained some new solutions of the two-dimensional problem in elasticity. Dr. P. A. Cushman (12) presented a paper dealing with some experimental results of the application of the soap-film method in studying stress distribution in bars submitted to the action of torsion and bending.

Papers by Dr. A. Nadai (13) and Dr. G. H. MacCullough (14) discussed several important problems on plastic flow of metals and set forth some new experimental results.

Last May, at the first International Congress for Bridge and Structural Engineering, which was held at Paris, a considerable number of papers dealing with applied elasticity were presented. Some of these dealt with problems of elastic stability, among them being one by Prof. L. Karner, in which the new developments in columns were discussed. A paper by F. Bleich was devoted to local buckling in the wall of a tubular compressed member; another, by Prof. M. Ros, discussed the problem of bending of eccentrically loaded columns; while a third, by the writer, dealt with the buckling of the webs of plate girders and on the design of stiffeners.

Several important papers on the theory of elasticity have appeared abroad during the last few years, of which four will be mentioned. One by O. Göhner (*Ingenieur-Archiv*, vol. 1, p. 619, and vol. 2, pp. 1 and 381) deals with stresses in helical springs. In the case of heavy helical springs, the diameter of the wire is not small in comparison with the diameter of the helix, and the usual elementary theory gives too low stresses. Some improvements in the theory have been made by V. Roever, and more recently by A. M. Wahl, the latter giving also an experimental proof of his formula. Göhner presents a complete theoretical solution of the problem, and the final results are in good agreement with Wahl's approximate formula. A paper by E. Melan (*Ingenieur-Archiv*, vol. 3, p. 123) deals with problems which are of practical interest in investigating stresses in welded structures. The author considers the case of a plate reinforced along a straight edge by a rib, and the stresses produced by a longitudinal force applied to the rib. A paper by H. Reissner (*Zeit. f. angew. Math. u. Mech.*, vol. 11, p. 1, 1931) gives the general theory of stress distribution produced in elastic bodies by non-uniform heating or by some permanent set. Another, by L. Föppl (*Zeit. f. angew. Math. u. Mech.*, vol. 11, p. 81, 1931), gives a new method of solving two-dimensional problems in the theory of elasticity, and applies it to the discussion of stresses around an elliptical hole.

During the last year two important books in the field of theory of elasticity have appeared. One, by E. G. Coker and L. N. G. Filon, on "Photoelasticity" (Cambridge University Press) deals at length with the technic of this kind of experimental study of stress distribution, but also presents a theory of the two-dimensional problem in elasticity and a description of the physical properties of materials used in experiments of this nature. It is the most complete treatise on photoelasticity that has yet appeared, and it will be of interest to all who are working in this field.

The second book, by Dr. A. Nadai, on "Plasticity" (McGraw-Hill) is an enlarged English edition of a work first published in German. It treats systematically various problems on plastic flow in metals, and will be of decided value to engineers working in the field of plastic deformation.

Rigid Dynamics[3]

In the field of dynamics two outstanding papers were presented at the New Haven meeting of the A.S.M.E. The first comprises an elegant analytical solution by F. M. Lewis (15) of the problem of the amplitude of a vibrating system which is accelerated or decelerated at a given rate through its critical speed. The other paper, by J. G. Baker (16), throws new light on the phenomenon of self-induced vibrations in engineering. It propounds the thesis that too much importance is usually attached to "resonance," because a large number of trouble cases are caused by "self-induction" rather than by "resonance."

At the Bigwin meeting in June, 1932, a symposium (17) was devoted to the subject of model experiments and dimensional theory.

In Germany a book on automobile shimmy was published by

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[1] Compiled by Dr. S. Timoshenko.

[2] Numbers in parentheses refer to similarly numbered references given at the end of the report.

Presented at the session of the Applied Mechanics Division, Tuesday morning, December 6, 1932, during the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers, 29 West 39th Street, New York, N. Y.

Personnel of the Executive Committee of the Applied Mechanics Division for 1932: Geo. B. Pegram, Chairman; J. M. Lessells, Secretary; J. A. Goff, F. M. Lewis, and E. G. Waters.

[3] Compiled by J. P. DenHartog.

Becker, Fromm, and Maruhn, giving a discussion of the underlying theory and elaborate of tests on this complicated subject.

It is well known that during the last decade or two great advances have been made in the applications of the gyroscope, with which the great name of Sperry is closely associated. During the past year a book entitled "Applied Gyrodynamics," by E. S. Ferry (18), has been published which describes this entire development in a logical and easily readable form. It will undoubtedly help to bring this fascinating subject within the understanding of a greater group than heretofore.

Mechanics of Materials[4]

The New International Association for Testing Materials held a congress at Zurich, Switzerland, September 6 to 12, 1931. Four groups were organized: A, Metals; B, Nonmetallic inorganic materials; C, Organic materials; and D, Questions of general importance. In group A, papers were presented by leading authorities on such subjects as testing of cast iron, strength of metals at high temperatures, fatigue, impact testing, properties of light alloys, and strength of welds. The papers presented at the congress are available in four volumes corresponding to the above groups (19).

Considerable interest is being shown in the properties of rubber as an engineering material. A Symposium on Rubber was held by the A.S.T.M. on March 9, 1932, in Cleveland. Seven papers were presented on various properties of rubber (20). A noteworthy paper on the elastic behavior of rubber (21) was presented at the New Haven meeting of the A.S.M.E. A rubber-litharge material, having the same specific gravity as concrete, but with a much lower modulus of elasticity, has been used in constructing a test model of an arch dam (22).

Several contributions have been made to our knowledge of creep of metals, both from the experimental (23) and theoretical (24) aspect. A book collecting much valuable information on the subject has been published in England (25).

In the fatigue-testing field more attention is being paid to elements of construction (26) and stress-concentration effects (27). The strengthening effect in fatigue of cold rolling a surface has been further demonstrated (27). In this connection it is of interest to find that an indentation made with the conical end of a center punch into the surface of a fatigue specimen did not lower the endurance limit (27); whereas a similar-sized countersunk hole (made with a conical end cutter) decreased the endurance limit considerably. In the former case, a cold-working effect is present. An interesting paper, showing the variation of damping with the number of cycles in the case of specimens tested at the endurance limit, has been published recently (28).

A means of testing models under the action of centrifugal force, simulating the action of gravitational force on large structures and geological strata, has been developed (29).

Several papers on the properties of zinc-coated steel were presented at the A.S.T.M. annual meeting at Atlantic City, June 23, 1932.

Other important contributions have been made concerning the strength of cast iron, effect of nitriding, strengthening of machine parts (screws) by cold working, etc.

It is impossible to cover the subject completely in a report of this kind, but it is hoped that at least the directions of progress have been indicated.

Hydrodynamics and Aerodynamics[5]

The problems of turbulence and of fluid resistance are still in the foreground of research work in hydrodynamics.

Turbulence. Considerable work has been done in investigating the mechanisms of turbulence (30). Experiments are being continued regarding the effect of a temperature gradient perpendicular to the direction of flow with respect to the rate of turbulence. By these investigations some light has been thrown on the complicated nature of the air movement within the atmosphere.

A new and very interesting experimental method for studying the internal movement of turbulent flow has been developed (31). By using an ultramicroscope, very interesting results have been obtained, particularly regarding the detail of the flow in the immediate neighborhood of a solid wall.

Fluid Resistance. Another step forward has been taken in the theoretical treatment of frictional resistance due to turbulent flow (32). The theory of fluid resistance developed by L. Prandtl and Th. von Kármán has been checked by extensive tests on turbulent flow through smooth and rough pipes (33).

The problem of fluid flow through bendings of pipes and the loss of pressure involved has been the subject of several papers.

Tests on wind pressure on buildings have been carried out at various laboratories (34).

Experimental work at laboratories handling shipbuilding problems has been focused on the mutual interference between ship and propeller. Several attempts have been made to obtain a more uniform velocity field at the propeller area by applying different types of guide vanes in front of the propeller. A large number of tests have been carried out to investigate the performance of seaplane hulls, particularly regarding the impact pressures due to waves when starting or landing the seaplane.

Cavitation. In connection with shipbuilding problems, considerable research work has been done on cavitation (35). The question of the influence of various shapes of propeller-blade sections on cavitation has been investigated. In addition, a study has been made to determine whether a certain time is necessary for developing cavitation and whether such a time effect may be of importance in engineering problems. Furthermore, an experimental investigation has been carried out to learn how far cavitation actually extends in special cases.

Fluid Flow. Extensive tests on the effect of drawing off parts of the boundary layer of airfoils by suction in order to increase their maxmum lift have been completed (36).

Several tests have been carried out with the purpose of influencing the flow around bodies or through diffusers of large diffuser angles by means of guide vanes in order to enforce potential flow lines. If this could be accomplished, a considerable energy loss due to eddying flow could be avoided.

In connection with wind-tunnel work an interesting paper has been published (37) on the noise produced by the air flowing with high speed through the open throat of a wind tunnel.

Considerable effort has been spent in investigating the influence of the compressibility of air in connection with the performance of thin airfoil sections having speeds near or beyond the acoustic velocity. This problem is of great importance due to the fact that the peripheral speed of a modern airplane propeller sometimes reaches, and even exceeds, the acoustic velocity.

The problem of the resistance of projectiles moving with supersonic velocities has been treated in a theoretical manner (38). The results obtained are in good agreement with experimental data.

Lubrication[6]

Continued interest has been demonstrated in this branch of applied mechanics throughout the past year.

Further work has been done in verifying the applicability of

[4] Compiled by R. E. Peterson.

[5] Compiled by O. G. Tietjens.

[6] Compiled by G. B. Karelitz and L. M. Tichvinsky.

the hydrodynamic theory of lubrication to actual bearings. The pressure distribution and journal location in the bearing have been measured, and additional evidence of the correspondence of the hydrodynamical theory with experimental observation has been thus collected (39). The effect of end sealing with pressure oil has been investigated experimentally (40). The friction in bearings with concentric and eccentric shafts has been measured. It has been demonstrated that with concentric journals the observed friction is close to the value computed on the basis of the hydrodynamic film, while a deviation was found when the journal ran eccentrically (41).

The analytical treatment of oil films has been furthered in a paper by A. Kingsbury (42), whose numerical tables and diagrams render the theoretical considerations regarding optimum conditions in bearings of assistance in bearing design.

Much attention has been paid to proper methods of testing oils and to the lubricating qualities of oils. A number of oil-testing machines have been proposed (43–45). A method of evaluating lubricating quality by means of the wetting ability of an oil has been proposed before the Lubrication Committee of the Oil Power Division. Lubrication of rolling-mill bearings has continued to attract attention (46, 47). In connection with these bearings, attempts have been made to apply new materials other than bronzes or lignum-vitae. Celoron (48) and micarta have been suggested as heavy-duty bearing materials. Observations on water-lubricated rubber bearings have been published, showing a very low coefficient of friction (49).

The question of wear of metals has brought forth several contributions giving results of experimental investigations (50, 51). Measuring the finish of metal surfaces has been also given attention, and instruments for the quantitative evaluation of finish are being developed and will be reported to the Committee in the near future.

Thermodynamics[7]

During the year just past, several important contributions to the knowledge of the thermal properties of steam have been made both in this country and abroad. Osborne and Stimson (52), reporting on the progress of research at the Bureau of Standards, have described the design, construction, and preliminary operation of a complete new saturated-steam calorimeter with which they are equipped to extend the range of direct measurements of the enthalpy of saturated steam and saturated water. Smith and Keyes (53) have submitted a body of new and more accurate pressure-volume-temperature data covering the range of volumes from 140 to 5 cu cm per gram, of pressures from about 14 to about 360 atm., and of temperatures up to 460 C. Koch (54) has published new measurements of the specific heat (at constant pressure) for superheated steam which have extended the range of pressures from 120 to 240 atm. with temperatures running from saturation to about 450 C. Later, Koch (55) has published an account of his investigations into the behavior of water in the critical region. His measurements of specific heat along the 240-atm. isobar flatly contradict the contention of Callendar that a discontinuity should exist to indicate a considerable value for the latent heat of vaporization at this pressure.

Hausen (56) has attacked the problem of formulating the properties of steam in a new and very interesting manner. Starting with a quantum-theory calculation of C_{p0}, the specific heat for zero pressure, based on optical measurements of the so-called characteristic frequencies for the water molecule, he forms the difference $C_p - C_{p0}$ from available specific-heat data and finds an empirical equation which represents the variation of this difference with temperature and pressure. Applying the well-known thermodynamic relations, he derives equations for the volume and the enthalpy to within certain unknown pressure functions, which are then evaluated from the available experimental measurements of these quantities. The very satisfactory agreement of the resulting formulation with the experimental data marks it as a distinct contribution to steam research.

Keyes and Smith (57), in their efforts to correlate various data on properties of saturated water and steam and from a consideration of various gas-thermometry investigations, have been led to suggest a revision of the accepted value for the absolute temperature of the ice point. Preliminary calculations indicate a value of 273.16 C instead of 273.1 C.

Jakob (58) has given a masterful survey of steam research in Europe and America in a series of four lectures before the University of London. The fourth lecture is of especial interest as it calls attention to the increasing importance of investigations into what Professor Jakob calls the "dynamical properties" of water and steam. These include viscosity, heat conductivity, and coefficient of radiation. E. Schmidt (59) has recently published measurements on the total radiation of steam for temperatures up to 1000 C.

Only briefest mention can be made of other important investigations in the field of thermodynamics. Factors governing the transfer of heat by fluids in motion are being studied with a view toward rationalizing the design of heat-transfer equipment. The actual mechanism of evaporation is being subjected to careful examination. Studies of the mixing process so important in the application of rectification to the separation of various mixtures into their constituents have been described. The measurement of fluid flow by means of orifices and nozzles has been so far advanced as to almost place it on a rational basis.

Of immediate interest to the turbine engineer are the experiments of Burghauser (60), which indicate that friction at the walls of blade passages can only account for a very small proportion of the loss of efficiency actually measured. Keenan (61) has called attention to a paper by Darrieus (62), and, on the basis of it, has recommended certain changes in the customary method of judging the performance of the steam turbine.

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Self-Induced Vibrations

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A "self-induced" vibration is defined as a phenomenon in which the alternating forces furnishing the energy to the vibration are controlled by the motion, in contradistinction to a "forced" vibration, where the force depends on time only. The following are examples of self-induced vibrations: (1) All bowed string or blown musical instruments, (2) the vacuum-tube oscillator, (3) fluttering of valves in air or water lines, (4) nosing of locomotives and street cars, (5) airplane-wing flutter, (6) certain cases of hunting of generators, and (7) certain cases of vibration of transmission lines due to wind. In physics and electrical engineering self-induced vibrations are common and often very useful, whereas most mechanical or machinery vibrations do not fall in this classification, since unbalance forces and other alternating forces unaffected by the motion are very common. Kimball and Newkirk[2] were the first to call attention to and to explain self-induced vibration phenomena capable of causing serious mechanical difficulties. This paper discusses methods of studying self-induced vibrations and describes several representative cases that have been studied by the author and his colleagues in the last few years.



1 Introduction

A PHENOMENON in which a force causes a vibration and the vibration controls the force will be called herein a self-induced vibration. Before the time of modern machinery, this type of vibration was most important. Except for the piano and other instruments in which the vibrations are caused by suddenly applied or suddenly removed forces, practically all musical instruments produce sound by means of self-induced vibrations; for instance, the flute, organ, clarinet, and cornet.[3]

Many toys that have appeared on the market recently are excellent examples of this type of motion.[4] Nearly all pendulum and balance-wheel oscillations in clocks and other timing devices are self-induced vibrations. The doorbell, the automobile horn, the oscillating vacuum-tube circuit, and the power-driven tuning fork are apparatus that function on account of self-induced vibrations.

To the electrical engineer who has encountered the vacuum-tube oscillator and the d-c to a-c inverter, the phenomenon of self-induced vibration has been duly emphasized. To the mechanical engineer, on the other hand, the importance of this type of motion has been obscured somewhat by the great attention he has been required to give to resonance phenomena.

In the last few years several serious cases of self-induced vibration trouble have appeared in the field of mechanical engineering. It was therefore thought appropriate to review the subject with the point of view of the mechanical engineer, and to describe and attempt to explain some typical self-induced vibration phenomena.

2 General Relations

The solution of the problem in general consists in answering the following questions:

1 Given a system of masses, flexibilities, etc., will the system be capable of self-induced vibrations? And if so:
2 How will it vibrate—e.g., what will its mode of vibration be?
3 What frequency and what amplitude will it have?
4 To what extent may the vibration be affected? and
5 By what means?

The system in question may or may not be linear. Consider first that it is linear—viz., that each force in the system is proportional to the displacement or to the derivative of the displacement of some mass of the system. Let the masses be $m_1, m_2, m_3, \ldots . m_n$. Let the displacements of these masses be $x_1, x_2, x_3, \ldots . x_n$. With all displacements held permanently at zero except x_1, let the force on m_1 (due to the displacement x_1) be $-k_{11}x_1$. Let the force on m_2 (due to the displacement x_1) be $-k_{21}x_1$, etc. With all the displacements held at zero except x_1, and with x_1 zero but the first derivative dx_1/dt not zero, let the force on m_1 (due to the velocity dx_1/dt) be $-c_{11}(dx_1/dt)$ and the force on m_2 be $-c_{21}(dx_1/dt)$, etc. Finally, considering the inertia as a force, the inertia force of m_1 will be $-m_1(d^2x_1/dt^2)$. Due to the same acceleration, the force on m_2 will be $-m_{21}(d^2x_1/dt^2)$. Thus the first digit of the subscript indicates upon which mass the force acts, and the second digit indicates by which displacement the force is caused. For example, $-c_{nm}(dx_m/dt)$ would mean a force on mass m_n due to the velocity dx_m/dt of mass m_m. From the conditions of equilibrium we may write one equation for each mass, as follows:

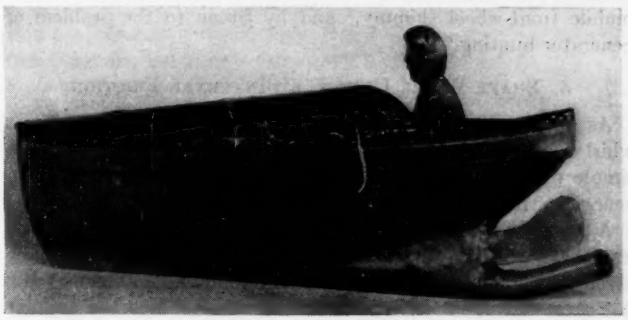


Fig. 1 Toy Boat

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[2] Articles by B. E. Newkirk and A. L. Kimball in *Gen. Elec. Rev.*, March and April, 1924.

[3] A. B. Wood, "A Textbook on Sound," p. 175.

[4] They consist mostly of animals of various shapes having the legs freely pivoted to the body. When put on a floor and pulled forward by means of a string, the toy does not move forward uniformly, but walks in definite steps, accompanied by a sideways oscillation. Another interesting toy is the "put-put boat" shown in Fig. 1 and explained later in the paper.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

$$\left.\begin{aligned} m_{11}\frac{d^2x_1}{dt^2} + c_{11}\frac{dx_1}{dt} + k_{11}x_1 + m_{12}\frac{d^2x_2}{dt^2} + c_{12}\frac{dx_2}{dt} + k_{12}x_2 \\ + m_{1n}\frac{d^2x_n}{dt^2} + c_{1n}\frac{dx_n}{dt} + k_{1n}x_n = 0 \\ m_{21}\frac{d^2x_1}{dt^2} + c_{21}\frac{dx_1}{dt} + k_{21}x_1 + m_{22}\frac{d^2x_2}{dt^2} + c_{22}\frac{dx_2}{dt} + k_{22}x_2 \\ + m_{2n}\frac{d^2x_n}{dt^2} + c_{2n}\frac{dx_n}{dt} + k_{2n}x_n = 0 \\ m_{n1}\frac{d^2x_1}{dt^2} + c_{n1}\frac{dx_1}{dt} + k_{n1}x_1 + m_{n2}\frac{d^2x_2}{dt^2} + c_{n2}\frac{dx_n}{dt} \\ + k_{n2}x_n + m_{nn}\frac{d^2x_n}{dt^2} + c_{nn}\frac{dx_n}{dt} + k_{nn}x_n = 0 \end{aligned}\right\} \quad \ldots [1]$$

Equations [1] are representative of any linear system. The complete solution of [1] for x_n in general will be the sum of a number of terms of the form:

$$At^m e^{at} \sin(\omega t + \theta) \quad \ldots [2]$$

in which A and θ are constants determined by the initial conditions of the system; t is time; m is 0, 1, 2, ... (usually zero), and ω and a are constants determined by the parameters of the system in question. If any term of this solution has a positive value of a, the system is evidently capable of self-induced vibrations.

The foregoing method of attack of the problem has been used extensively, among others by Lord Rayleigh in determining the stability of certain hydrodynamic flows, and by the electrical engineers in determining the conditions for oscillation of vacuum-tube circuits. Applications to mechanical problems have been made by Becker, Fromm, and Maruhn on the problem of automobile front-wheel shimmy,[5] and by Stone to the problem of generator hunting.[6]

3 Shaft Whirl Caused by Internal Friction

As an example of the foregoing treatment, the case of shaft whirl due to damping in a rotating shaft will be considered.[7] A simple explanation of the effect can be given by means of an energy consideration.

A perfectly balanced disk m (Fig. 2) is mounted on the weightless shaft S. The latter is straight without load. Let m be given a constant angular speed ω. The angular displacement of a dot d on m will be ωt, where t is the time. Since the angular acceleration of the rotor is zero and since the shaft is attached

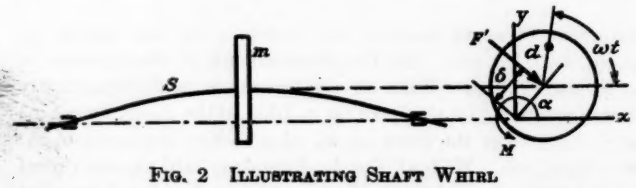


Fig. 2 Illustrating Shaft Whirl

at the center of mass of the rotor, neglecting air friction and assuming m to be mounted symmetrically between the bearings, the only reaction m can exert on S is a *force*. Let α be the angular displacement of the center of the disk m. The angular velocity of this point will then be $d\alpha/dt$. With a given radial displacement of the center of m, the material of S is undergoing alternating stresses at a frequency proportional to $\omega - (d\alpha/dt)$.

Kimball's experiments[7] indicate that the energy absorbed per cycle of alternating stress is independent of the frequency of stress alternation and is proportional to the square of the stress. Other experiments[8] are not in all cases in agreement with the latter of these conclusions; however, for the purpose at hand we shall assume both of Kimball's conclusions to be correct. Imagine that we are rotating at an angular velocity $d\alpha/dt$ and observing the shaft from the right end as represented in Fig. 3. We shall see the dot d rotating at an angular velocity $\omega - (d\alpha/dt)$. Each revolution we observe will represent one cycle of change in stress of any particle of the shaft material. The energy dissipated during this cycle will be proportional to the displacement squared, thus:

$$E = 2\pi D\delta^2 \quad \ldots [3]$$

where E is the energy dissipated per cycle and D is a constant of proportionality determined by integrating over the shaft. This energy must be supplied by an external moment M applied to S or m, if ω is to be kept constant.[9] From the conservation of energy in the shaft, we have:

$$M \cdot 2\pi = 2\pi D\delta^2 \quad \ldots [4]$$

But M must have a reaction. A reaction at the bearing would mean more energy dissipated and serve only to make M larger than represented in [4], We have also seen that m cannot exert a *moment*. It is concluded. then, that the reaction of M must be a force F' exerted by m on S perpendicular to the direction of the deflection δ; and the shaft exerts a force F on m equal and opposite to F'. So that, from [4]:

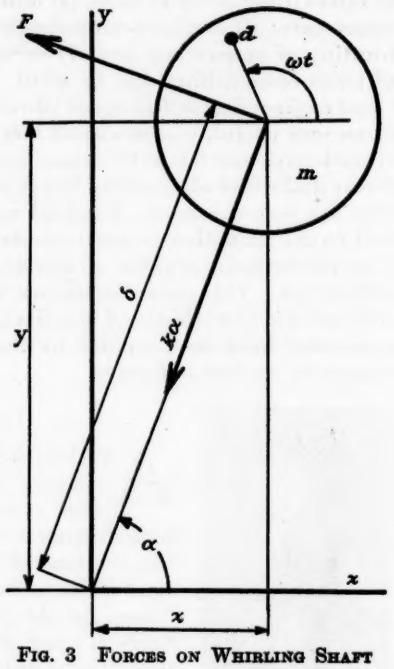


Fig. 3 Forces on Whirling Shaft

$$F\delta = D\delta^2$$

or

$$F = F'v = D\delta \quad \ldots [5]$$

If the shaft be rotated so that the frequency $\omega/2\pi$ is higher than the natural frequency of the rotor on the shaft, a free vibration of the center of M such that δ is constant and $d\alpha/dt$ is positive will result in F being in the direction of motion of the center of m. Thus F will maintain or increase this type of motion of m. On the other hand, if the rotational frequency $\omega/2\pi$ is lower than the natural frequency, it is impossible for F to act wholly or partly in the direction of motion. F therefore cannot maintain a vibration with $\omega/2\pi$ below the natural frequency.

The foregoing statements are better demonstrated by the

[5] G. Becker, H. Fromm, and H. Maruhn, "Schwingungen in Automobilaufhängungen," Berlin, 1931, p. 109.

[6] M. Stone, "Non-unilateral Oscillations in Governor-Controlled Engine-Driven Generators," *Jl. of the Franklin Inst.*, vol. 214, no. 6, Dec., 1932.

[7] Kimball and Lovell, *Mechanical Engineering*, May, 1927, p. 440.

[8] Föppl, Becker, and Heydekampf, "Die Dauerprüfung der Werkstoffe," Berlin, 1929.

[9] The conclusions are not altered when the shaft is allowed to slow down freely.

following analysis, similar to that described under section 2. In Fig. 3 the rotor is shown isolated, while the effect of the shaft is represented by the elastic force $k\delta$ and the tangential force F. Let x and y be the coordinates of the rotor center. The equilibrium equations in the x and y directions give:[10]

$$\left.\begin{aligned} -F\sin\alpha - kx &= m\frac{d^2x}{dt^2} \\ F\cos\alpha - ky &= m\frac{d^2y}{dt^2} \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [6]$$

where m indicates the mass of the rotor. From [5] and [6]:

$$\left.\begin{aligned} m\frac{d^2x}{dt^2} + kx + Dy &= 0 \\ m\frac{d^2y}{dt^2} + ky - Dx &= 0 \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [7]$$

which are seen to fit the general scheme [1].

The solution of [7], derived in the Appendix, is:

$$\left.\begin{aligned} y &= e^{nt}[C_1\cos qt + C_2\sin qt] \\ &\quad + e^{-nt}[C_3\cos qt + C_4\sin qt] \\ x &= e^{nt}[C_1\sin qt - C_2\cos qt] \\ &\quad - e^{-nt}[C_3\sin qt - C_4\cos qt] \end{aligned}\right\} \quad \ldots\ldots [8]$$

C_1, C_2, C_3, and C_4 are constants of integration determined by the initial conditions, while n and q are given by:

$$n = \sqrt{\frac{-k + \sqrt{k^2 + D^2}}{2m}} \quad \ldots\ldots\ldots\ldots [9]$$

$$q = \sqrt{\frac{k + \sqrt{k^2 + D^2}}{2m}} \quad \ldots\ldots\ldots\ldots [10]$$

Usually D is small compared to k, in which case:

$$n = \frac{1}{2}\frac{D}{\sqrt{km}} \quad \ldots\ldots\ldots\ldots [9a]$$

$$q = \sqrt{k/m} \quad \ldots\ldots\ldots\ldots [10a]$$

so that the motion in any single direction is a vibration at the natural frequency $\frac{1}{2\pi}\sqrt{k/m}$ of the rotor on the shaft, which of course corresponds to the ordinary critical or whirling speed. The second terms of Equations [8] will disappear with increasing time, so that they can be neglected. The whirling amplitude $\delta = \sqrt{x^2 + y^2}$ can then be calculated with the help of [8]:

$$\delta = e^{nt}\sqrt{C_1^2 + C_2^2} = \text{const}\, e^{nt} \quad \ldots\ldots\ldots [11]$$

Thus the result [8] shows that the motion will be a rotation of the center of m at a radius δ, at a frequency $q/2\pi$ *in the same direction* as the rotation of the shaft material. It should be kept in mind that Equations [6] only apply if $\omega - (d\alpha/dt)$ or $\omega - q$ is positive—that is, for running speeds above the critical speed. For the case of a running speed below the critical speed, the phenomenon cannot be described by a single set of differential equations, because the direction of F will depend on the direction of whirl. A qualitative examination, however, will show that, below the critical speed, the shaft damping effect retards any vibratory movement that may be started.

[10] The deflection due to gravity introduces a complication in the analysis which would obscure somewhat our present purpose. Taking gravity into account, Equation [6] applies if the dynamic deflections are large compared to the gravity deflections. Practically, gravity helps in that the shaft will not vibrate due to elastic hysteresis until an initial disturbance occurs, producing deflections at least as large as the gravity deflection. Thus in passing through a critical speed with a horizontal shaft having appreciable internal damping, we may expect the shaft to continue to vibrate at its natural frequency even though the rotating speed of the machine is far above the speed corresponding to the natural frequency; but if in any way the vibration can be momentarily reduced to below the value of the static deflection, the vibration will be damped out.

4 Amplitudes of Vibration

It may be concluded from [2] that self-induced vibrations in linear systems can never be of constant amplitude except in the trivial case corresponding to $a = 0$ in [2]. The inverted statement is perhaps more important: namely, any vibrating system with self-induced vibrations of constant amplitude is not linear, with the same exception as before.

In the case of non-linear systems, the answers to the questions in a previous paragraph are more difficult. Some non-linear systems can, with very good accuracy, be treated as linear for small vibrations; in fact, this is perhaps the most common use of the linear treatment. For this type of system, the question, Will the system start to vibrate? may be answered.

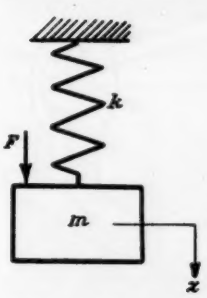


Fig. 4 Scheme of Pendulum Clock

The building up of self-induced vibrations in non-linear systems has been studied with non-linear differential equations for some cases by Van der Pol.[11]

5 Frequency of Vibration

Consider a mass m connected to a linear spring k, as shown in Fig. 4. Let this mass be kept in a state of steady oscillation by impulses F timed with its position. Under these conditions, the displacement x can be represented by:

$$x = S_1\sin\omega t + S_2\sin 2\omega t + R_2\cos 2\omega t + S_3\sin 3\omega t \ldots [12]$$

and the impulse force F can be written as:

$$F = A\sin\omega t + B\cos\omega t + A_2\sin 2\omega t + B_2\cos 2\omega t \ldots [13]$$

S_1, S_2, etc., A, B, etc., are constants. The ratio A/B depends on the timing of the impulses with respect to the motion; A_2, B_2, etc., will depend on A and B and on the nature of the impulses. S_1, S_2, R_2, etc., are found in terms of A, B, A_2, B_2, etc., by substitution of [12] and [13] in

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx - F = 0 \quad \ldots\ldots [14]$$

which expresses the equilibrium of the mass. From this substitution follows:

$$\left.\begin{aligned} A &= -mS_1\omega^2 + kS_1 \\ B &= cS_1\omega \end{aligned}\right\} \quad \ldots\ldots [15]$$

Dividing these by each other and solving for ω:

$$\omega = \frac{-\frac{A}{B}c \pm \sqrt{\left(\frac{A}{B}c\right)^2 + 4mk}}{2m} \quad \ldots\ldots [16]$$

Expressing the radical of [16] by means of the binomial

[11] B. Van der Pol, "On Relaxation Oscillations," *Phil. Mag.*, vol. 2 (1926), p. 978.

theorem and neglecting higher-order quantities, [16] reduces to:

$$\omega = \frac{-A}{B} \cdot \frac{c}{2m} + \sqrt{\frac{k}{m}} \quad \text{[17]}$$

In words, the frequency of the self-induced vibration is determined by the timing of the impulses and depends only on the phase between the *fundamental* component of the motion and the *fundamental* component of the impulse.

Equation [17] therefore expresses the deviation from the natural frequency caused by the impulses. In case $A = 0$ from [13], the impulse is proportional to $\cos \omega t$, which, comparing with [12], is seen to be in phase with the velocity dx/dt. In that case [17] shows that the mass vibrates at its natural frequency. In a real case the actual impulse has both A and B components; the A component doing work and not disturbing the frequency, while the B component does not perform work and only modifies the frequency.[12]

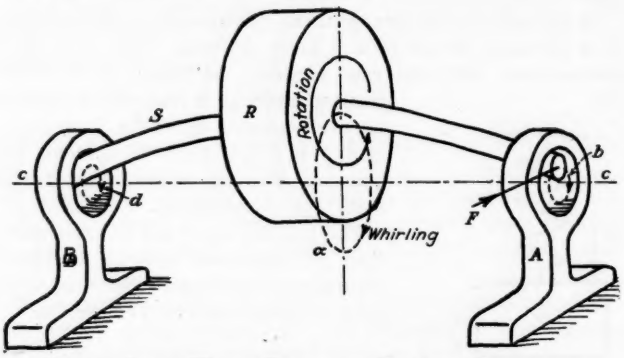


FIG. 5 SHAFT WHIRL CAUSED BY BEARING CLEARANCE

6 APPLICATION TO ACCURATE CLOCKS

The pendulum of a clock is an illustration of the foregoing case. The energy lost in friction is supplied by impulses timed with the position of the pendulum. The difference between the frequency of swing and the natural frequency of the pendulum is given by [17]. This difference is very small and can ordinarily be neglected unless great accuracy is desired. For extreme accuracy it is apparent that the quantity $-\frac{A}{B} \cdot \frac{c}{2m}$ must be kept constant as well as the natural frequency $\sqrt{\frac{k}{m}}$.

7 SHAFT VIBRATIONS DUE TO BEARING CONDITIONS

Kimball and Newkirk[2] have experimented with vibrations occurring at or near double the ordinary critical speed in shafts with generously lubricated bearings. In the phenomenon they describe, the forces furnishing the energy to the vibration come from oil pressures in the bearings. These oil pressures are dependent on the eccentricity of the journal with respect to the bearing. Although their explanation is based on the assumption that the shaft-center displacement in the bearing largely depends on the shaft speed, it is probably also somewhat dependent on the bearing load, and hence on the rotor displacement. If this is the case, the phenomenon is doubtless somewhat self-induced.

Stodola describes a shaft vibration with moderately lubricated bearings caused by a "periodical breaking down of the oil film."[13] A breaking down of the oil film is evidently affected by the load on the bearing, so that with the same reasoning as before, the phenomenon is probably classified correctly as a self-induced vibration.

With poorly lubricated bearings or bearings with considerable clearance, another type of self-induced vibration may be expected, which will now be discussed.

Fig. 5 represents a set-up with two rigid bearings A and B carrying the shaft S, on which is mounted the balanced rotor R. Without distortion, the axis of the shaft is the straight line c-c. With the shaft rotating counterclockwise, imagine the rotor center whirling in the circular path α clockwise. With only a small amount of oil or with large bearing clearances at any instant, the centers of the ends of the shaft will be displaced in the same direction as the center of the rotor, so that the shaft ends will move in the circles b and d, respectively, in a clockwise direction. Due to friction, the bearing will exert on the shaft a moment and a force F acting perpendicularly to the instantaneous displacement of the shaft center and in the direction of its movement. The motion imagined will then have work done upon it equal to $F = 2\pi b$ per cycle of whirling per bearing, where b

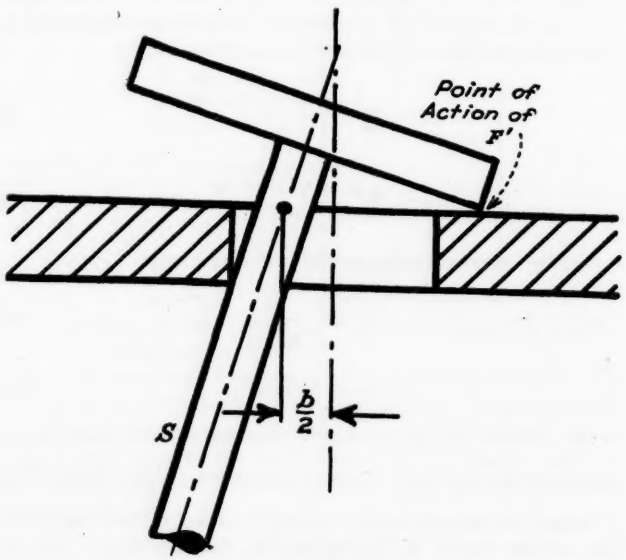


FIG. 6 THRUST BEARING

is the diameter of the circle b, which is approximately equal to the clearance between shaft and bearing. If this energy is sufficient to supply the damping involved in the whirling motion, the latter will usually be maintained. The whirling frequency will be independent of the rotational speed and will be at approximately the natural frequency of the set-up. The whirling may happen at any shaft rotational speed above a certain very low limit. This lower limit is apparently such that the direction of relative motion in the bearing at the contact point between shaft and bearing is zero, due to the whirling action and rotational action, in this respect, just canceling each other. This speed obviously bears the ratio δ/D to the speed corresponding to the natural frequency, which is the ordinary critical speed. δ is the diametral clearance and D is the bearing diameter.

Practically the same phenomenon can also occur in thrust

[12] The amplitude of the motion can be calculated from Equation [15]:

$$S_1 = -\frac{A}{m\omega^2 - k} = \frac{B}{c\omega}$$

or denoting by $F_1 = \sqrt{A^2 + B^2}$ the magnitude of the fundamental component of the impulse, this can be written as:

$$S_1 = \frac{F_1}{\sqrt{(m\omega^2 - k)^2 + c^2\omega^2}} \quad \text{[18]}$$

Expressions for the higher harmonics of the motion S_2, R_2, etc. can be found in a similar manner.

[13] "Steam and Gas Turbines," vol. 1, p. 491, New York, 1927.

bearings (Fig. 6). During the whirling motion, one point of the thrust-bearing runner will have greater pressure than other points, resulting in a greater tangential friction force at that point. If there is at the same time clearance in the journal bearing below, this tangential friction force can drive the shaft in its whirling motion and thus sustain the vibration.

8 Thrust-Bearing Model

The model shown in Figs. 7, 8, and 9 was constructed to study this effect. Dynamically the model was designed to simulate a vertical water-wheel generator installation, in which trouble of this sort was suspected. The rotor R is mounted in the center of the long shaft S (Fig. 7), and carried in the bearings A and B, which are supported by three radial coil springs each. The top coil springs P are held by the mass M, which simulates the inertia of the electric-generator frame. The lower springs Z are attached to the foundation. Both sets of springs serve to allow relative lateral motion between the shaft and the frame, corresponding to the clearance in the guide bearings. The springs U are attached to the bearing B and to a screw I, which can be moved up or down. This arrangement allows a variation in the thrust on the thrust bearing T. The mass M is held in position by three long vertical columns. These columns prevent M from rotation about any horizontal axis and from vertical motion, but offer little resistance to horizontal translation or rotation about a vertical axis. They are intended to simulate the flexibility of the generator frame. The model has two important modes of vibration: one in which the mass centers of M and R are practically stationary and the shaft moves in such a way that its center line describes a cone; and the other in which the rotor moves always in a direction opposite to that of the motion of the stator, both moving in circles (see Fig. 7). The vibration is usually purely in one mode or the other, depending on starting conditions. It can be changed from a given mode to the other by holding the shaft with the hand at an elevation which introduces damping in the existing mode, but not in the other. In any case, the whirling is at constant amplitude and frequency (the latter being a natural frequency), almost entirely independent of the speed of rotation of the shaft, except that the direction of the whirling motion is the same as that of the shaft rotation.

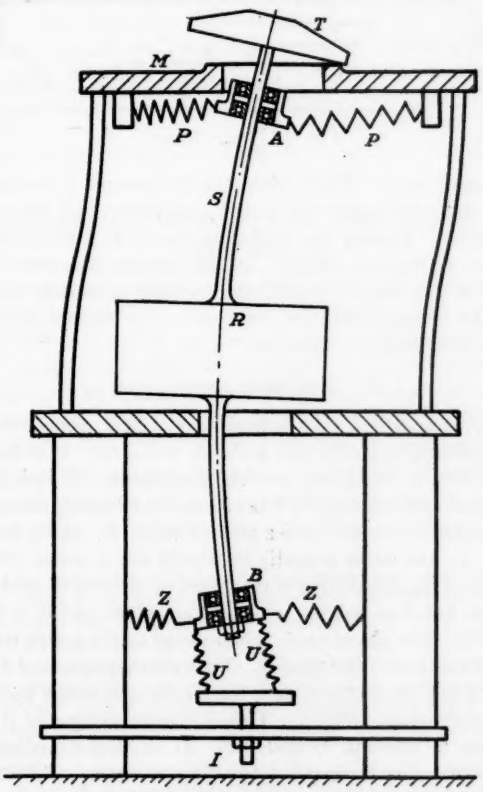


Fig. 7 Schematic View of Model

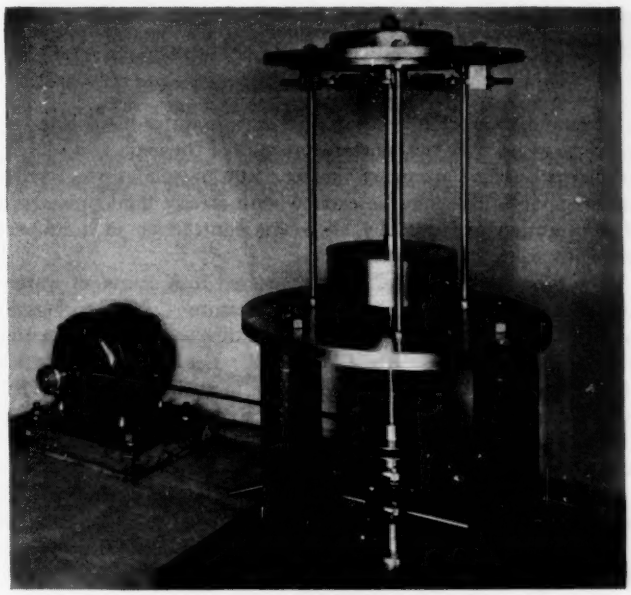


Fig. 8 Model for Demonstrating Vibration Caused by Thrust Bearing

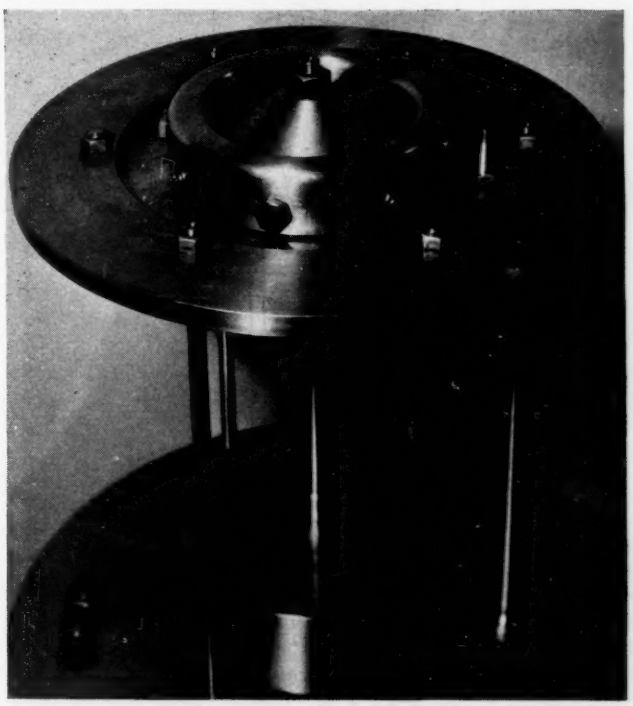


Fig. 9 Detail of Model

The whole phenomenon stops when the thrust on the bearing T is relieved. It was also impossible to make the model vibrate with thrust on the bearing T when it was arranged with an equalizer in such a way that relative angular motion about any horizontal axis between the thrust-bearing runner and stator was prevented. This latter arrangement is shown in Fig. 9. The

pins, of which one is removed, prevent the equalizer from functioning when in place.

9 Periodic Flow Phenomena

It is well known that violent vibrations in water mains can be produced with only the constant pressure of the main as a source of energy. There are many similar conditions under which self-induced vibrations may be excited with steady fluid pressures. The vibration produced by an elastic contraction in a rubber water tube is a good illustration.

The tube shown in Fig. 10 is connected to a source of water pressure p_1 and discharges at constant pressure p_3. The springs S and T force the cylinders A and B on the tube to furnish an elastic contraction that will be denoted by AB. Let d be the variable dimension of AB, and p_2 the pressure immediately below AB. The spring forces are so adjusted that with $d = 0$ and $p_2 = p_3$, there is a slight tendency for d to increase—i.e., for AB to open a small amount.

Assume p_1 is constant. This condition may be realized either by the use of a large container immediately above AB as a source of pressure, or if the elasticity of the tube above AB prevents pressure changes due to flow variation.

Taking as the starting condition $p_2 = p_3$, zero velocity of the water column below AB, and $d = 0$, d will of course immediately increase, making $p_1 = p_2$ and $p_2 > p_3$, with consequent further

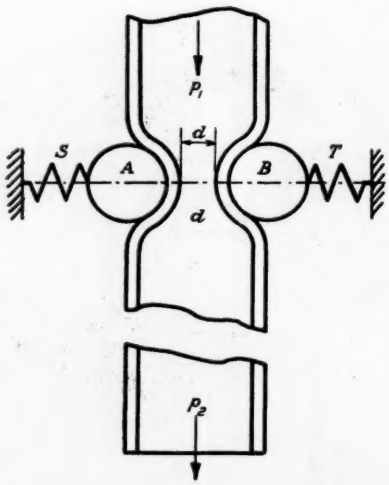


Fig. 10 Vibrating Water Column in Rubber Tube

increase of d and acceleration of the column of water below AB. When the velocity at the outlet increases to nearly that corresponding to p_1, the acceleration of the water column below AB disappears, and p_2 becomes less than p_3 on account of higher velocity at the contraction than at the outlet, so that the d immediately becomes zero. This causes deceleration of the water column below the contraction, which results in p_2 remaining less than p_3 and d remaining zero. The water flow now continues outward until sufficient time elapses for the difference $p_3 - p_2$ to stop the water column and reverse it. It has been found experimentally that p_2 easily becomes small enough to cause collapse of the tube walls. The reverse flow soon builds up the pressure p_2 to the value where $p_2 = p_3$, and the cycle repeats.

Evidently the time required to complete a cycle, as described, depends primarily on the time necessary to start and stop the column of water below AB. In view of this, the experimental fact that the period of such pulsations increases with increase in the water column length below AB is understood.

10 Air Horn

The operation of an air horn is another good illustration of self-induced vibrations of flow. An air pressure p_1 is applied to the pipe Q of the horn (Fig. 11). The air is throttled through the orifice A into the annular chamber C, where the pressure is p_2. The diaphragm D acts as a valve on the seat S. For simplicity of explanation assume that the pressure p_2 varies by an amount which is small compared to either $(p_1 - p_2)$ or $(p_2 - p_3)$, p_3 being atmospheric pressure. With this assumption, the flow through A is constant and the rate of flow into the horn from C depends only on the opening between D and its seat S. With D vibrating, the pressure p_2 will depend on the amount of air present in C. This amount at any instant depends on the history of the

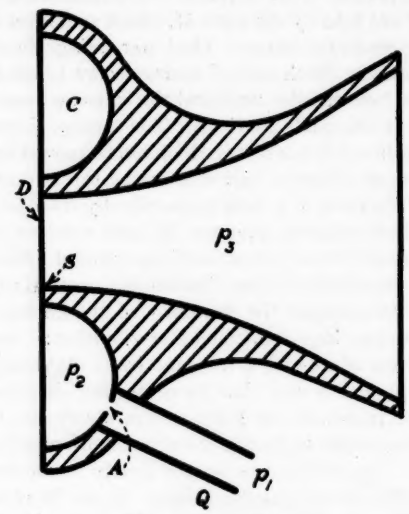


Fig. 11 Air Horn

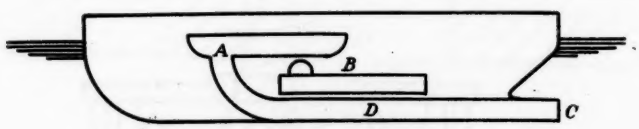


Fig. 12 Toy Boat

displacement of D. Thus while the diaphragm is moving outward, it just previously was closed completely, and the pressure in C is great. During the closing motion of D it previously was wide open, so that p_2 is small. In this manner it is seen that the pressure in C is greater during the opening operation of D than during the closing operation, so that a net amount of work is available to maintain a vibration.

11 Toy-Boat Propulsion

A year or so ago a very ingenious toy boat appeared in the stores, a photograph of which is shown in Fig. 1. It is described here because it furnishes another illustration of periodic-flow self-induced vibrations as well as of other interesting phenomena.

The source of energy is the alcohol lamp B, which heats the chamber A; the latter is partly or wholly full of water (Fig. 12). The top of A is a diaphragm constructed of bimetal in such a way that when hot it is bulging upward and when cold it is bulging downward. The chamber A is connected to the water astern of the boat by means of the tube D. With the diaphragm of A down, B heats A and its contents until the diaphragm snaps upward to its hot equilibrium position. This change in volume of A draws cold water in through D and cools A, so that the diaphragm snaps down again, forcing out the surplus water. With further heating, the cycle repeats several hundred times per minute.

During the time water is flowing *out*, it forms a "jet" issuing from C, which pushes the boat forward by its reaction. When the water is going *in*, however, the point C is a "sink;" the water is attracted to it from all sides, so that no retarding reaction is felt by the boat.[14] In total, therefore, there is a forward driving force during a full cycle. It is seen that the steady source of energy of the flame is inverted into the alternating flow, which in turn is rectified hydrodynamically into a unidirectional effect, driving the boat forward.

12 Transmission-Line Vibration Due to Wind and Sleet

There are two types of transmission-line vibrations of importance caused by a transverse wind: (1) Vibrations in moderate winds at rather high frequency due to "Kármán" vortices generated on the lee side of the wire. This phenomenon is to some extent self-induced in that the vibrating motion of the wire alters the flow in such a manner that vibrations may be sustained at frequencies considerably different from the Kármán frequency as determined without vibration. (2) Vibrations at very low frequency when the wires are loaded with sleet. This phenomenon, which is a typical self-induced vibration, has been explained by Den Hartog.[15] The explanation is briefly as follows: The resultant force of a wind on a round wire is always in the direction of the wind. On the other hand, when the cross-section of the wire is unsymmetrical, as may occur with sleet loads, the resultant force in general is not in the direction of the wind. For example, in Fig. 13, a rectangular section is shown. With a wind blowing in the direction a-a, the resultant wind force will be F_a. However, if the wind blows in the direction b-b, the resultant force will be F_b. If the wind is blowing in the direction a-a and the wire is given a velocity upward, the wind relative to the conductor will be in a direction b-b, and, as before, the resultant force will be F_d, which has an upward component. Thus, on account of the upward motion, an upward force is brought into play. Any vertical movement will be encouraged, so that self-induced vibrations will take place if the damping is not too great. Many sections do not have this characteristic; e.g., for a wind in the direction b-b the resultant force F_b might have a downward component, in which case vibration would be damped by the wind. Den Hartog gives a simple experimental means for testing this effect and an analysis whereby the stability of any cross-section can be determined from its aerodynamic drag and lift curves.

13 Self-Induced Vibrations From Fixed High-Frequency Disturbances

In all examples discussed so far, the source of energy maintaining the vibrations was invariably unidirectional or steady. There exists, however, another group of phenomena whereby a vibration approximately at a natural frequency may be self-sustained by an energy source which is alternating with a frequency mostly much higher than the frequency of the self-induced vibration. To be sure, this does not take place in purely linear systems; some pronounced non-linearity is necessary.

Typical examples of this type are shown in Fig. 14, (a) and (b). In Fig. 14(a), the magnet B is excited by a constant-frequency alternating current. The iron block A is mounted so that it may oscillate freely. If the motion of A is small compared to the total air gap, the force between A and B will be proportional to the square of the current in B and independent of the position of A. With a current frequency f, the frequency of the force would be $2f$. On the other hand, if the displacement of A is large with respect to the total air gap, the magnetic force depends a great deal on the position of A.

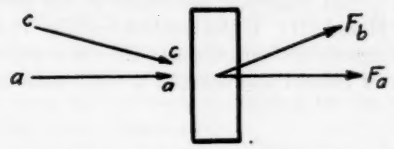


Fig. 13 Forces on Vibrating Transmission Line

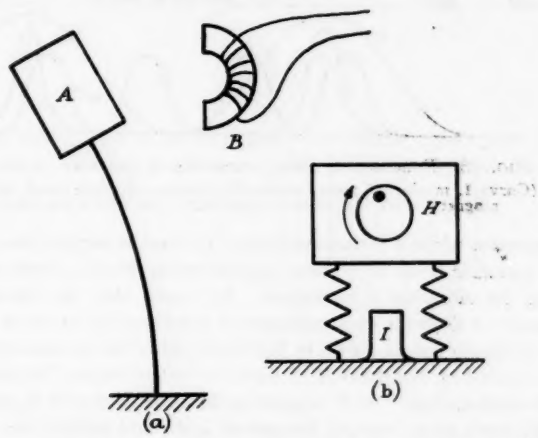


Fig. 14 Two Cases of Vibration Caused by High-Frequency Disturbance

Assume that the vibration of A is steady, represented by:

$$x = E \sin \omega t + F \cos \omega t + E_2 \sin 2\omega t + F_2 \cos 2\omega t \ldots E_n \sin n\omega t + F_n \cos nt$$

Let the periodic force on A due to the magnet B be $G \sin \gamma t$ when x is held constant. The magnetic force P at any instant will then be:

$$P = G \sin \gamma t \cdot f(x)$$

where $f(x)$ means a function of x. Since x is a periodic function, $f(x)$ will also be a periodic function, the lowest frequency component being the same as the frequency $\omega/2\pi$ of x. Therefore [19] may be written:

$$P = G \sin \gamma t[C_1 \sin \omega t + D_1 \cos \omega t + C_2 \sin 2\omega t + D_2 \cos 2\omega t \ldots + C_n \sin n\omega t + D_n \cos n\omega t] \quad \ldots\ldots\ldots [20]$$

G, C_1, D_1, $\ldots$ C_n, D_n, etc., are constants. Equation [20] can be changed algebraically to a sum of terms of the form $F \sin (\gamma \pm n\omega)t$.

In order to furnish the losses of the vibration, there must be an energy-supplying force on A of the same frequency as x; hence ω must be such that

$$(\gamma \pm n\omega) = \omega$$

or

$$\gamma = (1 \pm n)\omega \quad \ldots\ldots\ldots\ldots\ldots [21]$$

This indicates that γ cannot be less than and must be an even multiple of ω.

Satisfying [21] results in the magnetic force having the proper

[14] It is an experimental fact that the pressure of water issuing from a tube under water is the static pressure of the water at the level of the tube and that the pressure of water being sucked in under similar conditions is less than the static pressures by an amount corresponding to the velocity with which it is sucked in. This difference is well known to be the effect of the viscosity of the water at the surfaces of the tube. A brief study of the equilibrium of the water in the boat shows that the foregoing pressure cycle results in an average forward force on the boat hull.

[15] "Transmission-Line Vibration Due to Sleet," paper presented at the A.I.E.E. Summer Convention, Cleveland, June 20, 1932.

frequency to do work on the motion of A. This in itself, however, it not sufficient. The phase and stability requirements must be fulfilled if the motion is to be sustained. Physically the phenomenon can be understood by studying Fig. 15. Curve I indicates the motion of A; curve II is the magnetic force which this motion would cause if the magnet were excited by a constant current; curve III indicates the magnetic force variations due to the alternating current for a constant position x of A. Clearly the actual magnetic attraction is the product of the ordinates of II and III. If the condition of Fig. 15 is satisfied—i.e., if the magnetic-attraction curve III has a negative slope at the moment of nearest approach, it is clear that the attraction

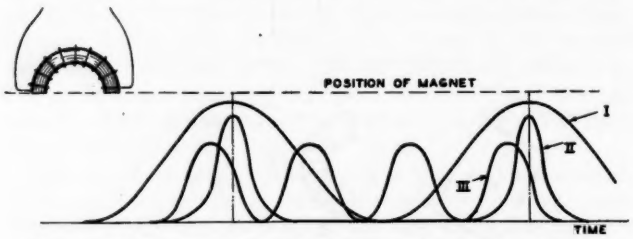


FIG. 15 FORCES AND DISPLACEMENTS IN SYSTEMS OF FIG. 14
(Curve I, motion of mass; curve II, force on moving mass with d-c magnet; curve III, force on stationary mass with a-c magnet.)

is greater while A is moving toward B than during the time that A is receding from B. Then, consequently, work is done upon A and its vibration is sustained. It is seen that the whole phenomenon depends on a pronounced non-linearity in curve II.

A similar case is shown in Fig. 14(b), where an unbalanced rotor is rotated on the mass H at high above the natural frequency of H on its springs. If H is given an impulse so that it moves back and forth at its natural frequency $\omega/2\pi$ and strikes the stud I, it will receive impulses that can be represented by [20], and hence the same conclusions can be drawn as before.[16] The details of the impact on the stud must be such that in the first half of it the force on H caused by the rotating unbalance tends to drive H into the stud, whereas during the other, receding, half of the impact, the unbalance force acts in the direction away from the stud. In this manner it is clear that energy is put into the slow vibration by means of the fast rotation.

This knocking phenomenon probably occurs often in high-speed machinery if there is any lost motion. In this way a vibration near a natural frequency may be maintained by a rotation far above this natural frequency.

14 CONCLUSIONS

1 Self-induced vibrations in linear systems cannot be of constant amplitude (except in trivial cases); for steady finite amplitudes some non-linearity in the system is required.

2 For linear as well as for non-linear systems, the possibility of self-induced vibration can be determined by calculation of the small vibrations of the system, the criterion being a positive exponent of the e-functions appearing in the solution. This is explained in section 2.

3 While most cases of self-induced vibration are caused by steady sources of energy, another type of self-sustained motion can occur which is kept up by an alternating-energy source of high frequency. Two cases of this type are discussed in section 13.

Appendix

SOLUTION OF EQUATIONS [7]

The equations in question are:

$$\left.\begin{aligned} m\frac{d^2x}{dt^2} + kx + Dy = 0 \\ m\frac{d^2y}{dt^2} + ky - Dx = 0 \end{aligned}\right\} \quad [7]$$

Eliminating x by differentiation, we obtain:

$$\frac{m^2}{D}\frac{d^4y}{dt^4} + \frac{2mk}{D}\frac{d^2y}{dt^2} + \left(\frac{k^2}{D} + D\right)y = 0 \quad [22]$$

Assuming $y = e^{rt}$ substituting in [22], and canceling out the exponentials:

$$r^4 + \frac{2k}{m}r^2 + \frac{k^2 + D^2}{m} = 0 \quad [23]$$

$$r = \pm\sqrt{\frac{-k \pm Di}{m}}$$

Assume:

$$+\sqrt{\frac{-k + Di}{m}} = n + qi \quad [24]$$

Squaring [24]:

$$-\frac{k}{m} + \frac{Di}{m} = n^2 + 2nqi - q^2$$

Equating real parts and imaginary parts:

$$\frac{k}{m} = q^2 - n^2$$

$$\frac{D}{m} = 2nq$$

Solving these for n and q and remembering that they are real:

$$\left.\begin{aligned} n &= +\sqrt{\frac{-k + \sqrt{k^2 + D^2}}{2m}} \\ q &= +\sqrt{\frac{k + \sqrt{k^2 + D^2}}{2m}} \end{aligned}\right\} \quad [25]$$

Similarly it may be shown that

$$\sqrt{\frac{-k - Di}{m}} = n - qi$$

The solution for y becomes:

$$y = Ae^{(n+qi)t} + Be^{-(n+qi)t} + Ce^{(n-qi)t} + De^{-(n-qi)t}$$

which may, if n and q are real, be written:

$$y = e^{nt}[C_1 \cos qt + C_2 \sin qt] + e^{-nt}[C_3 \cos qt + C_4 \sin qt] \quad [8a]$$

The solution for x, obtained by substitution of the expression [8a] for y in [7], is:

$$\alpha = e^{nt}[C_1 \sin qt - C_2 \cos qt] - e^{-nt}[C_3 \sin qt - C_4 \cos qt] \quad [8b]$$

Discussion

M. STONE.[17] The writing of a paper on self-induced vibration is a commendable undertaking, for it calls to the attention of engineers not steeped in the subject of vibration that there exists a large and important class of vibratory phenomena that cannot be treated under the simple category of motion stimulated by periodically varying forces, do not display what is ordinarily called resonance, and so forth. The writer has been interested in several of the cases discussed in detail by the author, such as shaft whirl caused by internal friction, shaft vibration due to bearing conditions, self-induced vibrations from fixed high-frequency disturbances, etc., for they have had to do with work falling under his jurisdiction, and he wishes to endorse the treatment that the author has given.

With reference to the general relation, under section 2 of the paper, the writer would like to add an important property of the set of Equations [1] that distinguishes them as possessing

[16] Curve II in Fig. 15 for this case represents the impact force, and naturally consists of sharp peaks even more pronounced than indicated in the figure.

[17] Engineer-in-charge of Development, Power Engineering Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

the property of self-induced vibration or not. In a paper[18] of the writer, it is shown that if $m_{nm} = m_{mn}$, $c_{nm} = c_{mn}$, and $k_{nm} = k_{mn}$, then the system is *not* capable of self-induced vibration. It is necessary that at least one or more of these quantities be unequal to have the system exhibit this property. This fact was elucidated in connection with the problem referred to by the author,[19] and is briefly described here as another case of self-induced vibration of a somewhat different type.

Consider the schematically sketched generator-governor system shown in Fig. 1. Consider the generator system for the moment; the differential equation of free oscillation is:

$$I\ddot{\theta} + c_s\dot{\theta} + k_s\theta = 0 \quad \ldots\ldots\ldots\ldots\ldots [26]$$

where:

I = total mass moment of inertia of the generator system—i.e., engine and flywheel and generator
θ = angular displacement from synchronous position
c_s = specific damping torque, as determined mainly by damper winding design
k_s = synchronizing torque of the generator, by virtue of its parallel operation with the network.

From a consideration of governor dynamics, it can be shown that it will vibrate freely according to:

$$m\ddot{x} + c_g\dot{x} + k_g x = 0 \quad \ldots\ldots\ldots\ldots\ldots [27]$$

where:

m = equivalent mass of governor, referred to translational motions of the sleeve
x = translational displacement of sleeve from equilibrium
c_g = damping constant in governor system, mainly provided by dashpot shown schematically
k_g = elastic restoring force/unit displacement of sleeve; this is a function of shaft speed.

Now by virtue of the way a governor and generator are interconnected for proper governing, Equations [26] and [27] are modified to give:

$$\left.\begin{aligned} I\ddot{\theta} + c_s\dot{\theta} + k_s\theta &= kx \\ m\ddot{x} + c_g\dot{x} + k_g x &= -c\dot{\theta} \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [28]$$

as is shown in the paper referred to. It is seen here that $k = kg$ and $c_s = c$, referring to the remark made earlier concerning the necessity for the inequality of coefficients, and so the system is capable of self-induced vibration. The solution of [28] is carried out in the paper, the condition of instability being established by studying the sign of the exponent, as mentioned by the author.

In connection with this problem, it has been possible to disclose an exact analog between two coupled vacuum-tube circuits and the oscillatory generator-governor system,[18] thus providing an easy way of experimentally studying the effect of various magnitudes and the conditions of instability.

[18] "Non-unilateral Vibrations," M. Stone, before the Franklin Institute.

[19] "Parallel Operation of A-C Generators, Action of Governors and Damper Windings," M. Stone, before A.I.E.E.

A. L. KIMBALL.[20] This paper is of particular interest to the writer because it takes up the phenomenon of the self-induced whirling of a rotor when running above its critical speed caused by internal friction within the rotor or shaft. This phenomenon was first discussed by the writer before the American Physical Society at its Washington meeting in 1923, at which a small model was exhibited.

There are several publications on this subject not cited by the author to which attention should be called.[21]

Referring to the author's analysis in which he derives the outward logarithmic-spiral motion of the center of mass of the rotor, the same result is more directly obtained by the use of polar coordinates using the standard expressions for the vectorial and angular accelerations. These are:

$$a_r = \frac{d^2r}{dt^2} - r\left(\frac{d\theta}{dt}\right)^2$$

$$a\theta = r\frac{d^2\theta}{dt^2} + 2\frac{dr}{dt}\frac{d\theta}{dt}$$

Since the elastic force acts along the radius vector and the disturbing force transverse to it, the following pair of simultaneous differential equations are obtained:

$$m\frac{d^2r}{dt^2} - mr\left(\frac{d\theta}{dt}\right)^2 = -kr$$

$$mr\frac{d^2\theta}{dt^2} + 2m\frac{dr}{dt}\frac{d\theta}{dt} = Dr$$

Assuming the solution $r = r_0\varepsilon^{nt}$, there is obtained at once:

$$\frac{d\theta}{dt} = \sqrt{\left(\frac{mn^2 + k}{m}\right)} = \frac{D}{2mn}$$

from which $n = \sqrt{\left(\frac{-k + \sqrt{(k^2 + D^2)}}{2m}\right)}$, as obtained by the author.

The disturbing force responsible for this type of whirl can be pictured as a direct result of frictional stress components set up within the shaft along with the elastic stresses. This viewpoint is presented in the article in the *Philosophical Magazine*.[21]

The writer wishes to mention that the internal friction in an ordinary steel shaft is very small, and other frictional effects produced by cramp fits may be several times larger.

An experimental study of the relative amounts of these two types of friction has been made by the writer and is presented by him in an article entitled "Measurement of Internal Friction in a Revolving Reflected Shaft," *Genl. Elec. Rev.*, August, 1925.

[20] Research Engineer, General Electric Company, Schenectady, N. Y. Assoc. A.S.M.E.

[21] "Internal Friction Theory of Shaft Whirling," A. L. Kimball, abstract in *Phys. Rev.*, June, 1923. Same title, A. L. Kimball, *Genl. Elec. Rev.*, April, 1924. "Shaft Whipping," B. L. Newkirk, *Genl. Elec. Rev.*, March, 1924. "Internal Friction as a Cause of Shaft Whirling," A. L. Kimball, *Phil. Mag.*, April, 1925.

APM-55-3

The Fatigue of Metals—Its Nature and Significance

By H. F. MOORE,[1] URBANA, ILL.

The essential feature of "fatigue" failure in metals is a spreading crack. Studies of fatigue failure in single crystals of metal indicate that such cracks frequently start by slip along planes of atomic weakness. The study of fatigue failure of metals has shown certain limitations of the mathematical theory of elasticity when applied to actual non-homogeneous metals. The importance of the ability of a metal to resist occasional slight plastic action without starting a fatigue crack is emphasized. The relation between atomic strength and the strength of a multi-grained piece of metal is briefly discussed and a "crack-interference" hypothesis of fatigue failure suggested. The phenomenon of spreading cracks in metal under steady load at elevated temperatures is noted, and its significance discussed.



THE phenomenon of the failure of metallic structural and machine parts under repeated stress has been extensively studied for some three-quarters of a century, and has been studied with special intensity since the development of the automobile and the airplane. This phenomenon is commonly, but rather inaccurately, called the "fatigue" of metals. All recent studies indicate that the origin of this so-called "fatigue" is in a crack in the material which, under repeated stress, spreads and finally so weakens the machine or structural part that sudden fracture occurs. Just how such a crack originates, and under what conditions a minute crack develops into complete fracture of a member, are matters concerning which our present-day knowledge is far from satisfactory.

The study of the fatigue failure of single-crystal specimens of metals by Gough and others[2] indicates that for most metals, perhaps for all metals, a fatigue crack originates in a plane of atomic weakness along which "slip" (failure of elastic strength) has occurred, possibly by a wrinkling action along the plane of slip under the cycles of repeated stress. This wrinkling action tends to break atomic bonds at the "wrinkles." The first slip planes are along planes of atomic weakness rather than along planes of maximum stress or maximum strain, but after a crack once starts spreading to fracture, its course seems to be determined largely by the direction of maximum tensile strain. In single-crystal specimens of certain metals with close-packed hexagonal space-lattice[3] structure, observers have been unable to detect any slip before detecting a crack, while for several metals with body-centered or face-centered space-lattice structure considerable slip has been detected, although no fatigue fracture had occurred nor any crack been detected even under 100,000,000 cycles of reversed stress. "Just when and how does a fatigue crack start?" is still an unanswered question.

Mathematical Theory of Elasticity Statistically Rather Than Exactly True

One result of the study of fatigue phenomena in metals is the growing conviction that the whole mathematical theory of elasticity, on which all our computations of stress and strain are based, must be regarded as statistically true rather than exactly true when applied to the behavior of actual metals under stress. This theory is founded on certain postulates, among them the following: metals are (1) homogeneous, (2) isotropic, and (3) indefinitely divisible without change of elastic properties. One glance through a metallurgical microscope at a polished and etched piece of metal raises grave doubts as to the rigid truth of these assumptions, and the phenomena of fatigue failure strengthens this impression.

The formulas for computing the strength of materials were developed by and for structural engineers, and they have been most interested, not in spreading fatigue cracks, but in the failure of elastic strength or in fracture under a few repetitions of load. Moreover, localized failure in a few crystalline grains of metal does not, in general, mean any appreciable damage to beams, columns, and tie rods in a structure. So the formulas in common use, which may be regarded as "statistically true" for an aggregation of crystalline grains, serve the structural engineer very well indeed.

However, when machine parts are considered, the importance of localized failure, leading to a spreading crack under loads repeated hundreds of thousands of times, becomes vitally important, and the limitations of the mathematical theory, as applied to actual metals, become apparent. For example, the mathematical theory of elasticity indicates that the localized stress at a notch, a screw thread, or a small hole depends not at all on the material, but only on the shape of the piece. Actually metals differ widely in the ratio between the load which, if oft repeated, causes failure in a piece containing a notch or screw thread and the load which will cause failure in a similar piece of the same metal in which the notch has been smoothed out or the screw thread turned off. The abilities of various metals to adjust themselves to high localized stress without starting a fatigue crack differ widely, and the "crackless plasticity" which allows such adjustment is not a simple function, either of static strength, fatigue strength (of smooth specimens), ductility as indicated by elongation and reduction of area, or of the results of impact tests on notched specimens. This property is important in machine parts, but as yet no reliable method of measuring it has been devised.

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[1] Research Professor of Engineering Materials, University of Illinois. Mem. A.S.M.E. Professor Moore was educated at the University of New Hampshire and Cornell University. He studied under Prof. Albert Kingsbury and Prof. John H. Barr. He has been connected with the Riehlé Brothers Testing Machine Company and the Materials Testing Laboratories of the University of Wisconsin and the University of Illinois. For the past twenty-five years he has been on the staff of the Engineering Experiment Station of the University of Illinois.

[2] For Gough's work on single-crystal specimens see Trans. Roy. Soc. (A), vol. 226 (1926), p. 1; Proc. Roy. Soc. (A), vol. 118 (1928), p. 498, vol. 123 (1929), p. 143, and vol. 127 (1930), p. 431.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., Dec. 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] Space lattice is the pattern of orderly arrangement of atoms in a crystalline grain of metal.

In all cases so far studied, although the mathematical theory of elasticity cannot be regarded as rigidly accurate when used to determine significant localized stresses in actual metals, it does give results *on the safe side*, and hence to the engineer it is still a vitally important theory. Probably the machine designer of the near future will have to become more familiar with its formulas than are the present-day designers, and, in addition, will find it of great advantage to become familiar with the theory of plasticity as developed by Nadai and others.[4]

Analogy Between Spread of a Fatigue Crack and Spread of Bacterial Infection

Analogies in scientific discussions are sometimes illuminating and always dangerous. With this word of warning the author wishes to present what some may regard as a whimsical analogy between the spread of a fatigue crack in metal and the spread of bacterial infection. It is probable that all of us carry around in our bodies the germs of several deadly diseases; probably all metallic machine or structural parts have in them many incipient cracks and weak spots. Under stress of chill or overexertion the normal resistance of our bodies to the spread of bacterial infection may be so lowered that such spread may go on to the point of causing serious illness, or even death; under mechanical

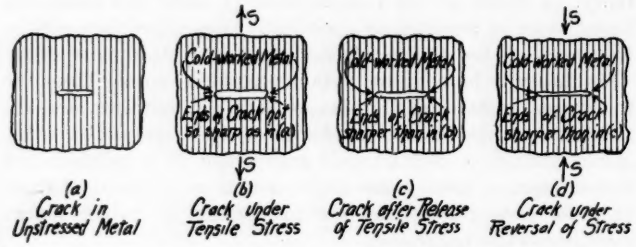


Fig. 1 Crack in Metal During Cycle of Stress

stress the resistance of a metal to the spread of an incipient crack may be so lowered that fatigue failure results. In the body the spread of bacterial infection calls up reserves of resistance which get into action to stop the spread of infection, frequently involving rise of body temperature, and sometimes involving overload on the heart or other organs; in most metals when a crack starts spreading there develop reserves of resistance in the shape of surrounding metal *strengthened* by slip (cold work), sometimes with rise of temperature, and sometimes with transfer of stress to adjacent parts of the metal. In both the human body and the metal there seems to be a conflict between beneficial and injurious influences with the result depending on the preponderance of one over the other.

Once again may the author call attention to the fact that the foregoing is an analogy, not a piece of rigid reasoning.

What is the relation between atomic cohesion and the fracture of a specimen or a machine part? Theoretical considerations, based on latent-heat values, indicate that the force necessary to actually break atomic bonds corresponds to a tensile strength 10 to 20 times that developed in static tension tests and 20 to 40 times that developed under cycles of reversed stress. On the other hand, tests of single-crystal specimens give an extremely low value of strength under repeated stress. Mechanical fracture seems to start either in minute areas of extremely high localized stress or to originate from the repeated sliding displacement of layers of atoms, with the resultant development of cracks.

In the multi-grained metal of machine parts the random orientation of crystalline grains may well set up barriers to the spread of cracks from unfavorably oriented grains to favorably oriented ones. If in the metal there are two or more kinds of grain, differing in strength, this barrier action may well be even more marked. A "crack-interference" theory of fatigue failure, similar in its general lines to the well-known "slip-interference" theory of Jeffries and Archer, seems worthy of consideration.

Why a Crack Spreads Under Repeated Cycles of Stress and Not Under a Steady Load

But why does a crack spread under repeated cycles of stress and not under a steady load? In the first place, it may be well to note that there is evidence that, under certain conditions of temperature, spreading cracks which differ markedly from fatigue cracks may sometimes occur under long-continued steady load. These cracks will be discussed in a succeeding paragraph.

Under a steady load, if an incipient crack is present in the metal, the crack probably spreads somewhat, and opens up with a rounding off of the ends. This is illustrated in Fig. 1 at (*a*) and (*b*). The spreading of the crack weakens the metal, but the rounding off of the ends lowers the stress concentration, and also cold-works and strengthens the metal. Thus there are both destructive and beneficial actions present, and, unless the load per square inch is very great, so great as to cause a rather general plastic flow of the metal, equilibrium is reached before complete fracture occurs. Even a very small amount of generally distributed plastic flow might well serve to prevent the spreading of a crack to complete fracture.

Now if the load is removed, the crack closes up and its ends become somewhat sharper than under load, as illustrated at (*c*), Fig. 1. Then if load is again applied the stress concentration at the ends of the crack is greater than it was when first under load, and may be either greater or smaller than it was at the ends of the incipient crack in the virgin metal. Moreover, at the ends of the crack, where the metal is strengthened by the cold-working it has received, its "crackless plasticity" is lowered, and it is less able to make crackless-plastic adjustments under succeeding cycles of stress. We do not believe that metal "crystallizes" under repeated stress, but it changes its properties, and then metal after many cycles of repeated stress may be either improved or damaged according as the beneficial or the injurious effects of the stress predominate. At the intensity of computed stress which we denote the endurance limit, the two influences just balance.

If the metal is subjected to cycles of *reversed* loading, the flattening of the crack and the sharpening of the ends are magnified, and with this action the stress intensification at the ends of the cracks is increased, so that the balance between beneficial and destructive effects takes place at a lower computed stress. This is illustrated at (*d*), Fig. 1.

Ductile Metal Does Not Resist Repeated Stress Better Than Brittle Metal

It is a very natural idea that a ductile metal will resist repeated stress better than a brittle metal, but the actual facts do not bear out this simple assumption. Probably some slight plastic action is involved whenever a machine part is subjected to repeated stress, especially if there are notches, screw threads, holes, or other "stress raisers" in it, but even the metals rated as brittle—cast iron, for example—can withstand some slight plastic action without starting a fatigue crack. For metal parts rarely subjected to severe plastic action—springs, for example—strength seems far more important than ductility. However, for many machine and structural parts, such as railway equipment, pressure vessels, bolts, etc., occasional severe localized stress is frequently developed, and the ability to resist such occasional plastic action without starting a destructive spreading crack is of great importance.

[4] Nadai, A., "Plasticity," McGraw-Hill Book Company, New York, 1931.

General ductility under a few loads, and the ability to withstand a few thousand cycles of stress involving localized plastic action, seem to be quite different properties. Under one load, or a few dozen loads, the ductility of the metal *as a whole* is called into play, and any incipient cracks which may form have very little chance of spreading to fracture. Under thousands of cycles of stress, *local* exhaustion of ductility under repeated plastic action tends to cause a crack to spread, as has been discussed in a preceding paragraph. Metals may show a good degree of ductility under a static tension test—may show good elongation and reduction of area—and yet may crack readily under thousands of cycles of repeated stress.

Fig. 2 shows micrographs of fatigue cracks in structural steel and in heat-treated alloy steel. The structural steel gave 38.8 per cent elongation in 2 in. and 63.9 per cent reduction of area; the chrome-nickel steel gave 24.2 per cent elongation in 2 in. and 6.87 per cent reduction of area. In the micrograph of the structural steel is seen a distinct "disturbed area" adjacent to the fatigue crack. Slip lines are seen several grains away from the crack, and evidently considerable energy has been absorbed in causing this relatively widespread plastic action. On the other hand, in the micrograph of the heat-treated alloy steel no such "disturbed area" can be detected, the general ductility of the metal does not seem to have been called into play, and much more of the energy of the cycles of repeated stress is available for starting and spreading a fatigue crack.



Structural Steel (Magnification, 200 ×)

Chrome-Nickel Steel (Magnification, 650 ×)

FIG. 2 MICROGRAPHS OF STRUCTURE NEAR FATIGUE CRACKS
(Traced from photomicrographs made by S. W. Lyon in the Metallographic Laboratories, University of Illinois.)

The terms "body," "dynamic ductility," and "crackless plasticity" have been suggested to denote this ability to develop localized ductility under repeated stress, and it is the author's opinion that the development of a practical test for this quality is a field of research offering no small promise of usefulness. Comparative fatigue tests of notched and unnotched specimens, comparative impact tests of notched and unnotched specimens, and damping tests in which the rate of dying out of vibration is measured[5,6] have been suggested, but none of them have proved their useful reliability as yet.

ENDURANCE LIMIT AN IMPORTANT PHYSICAL PROPERTY OF A METAL

From the great mass of fatigue-test data which has accumulated during the past twelve years has come the conclusion that the endurance limit(or fatigue limit)—the limiting stress below which unnotched specimens will withstand an indefinitely large number of cycles of stress—is an important physical property of a metal, and that it is a good index of the ability of the metal to withstand millions of cycles of repeated working stress, if there are very few occasional cycles of stress involving appreciable localized plastic action. Fatigue testing is so time-consuming, and the cost of specimens is so high, that it has not found a place in practical acceptance tests. So far no reliable short-time test for fatigue limit has been developed, but research for such a test is still in progress.

Much work remains to be done to determine the fatigue strength of metals under various ranges of stress, under cycles of shearing stress, and under combinations of steady shearing and varying flexural stress. A discussion of this subject would extend beyond the limits of this paper, and reference may be made to the article by Prof. J. B. Kommers in the 1930 report of the American Society for Testing Materials Research Committee on Fatigue of Metals.

Similarly, the very important subject of "corrosion fatigue" cannot be discussed in any detail in this paper, and reference may be made to the 1932 report of the American Society for Testing Materials Research Committee on Fatigue of Metals, to the papers of Dr. D. J. McAdam published in the Proceedings of the American Society for Testing Materials from 1926 to 1931, inclusive, to the paper presented by him before the 1932 Congress of the International Congress for Testing Materials at Zurich, and to the 1932 autumn lecture by Dr. H. J. Gough, of the British National Physical Laboratory, before the British Institute of Metals.

The formation and spread of cracks under long-continued static load has been noticed by investigators of the behavior of metals at high temperatures,[7] and some evidence of such cracks in lead under long-continued static load at room temperatures has been observed in the Materials Testing Laboratory of the University of Illinois. In steel and in lead these cracks seem to be intercrystalline rather than transcrystalline, as is the case with ordinary fatigue cracks. The study of the occurrence of such cracks under static load is an almost unexplored field of investigation, and one which seems very promising.

Discussion

R. E. PETERSON.[8] Professor Moore has given us a most valuable bird's-eye view of the entire field of fatigue of metals—a view which should be kept in mind in incorporating the detailed phases of the subject into the general scheme of working stresses.

In connection with Professor Moore's remarks concerning the origin of fatigue failure and the detection of the first discernible cracking, it may be of interest to mention some experiments

[5] Föppl, O., "Bedeutung der Dämpfungsfähigkeit," *Zeit. für Metallwirtschaft*, heft 18 (1929), p. 419.

[6] Heydekampf, G. S., "Damping Capacity of Materials," Proc. Am. Soc. Test. Matls., vol. 31 (1931), part 1, p. 157.

[7] Hanson, D., and Wheeler, M. A., "The Deformation of Metals Under Prolonged Loading," Journal (British) Institute of Metals, vol. xlv (1931), no. 1, p. 229.

[8] Manager, Mechanics Division, Westinghouse Research Laboratories, East Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

made by Davidenkoff.[9] Fifteen identical notched fatigue specimens were made of 0.2 per cent C steel. All specimens were run at the same load (a load causing fatigue fracture at 725,000 cycles), but were stopped and removed from the fatigue machine after different durations of running, (i.e., 50,000, 100,000, 200,000, etc. cycles). The specimens were then broken in static bending while subjected to liquid air (to produce brittleness). The results are shown in Fig. 3. The first discernible crack occurred at about 300,000 cycles, corresponding to the knee of the curve. Thereafter the breaking load decreased rapidly due to crack penetration. However, a weakening is noted before 300,000 cycles, as shown by the initial slope of the curve. This would seem to indicate that some kind of damage occurs right from the start of the test. While the significance of these tests is not altogether clear, some further work along the same lines may be of value.

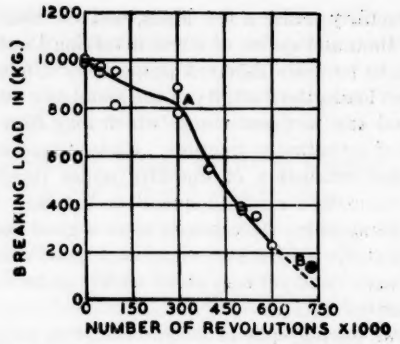


FIG. 3 RESULTS OF DAVIDENKOFF'S TESTS

[9] Davidenkoff, N., and Schwenandin, E., "Über den Ermudungsriss," *Metallwirtschaft, Metallwissenschaft, Metalltechnik*, Sept. 11, 1931.

APM-55-4

The Present State of the Turbulence Problem

By WALTER TOLLMIEN,[1] PASADENA, CALIF.

In this survey the author first describes certain types of turbulent flow, following which he deals successively with the production of turbulent motion; the instability of the laminar motion; fully developed turbulence; momentum interchange and mixing lengths; and relations between the shearing stress at the wall and the mean velocity distributions. Finally he takes up the calculation of skin friction for simple cases of fully developed turbulence, especially for that of the flat plate. Although the methods outlined have often led to practically useful results, it is the author's belief that they should be considered only as advances toward the real goal of the turbulence theory. The derivation of turbulence phenomena from the hydrodynamical equations will, in his opinion, be possible only by the application of statistical methods.

THE turbulence problem is especially interesting for two reasons. First, most flows occurring in technical applications and in nature are turbulent. Thus the turbulence problem has a great practical importance for aeronautics, hydraulics, meteorology, and so on. Secondly, this curious form of flow offers the greatest difficulties when we attempt to introduce it into the system of our usual hydrodynamics. By studying this problem we can hope to gain quite new basic physical insights.

Of course, the author can only present a survey of this complicated subject. For a long time experiments have been made on turbulence. Although we are still far from a complete comprehension even of the simplest turbulent motion, recently certain ideas have been developed that enable us to classify the very extensive experimental material on hand, as well as to forecast the character of some turbulent flows to a certain extent. In the following the author will outline these methods and give some results, but especially will he point out the problems still unsolved.

I—Some Types of Turbulent Flow

Instead of giving a definition of turbulence, which is difficult, the author will describe briefly the oldest example of a turbulent flow. Consider water flowing under a pressure drop through a straight tube of circular cross-section. If the material of the pipe is glass, then the flow can be made visible by injecting several color bands into the fluid, as did Osborne Reynolds, the father of turbulence research (1).[2] At low speeds of the fluid there are then seen at sufficient distances from the inlet of the pipe quite straight color bands parallel to the axis of the pipe. This form of flow is called "laminar." If we increase the speed of the fluid by increasing the pressure drop, at the so-called 'critical speed" the regular motion indicated by the color bands breaks down. We then see at first a quite irregular fluctuation of the color bands, and later only a colored cloud in the pipe, which means that the fluid is mixed to a high degree. The fluid now shows the so-called "turbulent" motion. Both facts, the irregularity of the motion and the mixing effect, are quite typical for turbulent flows.

In considering circular pipes with different diameters, and other fluids, e.g., air instead of water, we have to refer the observations to the non-dimensional Reynolds number, defined in this case as the mean velocity through the cross-section of the pipe times the radius r of the pipe divided by the kinematic viscosity ν. Neglecting the compressibility of the fluids, the transition between the laminar and turbulent motion is then determined by the "critical" Reynolds number.

Other fluid motions along rigid walls show similar behavior. Mention may be made of the change in the character of flow in the boundary layer[3] around a sphere causing the well-known sudden drop of the drag at the critical Reynolds number. In addition to this "wall-bound turbulence," essentially determined by the proximity of rigid walls, there exists also "free turbulence," i.e., turbulent flow far from the influence of walls. In this class may be mentioned the spreading out of an air jet in the still air or of a wake behind a body. We can say that "free turbulence" occurs when two fluid layers with a finite velocity difference come together. In this case the transition to turbulence seems to be more gradual, for there exists already at low Reynolds numbers a periodical motion (e.g., a von Kármán "vortex street") which is replaced at higher Reynolds numbers by the faster and more irregularly fluctuating turbulent flow.

The transition from simple regular motion to turbulent motion presents the first great problem.

II—Production of Turbulent Motion and Instability of the Laminar Régime

To illustrate the ideas we shall consider again the simple example of the flow through a pipe. A second reason for the choice of this flow is that almost innumerable experiments have been carried out for this case. We know that at a sufficient distance from the entrance of the pipe the laminar velocity distribution plotted against the radial distance from the axis forms a parabola. Why does not this simple solution hold beyond a certain value of the Reynolds number? The first answer that suggests itself is that this velocity distribution is only a solution of the hydrodynamical equations in a finite region of Reynolds numbers. We can convince ourselves at once that this conjecture is not correct. The flow in question satisfies the hydrodynamical equations of Navier-Stokes for all Reynolds numbers, and this holds generally. Considering the flows through arbitrary channels or past bodies of whatever kind, laminar solutions of the hydrodynamical equations for all Reynolds numbers always exist. Therefore turbulence can only originate because the laminar motion becomes unstable.

What do the experiments on this problem show? First, let us refer again to the flow through a tube of circular cross-section. The first finding of the experiments is that the critical Reynolds number is not a constant. Of course, the critical Reynolds number depends largely on the form of the mouth

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[1] California Institute of Technology.

[2] Numbers in parentheses refer to items in bibliography at end of paper.

Presented at the Pacific Coast Aeronautic Meeting held at Berkeley, Calif., June 9 and 10, 1932, under the auspices of the San Francisco Section of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] The (laminar or turbulent) "boundary layer" is defined as the thin layer where the velocity increases rapidly from the velocity of the wall to a value which can be calculated from the pressure drop along the wall by using Bernoulli's equation. This last possibility indicates that at that place there is practically no energy dissipation. This means that the boundary layer is just the region of the action of viscosity or of turbulent fluctuations.

through which the fluid streams from the reservoir into the pipe. If the entrance is sharp-edged, a critical Reynolds number of about 2000 is obtained, while for a carefully rounded entrance (trumpet mouth) it is in the neighborhood of 5000. These differences are not astonishing, for the two cases are not completely geometrically similar. But even for the same entrance quite different critical numbers are obtained. By carefully avoiding vibrations of the apparatus and leaving the fluid in the reservoir at rest for several days, the critical Reynolds number can be considerably increased. In fact, Ekman (2) succeeded in obtaining a critical Reynolds number of 22,000! This means that the critical Reynolds number depends largely on the disturbances that take place.

Here must be stated the great lack of all these experiments: namely, the disturbances causing the observed instability of the laminar flow are not defined. Some first attempts have recently been made in applying definite disturbances (3, 4). Hitherto the most important observation on the mechanism of the instability of the laminar motion was that the inlet disturbances are most effective. Even a slight scratch in the trumpet mouth can lower the critical Reynolds number to one-

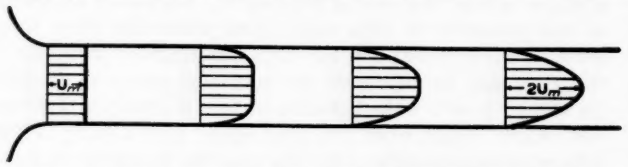


FIG. 1 FLOW IN THE LAMINAR INLET LENGTH OF A PIPE
(Inlet length about 0.15 $rU_m r/\nu$.)

half its previous value, while pronounced roughness of the walls far distant from the entrance has no influence at all. On the other hand, instability always begins in the "inlet length" of the pipe (Fig. 1). This length is defined as the region where the nearly constant velocity distribution at the entrance is gradually transformed into the final parabolic velocity distribution. The author does not know of any observation indicating that instability begins after the parabolic velocity distribution has been practically completely developed.

In other cases also experiments on the transition from laminar flow to turbulence have been made, although not to such an extent as for pipes. Mention may be made of investigations on the flow along a thin flat plate lying along the direction of motion, a case which will be treated later in detail. This transition is also observed in the boundary layers along the surface of airfoils or circular cylinders placed in an infinite stream. The boundary layer near the forward stagnation point is laminar; going downstream from this point along the surface of the body, however, the boundary layer sometimes becomes turbulent.

All these experiments refer to the so-called "upper critical Reynolds number," that is, their aim is to determine a limit for the stability of the laminar flow. The inverse problem is connected with the "lower critical Reynolds number." One then starts with a turbulent flow and finds, by gradually decreasing the Reynolds number, when the turbulent flow is no longer possible and is replaced by a laminar flow; i.e., the "lower critical Reynolds number" determines the limit of stability of the turbulence. In the experimental determination of this value the disturbances of the laminar flow are made "as great as possible," which is hardly in accordance with a proper formulation of the problem. According to L. Schiller (5), below a Reynolds number of about 1000, turbulence in pipes should be impossible.

To sum up, it may be said that the experiments that have been carried out show very little how turbulence is set up. What, now, can we learn from theory?

III—INSTABILITY OF THE LAMINAR MOTION: THEORY

The first method of attacking this problem is that proposed by Osborne Reynolds. A disturbance is superimposed upon the laminar flow, and the question is then asked whether the energy of the disturbance will increase or decrease. The disturbance assumed is to satisfy only the continuity and boundary conditions (e.g, all disturbance components vanishing at rigid boundaries), otherwise it is quite arbitrary. One part of the disturbance energy is dissipated by internal friction. Disturbances can increase their energy only by taking energy from the laminar basic flow. When this last amount of energy is greater than that dissipated, Reynolds assumes instability of the laminar flow (6). This conclusion is not justified because a momentary increase in the energy of the disturbance does not always mean instability of the laminar flow. Indeed, this method only shows whether some disturbance will in the next moment increase or decrease, and affords no information as to what will happen when the disturbance develops further according to the hydrodynamical equations. We obtain by this method only a limiting Reynolds number below which the laminar flow is certainly stable, since then every disturbance is decreasing in energy. However, the corresponding Reynolds numbers are so small that this limitation of the range of the critical ones is practically useless. Thus the Reynolds energy criterion fails completely with respect to the stability problem.

To describe the development of the disturbances according to the hydrodynamical equations, and thus to determine the actual instability of the laminar flow, the method of small oscillations is employed. Usually a two-dimensional laminar flow in the xy-plane is considered. For simplicity, this basic flow is assumed to have only a velocity component parallel to x and depending on y: $U(y)$. The disturbances are also taken as two-dimensional and can therefore be expressed by a stream function $\psi(x, y)$. Furthermore the disturbances are assumed to be small and all quadratic terms in ψ are neglected. Thus the following equation for ψ remains:

$$\frac{\partial \Delta\psi}{\partial t} + U\frac{\partial \Delta\psi}{\partial x} - \frac{d^2U}{dy^2}\frac{\partial \psi}{\partial x} = \nu \Delta\Delta\psi$$

Here $\Delta\psi$ is equal to $\frac{\partial^2\psi}{\partial x^2} + \frac{\partial^2\psi}{\partial y^2}$, i.e., the curl or the vorticity of the disturbance, t the time, and ν the kinematic viscosity.

Now, according to the method of small oscillations, the disturbance is to build up from "partial oscillations," for since our disturbance equation is linear, such a superposition is permissible. ψ is composed of terms such as

$$\varphi(y)e^{i(\alpha x - \beta t)}$$

that is, of waves traveling in the x-direction. The concise complex notation is used; it is to be understood that, e.g., the real part of the expression above is to be taken. α is a real constant giving the spatial cycle, therefore the wave length of the partial oscillation is $2\pi/\alpha$. β can be complex $= \beta_r + i\beta_i$; the real part β_r gives the periodic cycle, and the imaginary part β_i the logarithmic decrement, which is positive for increasing and negative for damped oscillations. The real part c_r of the magnitude $c(= \beta/\alpha)$ is the wave velocity. By introducing this partial oscillation into the disturbance equation, we obtain

$$(U - c)(\varphi'' - \alpha^2\varphi) - U''\varphi = -\frac{i\nu}{\alpha}(\varphi'''' - 2\alpha^2\varphi'' + \alpha^4\varphi)$$

Taking into consideration the boundary conditions (vanishing of the disturbance components at rigid walls, etc.), we now have to find the relation between the three parameters of the

problem, namely, the Reynolds number R of the basic flow, and α and c, the parameters of the partial disturbance. For positive imaginary parts of $\beta = c\alpha$, instability will arise.

The first problem attacked by this method refers to the "simple shearing motion" or the "straight Couette flow" (Fig. 2). This flow is produced between two plates parallel to x which are moving with opposite constant velocities in the x-direction. The basic velocity U is then a linear function of y, which simplifies essentially the mathematical problem. The investigations of Orr (7), Hopf (8), and von Mises (9) show no instability of this

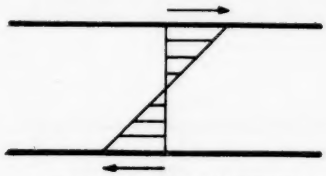


FIG. 2 STRAIGHT COUETTE FLOW

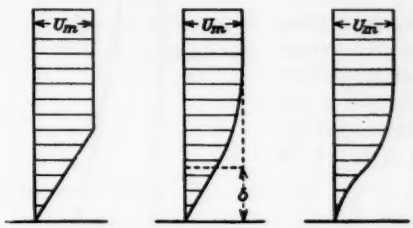


FIG. 3 VELOCITY PROFILES INVESTIGATED FOR STABILITY

flow, and accordingly it has been suggested that the proposed method fails in general for stability problems. This conclusion is not justified, because the straight Couette flow is too special an example.

Instability was first found by Prandtl (11) and Tietjens (12). But to avoid the mathematical difficulties involved by a persistent curvature (d^2U/dy^2) of the basic flow distributions, Prandtl replaced the gradually curved velocity distributions by polygonal profiles as first proposed by Rayleigh (10). He took into account the effect of a small viscosity, while Rayleigh had neglected the viscosity in the disturbance equation entirely. Now in certain cases this last effect causes an instability, e.g., in the first profile of Fig. 3, which shows a linear increase of the velocity from the wall over a certain distance, and further an indefinitely long continuation at constant velocity. But in the range of Reynolds numbers covered by the calculations some amplified oscillations always exist, while we should expect only damped oscillations below a certain Reynolds number. Of course this strange result of the calculations is due to the rough approximation of the profile by a broken line.

This explanation was corroborated by the author who investigated the case of smooth curved profiles (13). His discussion is carried out in detail for the laminar flow along a flat plate lying in the direction of the stream (Fig. 4). The corresponding velocity profile (second profile of Fig. 3) is characterized by starting with vanishing curvature (U'') at the wall and by reaching the maximum velocity U_m asymptotically. The form of the velocity distribution remains the same all along the plate, while the thickness of this laminar boundary increases slowly with the distance x from the leading edge [as $\sqrt{(\nu x/U_m)}$]. This small dependence on x can be neglected in the disturbance equation.

Figs. 5 and 6 show the results of the discussion for the case of neutral stability—where the transition from stability to instability occurs. The $\alpha\, c_r$ diagram gives the reciprocal wave lengths and the wave velocities of the corresponding partial oscillations, while the α-R diagram shows the corresponding Reynolds numbers. The Reynolds number is here defined as $R = U_m\delta^*/\nu$ where (cf. Fig. 3)

$$\delta^* = \int_0^\infty \frac{U_m - U}{U_m}\, dy$$

The usual definition of the boundary-layer thickness is rather arbitrary for the asymptotic transition of U into U_m, and is in this case $3.14\delta^*$. The inside of the curves gives the unstable region. The smallest Reynolds number where not all of the oscillations are damped is 420. The usual experimental value of the critical Reynolds number obtained by Burgers (14), van der Hegge Zijnen (15), and Hansen (16) is 950. By plotting this value in our α-R diagram fair agreement is shown, especially when we consider the strong influence of the special quality (wave length and wave velocity) of the disturbances. Disturbances of long wave length are those chiefly dangerous to the stability, and this is fairly well shown by certain photographic observations of the streamlines in the stage of incipient instability (11). In comparing theory with the experiments we should also not forget that the experimental critical Reynolds numbers correspond to an already considerable deviation of the flow from the laminar state instead of to the incipient instability in the theory.

This example is not so special as it would appear, since similar laminar profiles occur in the inlet length of a straight, deep channel with parallel walls, and generally in the laminar boundary layers along any bodies near a stagnation point.

On the other hand, certain modern investigations have confirmed a conjecture of Lord Rayleigh (17) that profiles with an inflection point (cf. third profile of Fig. 3) are highly unstable. We can realize that a low instability as found for the profile along the flat plate produces profiles with an inflection point, and that more violent breakdown of these profiles leads to turbulence.

Also the apparent contradiction between the theoretical

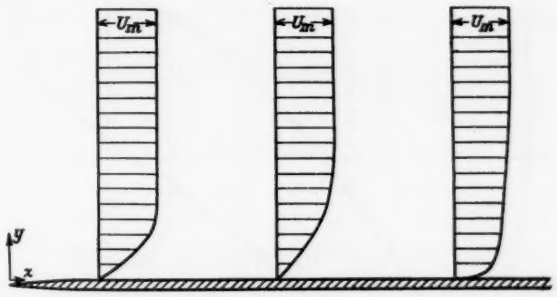


FIG. 4 TYPICAL TRANSITION BETWEEN LAMINAR AND TURBULENT FLOW ALONG A FLAT PLATE

[The first velocity profile is laminar, the second shows just a slight deviation from the laminar type, while the third is completely turbulent. The scale of y (normal to the wall) is about 50 times that of x (along the wall).]

stability of the straight Couette flow and the results of certain experiments has been recently cleared up. Indeed, the shearing motion between two straight parallel plates has never been observed, but the laminar motion between two concentric rotating cylinders (proper "Couette flow") shows a nearly linear velocity distribution over the radial distance between the walls if the difference of the radii of the two cylinders is small compared with the radii themselves. But observations when the outer cylinder was rotating and the inner one was at rest[4] showed the possibility of instability and turbulence. The reason is that at the starting of the whole system by the beginning

[4] The case when the inner cylinder is rotating is not available in this connection, because then, before the beginning of turbulence, the simple two-dimensional laminar motion is replaced by a regular three-dimensional periodical motion, as has been discussed by G. I. Taylor (18).

of the rotation of the outer cylinder, unstable profiles occur before the approximately linear profile has developed. This explanation is due to Prandtl (4) and has been checked by the calculations of H. Schlichting carried out according to the same method that was used by the author in the stability discussion of the profile along the flat plate. Of course, in the new problem the action of the centrifugal force has also been taken into account. The Reynolds number, defined as the circumferential velocity times the radius of the outer cylinder divided by the kinematic viscosity, has, according to certain observations, the critical value 70,000, while the theory places it at 66,000.

Although much detail work is still to be done along these lines, we can say that the instability of the laminar motion against small disturbances and the production of turbulence have been cleared up in principle. Of course, in reality the processes will often be more complicated than those assumed theoretically, because the disturbing oscillations will not be infinitesimal and often also three-dimensional.

IV—Fully Developed Turbulence: Momentum Interchange and Mixing Length

Proceeding to the second great problem of turbulence, the description of the fully developed turbulent flow itself, it must be stated at the beginning that we are still far away from an exact interpretation. But empirical rules and semi-empirical speculations provide useful material for technical application and future theory.

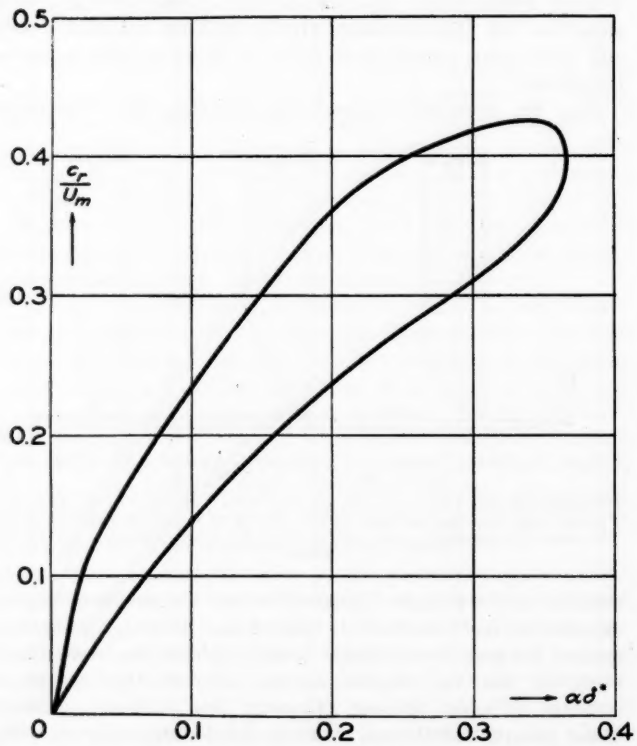


Fig. 5 Partial Oscillations in the Instability Region of the Laminar Flow Along a Flat Plate

Since fluid motion in the turbulent state is, according to all observations, so extremely irregular and complicated, we must at first confine ourselves to the consideration of mean values. Thus, according to Reynolds (6), we separate the actual velocity components u, v, w relative to the rectangular axes x, y, z and the pressure p into time averages $\bar{u}$, $\bar{v}$, $\bar{w}$, $\bar{p}$ at the point under consideration and the instantaneous deviations from the mean or the "fluctuations" u', v', w', p', so that, for instance,

$$u = \bar{u} + u' \qquad p = \bar{p} + p'$$

Now we can use the same form of equations of motion and continuity for the mean values as were used for the original values, if we but add a system of apparent stresses due to momentum interchange produced by the fluctuations.

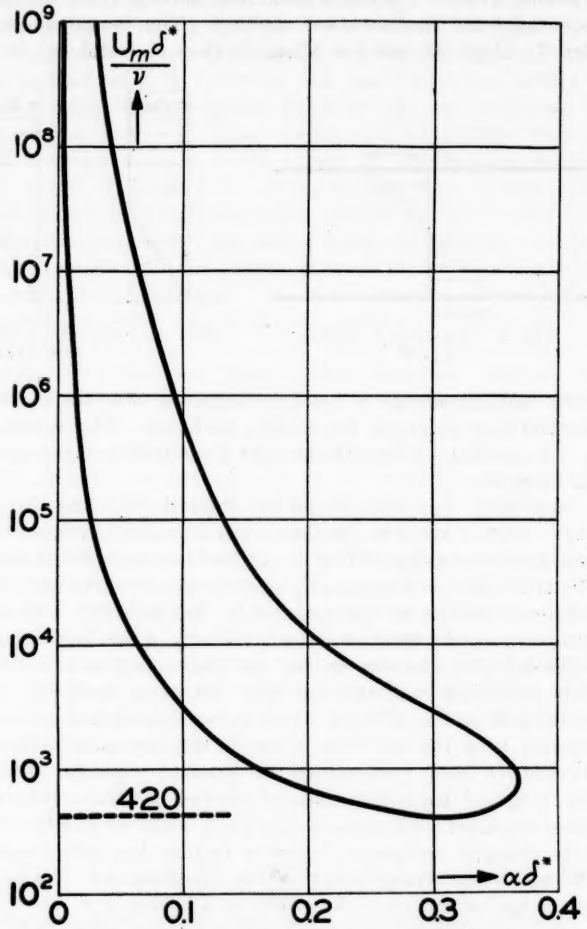


Fig. 6 Instability Region of the Laminar Flow Along a Flat Plate

For simplicity let us consider turbulent flows with a two-dimensional mean motion in the xy-plane. The fluctuations or the "relative motion" may be, and in general will be, three-dimensional. Furthermore, let us assume that the mean motion has its principal component $\bar{u}$ in the x-direction, in which case it will depend but slightly on x. In other words, we admit only a slight divergence or convergence or curvature of the streamlines of the mean motion. The following considerations are easily transferable to an axial symmetrical mean motion, y then being interpreted as the radial distance from the axis and v as the radial velocity component.

In the above-mentioned class of flows an apparent shearing stress like the effect of an apparent friction is of main importance in the additional stress system. Characterizing the apparent stresses by underscoring, we obtain this new shearing stress expressed by the mean momentum flux of the fluctuations:

$$\underline{\tau} = -\rho\overline{u'v'}$$

where ρ is the density of the fluid, and the overscoring again denotes mean values. In a deep channel with straight walls

parallel to x we can easily express the total shearing stress at large distances from the entrance where all mean values formed of velocities are already independent of x;

$$\tau = -\frac{d\bar{p}}{dx}(b - y)$$

y is taken from one wall of the channel, b is the half-breadth of the channel, the mean pressure drop is constant in the considered region, and τ is composed of the apparent stress $\underline{\tau}$ and the viscous stress $\mu d\bar{u}/dy$. Except for the immediate vicinity of the wall, this last stress produced by molecular or thermal motions is very small compared with the apparent stress. A quite similar relation exists in the case of the pipe.

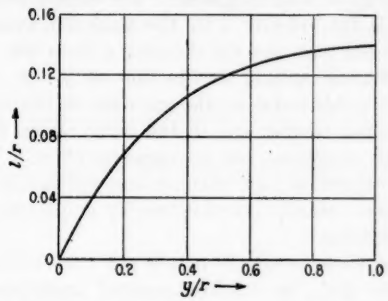


FIG. 7 THE MIXING LENGTH IN A PIPE

Statements of this kind, which all result from applying the formal process of forming mean values in the hydrodynamical equations, express the only exact relations which we know of at present between the apparent stresses and the mean velocities and pressure.

Yet Prandtl has put the apparent shearing stress into another form which at least facilitates further discussion in many cases (19). He introduces the "mixing length" l, a conception in some ways analogous to the idea of mean free path in the kinetic theory of gases. Let us consider the fluid particles that enter (by the turbulent fluctuations) a layer at a given y ($\bar{u}$ is essentially a function of y as mentioned before). Prandtl assumes that these fluid particles have a velocity u in the x-direction which corresponds approximately to the mean velocities $\bar{u}$ of layers at the positive or negative distances $\pm l$ from the given layer. In other words, the fluid particles travel the distance l without losing their individual properties, in this case the velocity u. Thus the mean value of the velocity fluctuation u' for the fluid particles coming from one side of the layer is $l d\bar{u}/dy$, and from the opposite side, $-l d\bar{u}/dy$. Prandtl further assumes that the velocity fluctuations v' in the y-direction are induced by the fluctuations u' in the x-direction when fluid globules with different velocities u meet or separate. The mean absolute value of v' can therefore be put proportional to $l\left|\frac{d\bar{u}}{dy}\right|$. Taking this proportionality factor together with l and also the correlation factor occurring when we average a product, we obtain Prandtl's final formula:

$$\underline{\tau} = \rho l^2 \left|\frac{d\bar{u}}{dy}\right| \frac{d\bar{u}}{dy}$$

Of course, it is assumed that l depends chiefly on the geometrical configuration of the flow considered, and to a lesser degree on the viscosity, etc., so that the shearing stress $\underline{\tau}$ is approximately proportional to the density and to the square of the velocities.

Now since we do not know the value of l, there seems little to be gained by using this formula. However, we can first calculate l from experimental values of $\underline{\tau}$, which latter is determined from the mean velocities and pressure as has been shown in an example. l is well suited to give an idea of the mixing process. For the present, consider a distribution of l in the turbulent flow through a pipe (Fig. 7). As would be expected, l increases linearly from the wall with good approximation over a considerable distance from the wall. This linear increase determines essential features of this turbulent flow, as will be seen later.

Secondly, in the cases of free turbulence a very simple assumption with respect to l gives good results. To illustrate, consider the mixing of a uniform stream with the still air along one side (Fig. 8). u_0 may be taken as the speed of the uniform stream, and x will be measured from the beginning of the mixing. Then the velocity profiles $\bar{u}(y)$ can be assumed to remain similar for all distances x. We now introduce this assumption into the equation of motion and take a shearing stress proportional to the density ρ and the square of the velocities by neglecting the effect of viscosity. Thus for $\bar{u}$ according to Prandtl, the form

$$\bar{u} = u_0 f\left(\frac{y}{x}\right)$$

is obtained where the boundary conditions are already considered. Thus the breadth of the mixing region is proportional to x. Now the mixing length l will be constant over every cross-section since no walls hinder the mixing, which means that l does not depend on y. On the other hand, for the form in which $\bar{u}$ is given above, l must be proportional to x, or

$$l = cx$$

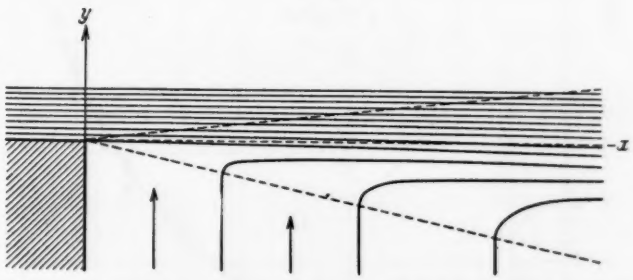


FIG. 8 STREAMLINES OF THE MEAN TURBULENT FLOW AT THE BOUNDARY BETWEEN A UNIFORM STREAM AND STILL AIR

c is the only experimental constant necessary for the calculations. The agreement of the velocity distribution calculated by the author (20) with experiment is very good. This case of turbulent mixing is of fairly great practical importance. It may be mentioned that the unavoidable part of the energy loss of an open-air wind tunnel is partially determined by this effect.

The spreading out of jets (20) and wakes (21, 22) has been treated in a similar manner. And in all cases this simple assumption that l is constant over every cross-section leads to a satisfactory agreement with observations.

Recently von Kármán has developed the mixing-length hypothesis from another standpoint (23). He assumes that the momentum transport of the turbulence is determined only by the field of flow in the immediate vicinity of the point considered, which means that the apparent shearing stress at *all* points in the fluid can be expressed in the same formal manner by the mean flow close to that point. This assumption will not seem so strange when we think of an analogous case: the laminar shearing stress produced by the thermal motion of the molecules obeys such a "local law," i.e., τ (laminar) $= \mu du/dy$. At any rate the assumption of a local law is the simplest possible. This

hypothesis leads to the further conclusion that the flow field of the fluctuations is *similar* about all points in the flow, on an average at least, only the time and length scales being changed. Introducing this assumption into the equation for the fluctuations where the quadratic terms are not neglected, von Kármán again obtains Prandtl's formula for the shearing stress, expressed by the mixing length, and in addition a formula for the mixing length itself:

$$l = \kappa \left| \frac{\frac{d\bar{u}}{dy}}{\frac{d^2\bar{u}}{dy^2}} \right|$$

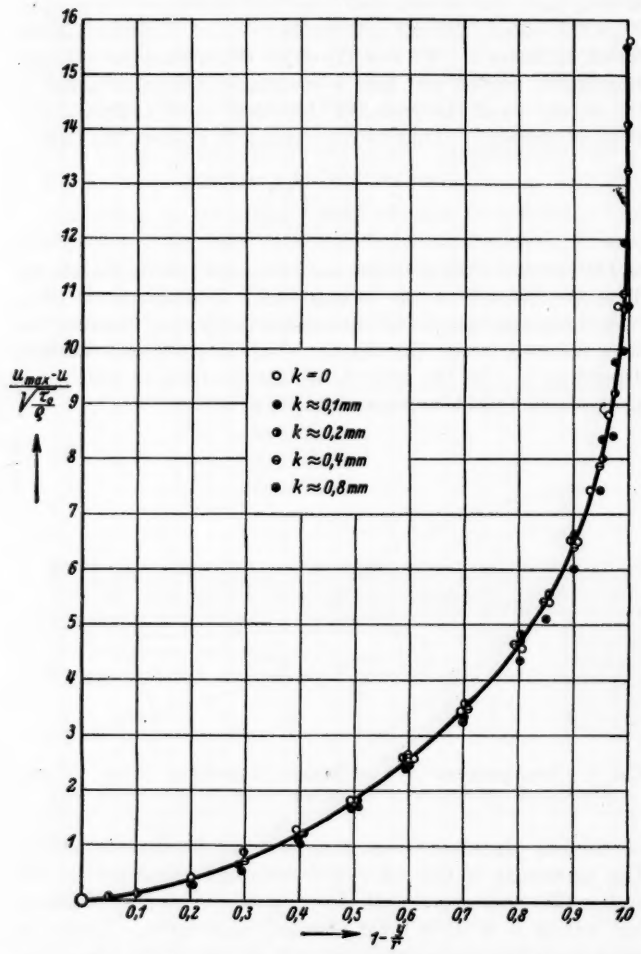


FIG. 9 DIAGRAM ACCORDING TO STANTON AND VON KARMAN OF MEAN VELOCITY DISTRIBUTIONS IN SMOOTH AND ROUGH PIPES AT A REYNOLDS NUMBER OF ABOUT 160,000

[In these experiments of Nikuradse (26) the walls had been roughened by gluing on grains of sand, the mean height of which over the wall was taken as the approximate length of roughness k.]

κ is a universal constant valid for all the types of flow considered where $\bar{u}$ is the principal component of the main flow. This constant κ, according to experiments, has a value of about 0.4. Although this formula breaks down in the region where the $\bar{u}(y)$-distribution has a maximum ($d\bar{u}/dy = 0$) or an inflection point ($d^2\bar{u}/dy^2 = 0$), we can eliminate in this way some of the previous rather arbitrary assumptions in regard to the mixing-length distribution. In particular this formula gives the linear increase of the mixing length near a wall ($l = \kappa y$) and its approximate independence of y in the cases of free turbulence.

V—FULLY DEVELOPED TURBULENCE: RELATIONS BETWEEN THE SHEARING STRESS AT THE WALL AND THE MEAN VELOCITY DISTRIBUTIONS

The second class of conceptions which make possible a much simplified description of the experimental facts, employs essentially certain empirical relations between the mean velocity distribution and the shearing stress at the wall which we shall call τ_0. Consider again the flow through a pipe. Then τ_0 is easily determined by the measured pressure drop, and the total-shearing-stress distribution is known as a linear function of the normal distance y from the wall as mentioned before.

Let us now compare measured velocity distributions in different cases, i.e., for different Reynolds numbers, and also for different degrees of wall roughness. For such a comparison we have to divide the velocity $\bar{u}$ by the maximum velocity $\bar{u}_{max}$ at the center of the pipe and the distance y from the wall by the radius r. Plotting $\bar{u}/\bar{u}_{max}$ as function of y/r we see that at increasing Reynolds numbers the gradient of the velocity near the wall becomes steeper and in the inner region flatter, since an increase in roughness has an opposite effect. Now it is a curious and important fact that we can essentially reduce the variety of these velocity distributions by employing a different method of plotting.

The first of these graphs is rather old (1911) and is due to T. E. Stanton (24), but its fundamental importance was first realized recently by Th. von Kármán at Aachen (25). Stanton makes the *difference* between the maximum velocity $\bar{u}_{max}$ and $\bar{u}$ non-dimensional by dividing by $\sqrt{(\tau_0/\rho)}$, and plots this quantity against the non-dimensional distance from the wall y/r. When we exclude the immediate vicinity of the wall and have sufficiently high Reynolds numbers (greater than about 50,000), this function g of y/r, or

$$\frac{\bar{u}_{max} - \bar{u}}{\sqrt{(\tau_0/\rho)}} = g\left(\frac{y}{r}\right)$$

is independent of the Reynolds number, that is, of the viscosity and the roughness of the walls (Fig. 9). Thus the viscosity and the wall roughness have no direct influence on the velocity distribution in the inner parts of the flow; they are of effect only through the medium of τ_0 and in the form of a kind of boundary condition, because the velocity distribution must of course have a continuous transition between the region close to the wall and the inner region of the flow.

The Stanton-von Kármán rule can be formulated in a very simple manner by making use of the mixing length, which procedure facilitates the physical conception. The complete equivalent of this rule is the statement that l has in all pipes the same distribution over the cross-section, or, as we can say, l/r is a function of y/r, given once for all (Fig. 7). This simple formulation again shows the advantage of introducing the mixing length and gives immediately a good idea of how the Stanton-von Kármán rule can be transferred to other conditions.

The counterpart of this rule is a relation between τ_0 and the velocity distribution near the wall and is due to L. Prandtl (27). But it is confined in its present state to smooth walls. Prandtl considers the flow near the wall, that is, in a region where the distance from the wall is small compared with the radius of the pipe. Then he assumes that $\bar{u}$ in this region depends only on the conditions in the neighboring surface, and is not affected by the flow conditions at a distance. This means that $\bar{u}$ shall only depend on y, τ_0, ρ, μ, but not on r and $\bar{u}_{max}$. We come to the most expressive formulation of this assumption by deriving from the five quantities $\bar{u}$, y, τ_0, ρ, μ a non-dimensional velocity and a non-dimensional distance from the wall:

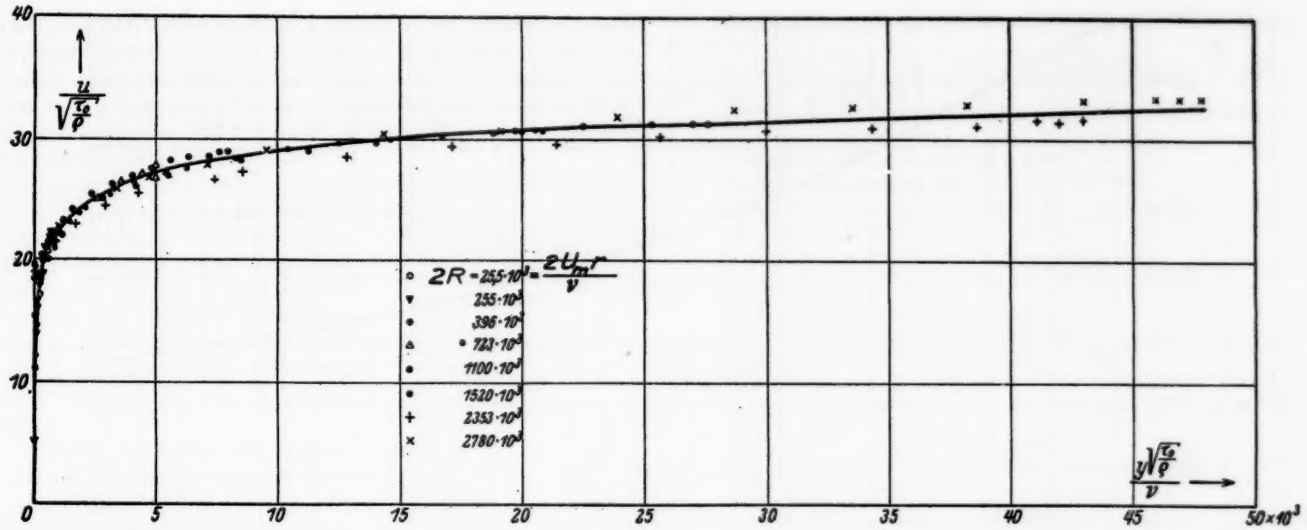


FIG. 10 UNIVERSAL u^* (y^*)-VELOCITY DISTRIBUTION CURVE NEAR SMOOTH WALLS ACCORDING TO MEASUREMENTS OF NIKURADSE (26) IN PIPES

$$u^* = \frac{\bar{u}}{} \sqrt{\frac{\rho}{\tau_0}}$$

$$y^* = \frac{y}{\nu} \sqrt{\frac{\tau_0}{\rho}}$$

Prandtl's assumption then means that u^* is only a function of y^* for values of y small against r. Plotting u^* against y^* we obtain a uniform curve in spite of the fact that for our diagram we used measured velocity distributions at quite different Reynolds numbers (Fig. 10). Of course, at lower Reynolds numbers smaller values of y^* are reached in the region near the wall than at higher Reynolds numbers; thus at a certain Reynolds number only a certain portion of the universal $u^*(y^*)$-curve beginning at $y^* = 0$ is realized. Fortunately this $u^*(y^*)$-curve can be represented by very simple formulas. The approximate formula

$$u^* = 8.7\, y^{*1/7}$$

which holds in a region of y^* between 20 and 1000, was found to have a great practical application. At higher values of y^* we can use the formula

$$u^* = \frac{1}{\kappa} \log y^* + \text{const}$$

in which the constant has a value of about 5.5.

Of course, these considerations can be immediately transferred from pipes to other flow conditions. We have to expect an analogous behavior near a smooth wall as long as the other boundaries have no great influence. Thus this rule checks experimentally very well in many other cases. Especially has the above-mentioned $^1/_7$-power formula often been found again, for instance, in flows through straight tubes with rectangular, triangular, and other cross-sections, in turbulent flow along a flat plate where the second logarithmic formula is also checked, and in accelerated and retarded flows as through convergent and divergent channels. On the other hand, this formula $u^* = 8.7y^{*1/7}$ has been used with great success for the calculation[5] of turbulent boundary layers and the distribution of τ_0, that is, the frictional forces at the surface of bodies. In this way von Kármán calculated the frictional resistance of a flat plate (28), to which case the author will return later, and the flow due to a rotating disk, and Clark B. Millikan (29) the frictional resistance of an airship. The agreement with the experiments is in all cases good. Only the calculated value of the frictional moment acting on both sides of the spinning disk will be given here: namely,

$$0.146\, \rho \frac{U^2}{2} r^3 \left(\frac{\nu}{Ur}\right)^{1/5}$$

where U is the circumferential velocity and r the radius of the disk.

VI—CALCULATION OF THE SKIN FRICTION FOR SIMPLE CASES OF FULLY DEVELOPED TURBULENCE, ESPECIALLY FOR THE FLAT PLATE

Furthermore, by a combination of these important conceptions, the mixing-length and the τ_0-rules, von Kármán succeeded in 1930 in formulating the law of the frictional resistance in pipes and along flat plates up to arbitrarily high Reynolds numbers (23, 30).

The last fact is extremely useful from the experimental point of view. Usually it is desired to make experiments with models of much smaller dimensions than those of the actual full-size objects. But, on the other hand, we have to extend our experiments to very high Reynolds numbers to be sure that we can transfer the results to practical cases. This last necessity generally involves much labor and great expense. For both problems this difficulty has now been eliminated, since we know the frictional resistance up to arbitrarily high Reynolds numbers.

The method will now be outlined. For the inner part of the flow, i.e., that not in the immediate vicinity of the wall, the viscosity can be neglected, as we have already seen, and the shearing stress calculated by means of the mixing length. According to the Stanton-von Kármán rule, l/r is always the same function of y/r. (For the plate we have of course to replace r by the thickness of the boundary layer.) Of this function only one

[5] For this purpose $u^* = 8.7y^{*1/7}$, which we can write in the original quantities as $\bar{u} = 8.7\ (\tau_0/\rho)^{4/7} \cdot (y/\nu)^{1/7}$ is extrapolated to the velocity at the outer end of the boundary layer. This last velocity is supposed to be known. Such extrapolation is generally risky, but causes no great errors in the following calculations. In the first $^1/_7$-power formula only the quantity τ_0 is still unknown, the distribution of which along the surface of the bodies is calculated by means of the momentum theorem.

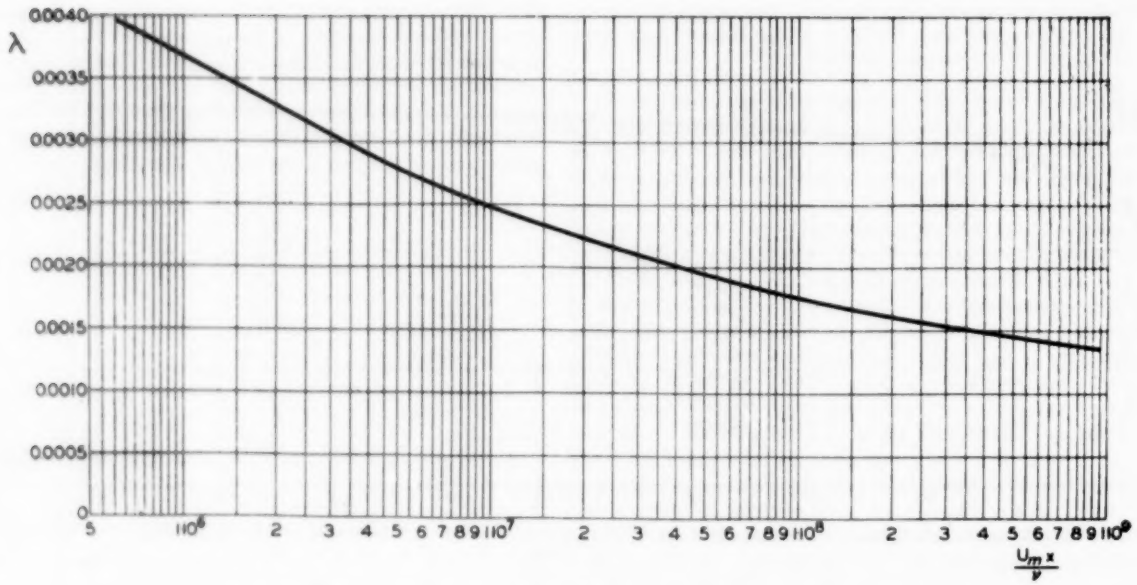


FIG. 11 SKIN FRICTION OF A TURBULENT FLOW ALONG A FLAT PLATE

property is used,[6] namely, that l/τ at first linearly increases with the distance from the wall, which means that $l/\tau = \kappa y/\tau$ or $l = \kappa y$. To connect this region of the flow with the region near the wall, Prandtl's consideration is used in the form that for a certain constant value of y^*, say, y_1^*, the corresponding value of u^* shall be a constant u_1^*. Both of these constants are supposed to be empirically known.[7]

The calculations of the frictional resistance based on these ideas check very well with experiment. In the case of the flat plate, let x be the distance from the leading edge of the plate, and U_m the velocity of the undisturbed flow far from the plate. Then in the diagram (Fig. 11) $\lambda = 2\tau_0/\rho U_m^2$ is plotted against $R_x = U_m x/\nu$. λ is a sort of local resistance coefficient just as R_x is something like a local Reynolds number. The total skin friction of a plate of certain length can easily be found by integrating τ_0 over the total surface of the plate. Of course, the boundary layer starts at the leading edge of the plate with a laminar portion, as has been frequently mentioned. For large values of $U_m x/\nu$ it is permissible to neglect this laminar portion and to begin at once with a turbulent motion as is done in the diagram and in the following approximative formula for large values of $U_m x/\nu$:

$$\kappa \sqrt{\frac{2}{\lambda}} = \log\left(\frac{U_m x}{\nu} \cdot \lambda\right) + \text{const}$$

the value of constant being about 1.01. It is well known that the case of a plate is often considered as a standard case of frictional resistance, and estimations are made therefrom for the skin friction of other bodies. The method outlined above can be employed to some extent in the case of rough walls (23).

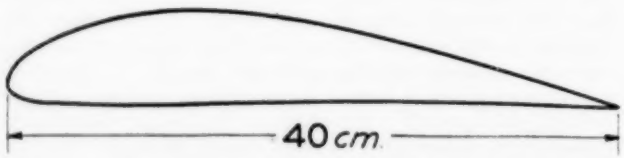


FIG. 12 AIRFOIL (GOETTINGEN PROFILE 387) USED BY GRUSCHWITZ (33) FOR MEASUREMENTS OF THE BOUNDARY LAYER ON THE UPPER SURFACE

VII—CONCLUSION

In concluding the present survey of modern investigations of turbulent phenomena, it must be admitted that in spite of recent surprising successes, only a small portion of the phenomena can be treated in such a systematic manner. Our considerations have all been limited to flows which have been mainly two-dimensional or axial-symmetrical. Moreover, the mean flow has been assumed to be essentially in one direction, x, with only slight curvature or divergence allowed.

Considering now only two-dimensional mean flow, the first outstanding problem is that of strongly accelerated or, in particular, strongly retarded flow. The last case is closely associated with the problem of investigating the conditions for the break away of flow from the upper surface of a body, i.e., the phenomenon of "burbling." This effect is very important in aeronautics because it determines the stalling of airplanes and therewith the limiting value of the maximum lift.

It has long been known that laminar flow has a greater tendency to break away than turbulent flow, but beyond this fact we know very little. Accurate measurements of retarded

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[6] It can easily be proved that this empirical proportionality constant is the same as the earlier constant κ, the value of which is about 0.4.

[7] Wada (31), as early as 1927, made a proposal which seems at first glance very similar to the procedure outlined above. He assumes that the total shearing stress is composed of the laminar stress $\mu d\bar{u}/dy$ and the turbulent stress, given again in the simple form $\kappa^2 y^2 (d\bar{u}/dy)^2$. But he does not exclude from this consideration the region quite near to the wall as did von Kármán, who replaced it by a boundary condition (u^* at y_1^*). Wada's interesting proposal leads, for the region quite near to the wall (τ is approximately τ_0), to the conclusion

$$u^* = \frac{1}{\kappa} \log y^* + \frac{1}{\kappa} (\log 4\kappa - 1)$$

since the experiments give for the additive constant a value of about 5.5 instead of —1.32 as given by the above formula. This means that this simple superposition of viscous stress and turbulent stress entirely uninfluenced by the viscosity is not possible. Close to the wall the viscosity has a decided effect on the development of the turbulent stress.

turbulent flow have been made, especially by Nikuradse, with divergent (wedge-shaped) channels of great depth (32). Attempts have also been made several times to transfer these results to other cases of retarded flow—however, without great success. There is still very little known about the conditions for break away of turbulent flow.

Figs. 13 and 14 show the recent measurements of E. Gruschwitz (33) of velocity profiles on the upper surface of an airfoil (Fig. 12) in the region of increasing pressure. The differences between the laminar and turbulent profiles are clearly seen. At an angle of attack of 6 deg the flow holds to the surface, while at 12 deg it breaks away.

Another important and difficult problem is that of turbulent flow along highly curved surfaces. In spite of many attempts to analyze the problem, there is still little known further than the fact that for surfaces convex with respect to the flow the turbulent momentum interchange is decreased, while for concave surfaces it is increased. The explanation is that the fluid particles near the wall are slowed down due to the influence of wall friction, and are therefore subject to less centrifugal force than the faster-moving particles at a greater distance from the wall. This effect is technically quite important, but does not generally come into consideration in aircraft engineering since the boundary-layer thickness for the usual airfoils is small compared with the radius of curvature of the surface.

The author would conclude this brief review of some significant problems of turbulence with a general remark. Although the methods outlined in the present report have often led to practically useful results, they should be considered only as advances toward the real goal of the turbulence theory. The derivation of turbulence phenomena from the hydrodynamical equations will, in the author's opinion, be possible only by the application of statistical methods. Using conceptions of the theory of probabilities we may represent turbulent flow as composed of random fluctuations whose mean values obey certain definite laws. There have already been several interesting attempts along this line (34, 35). Similar attempts are being made on the experimental side to go beyond the usual measurements of mean values and investigate the structure of turbulent flow by observation of the turbulent fluctuations themselves.

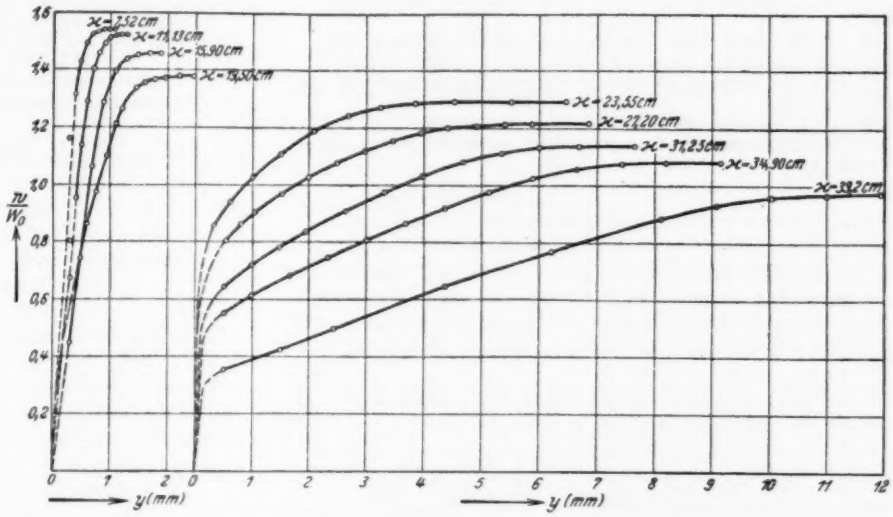


FIG. 13 DISTRIBUTIONS OF THE MEAN RESULTANT VELOCITY w AT THE UPPER SURFACE OF THE AIRFOIL OF FIG. 12 FOR AN ANGLE OF ATTACK OF 6 DEG AND A FLOW OF 30.8 M/SEC. VELOCITY (x is the distance along the upper surface measured from the leading edge. y is perpendicular to x.)

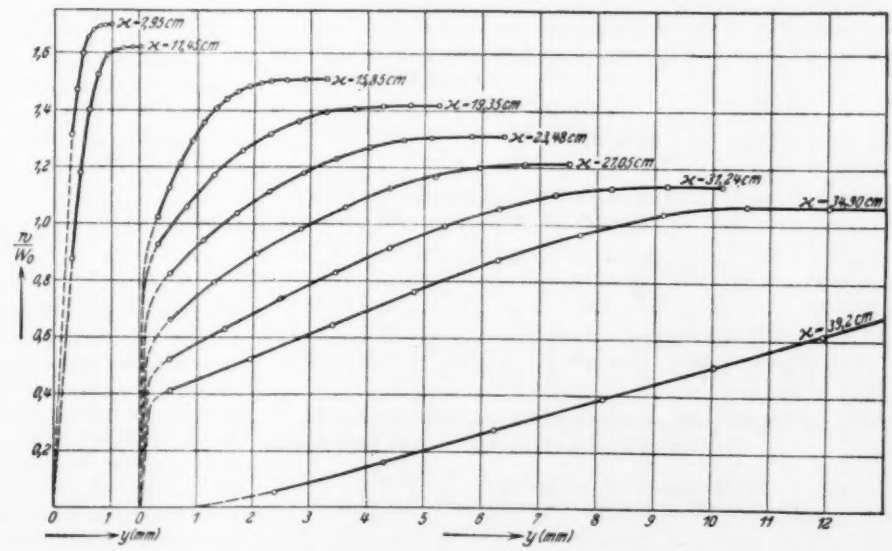


FIG. 14 SAME AS FIG. 13 BUT FOR AN ANGLE OF ATTACK OF 12 DEG

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APM-55-5

Analysis of Stresses in an Elliptic Ring

By MERIT P. WHITE,[1] PASADENA, CALIF.

Problems sometimes confront engineers the analytic solution of which requires quite a knowledge of mathematics. The author illustrates this with the calculations required in an investigation of a wind-tunnel problem. The method used is intricate and tedious. A simpler solution may be arrived at by use of the graphical method.



ENGINEERS are sometimes confronted with unusual problems whose analytic solution requires more knowledge of mathematics than most of them retain after completing their formal study of the subject.

Such a problem was encountered in connection with a recent investigation of a wind tunnel. This wind tunnel, Fig. 1, is made up of large circular pipes arranged to form a closed circuit and connected by stiff ring girders at the corners. These ring girders are of course elliptic in contour. The problem was to determine the stress at any point in the ring girder due to the dead weight of the structure.

In the case of ring a, Fig. 1, the horizontal pipe was considered cut just to the right of a, and a moment equivalent to the effect of the dead weight applied. An equal and opposite moment must exist within the vertical pipe. Each pipe is a beam with a stress at any point equal to My_c/I, where y_c is the perpendicular distance from the point to the neutral plane. This force is applied directly to the ring. The forces which one pipe exerts on the ring are shown in Fig. 2. The effect of the other pipe is exactly similar.

Let t be the thickness of the pipe; then in a distance ΔS_c (measured along a right section of the pipe) the total force acting in one pipe is:

$$P = \frac{My_c}{I}\ \Delta S_c t = \frac{My_c \Delta S_c}{\pi r^3}$$

which acts over a distance ΔS_e on the ellipse. Fig. 3 shows the relation between the elements of the ellipse and the pipe section. We have an equal force in the other pipe acting over the same portion of the ellipse. The effective force acting on the ellipse will be the resultant of these P's. Since the P's are at right angles, their resultant will be $P(\sqrt{2})$. This resultant will lie in the plane of the ellipse and will always be parallel to the major axis of the ellipse. This is evident from the fact that the plane of the ellipse is a plane of symmetry for both systems of forces, while, at any point, both forces and their resultant lie in a plane containing corresponding elements of both pipes, and therefore parallel to the major axis of the ellipse.

In a distance ΔS_e the force acting on the ellipse is

$$F = \frac{(\sqrt{2})My_c \Delta S_e}{\pi r^3}$$

We must first express $y_c \Delta S_c$ in terms of the elements of the ellipse. But $\Delta S_c = (r/y_c)\Delta x$. In a distance ΔS_e along the ellipse or in a horizontal distance Δx, the load on the ellipse equals $(\sqrt{2})M\,\Delta x/\pi r^2$. In other words, the horizontal distribution of the load is uniform.

Having completed the preliminaries, it only remains to apply enough equations of equilibrium (static and elastic) to determine the variation of thrust, shear, and moment along the ring.

The principal axes of the ellipse are axes of symmetry, so there can be no shear along these axes. Furthermore, as a result of this symmetry, we need consider only one quadrant of the ellipse.

Let the horizontal distribution of the load be K, where $K = M(\sqrt{2})/\pi r^3$. Then, considering the free-body diagram shown in Fig. 5:

$$\Sigma v = Kr - T_B = 0;\ T_B = Kr$$

$$\Sigma H = T_A = 0$$

$$\Sigma M \text{ about } B = M_A - M_B + (Kr^2/2) = 0$$

We need one equation from elasticity to determine M_A and M_B. This equation follows from the fact that the relative rotation of the tangents at A and B equals zero (from symmetry) or: The area of the M/EI diagram must be zero. E and I are constant. Therefore

$$\int_A^B M ds = 0$$

The major diameter of the ellipse is $(\sqrt{2})$ times the minor diameter. The equation of the ellipse is:

$$x^2 + \frac{y^2}{2} = r^2;\quad ds = \sqrt{\frac{r^2 + x^2}{r^2 - x^2}}\,dx$$

$$M = M_A - \frac{Kx^2}{2};\quad \int_0^r \left(M_A - \frac{Kx^2}{2}\right)\sqrt{\frac{r^2 + x^2}{r^2 - x^2}}\,dx = 0$$

Unfortunately, the latter expression is non-integrable. The substitution of $x = r\sin\theta$ gives a more simple form:

$$\int_0^{\pi/2}\left(M_A - \frac{Kr^2\sin^2\theta}{2}\right)\sqrt{1 + \sin^2\theta}\,d\theta = 0$$

This expression may be evaluated by expanding the radical in an infinite series, multiplying each term by the expression within the parentheses, and integrating. This gives a fairly satisfactory series. M_A can then be found as accurately as desired.

This method works, but is exceedingly tedious and requires a certain familiarity with mathematics. A simpler solution is desirable. For this reason, a graphical method was attempted.

We wish to determine the value of M_A which will make the area of the moment diagram equal to zero.

The first step is to lay out one quadrant of the ellipse (Fig. 6-*A*). Directly below this is drawn the moment diagram for the external forces, in this case a parabola (Fig. 6-*B*). Then,

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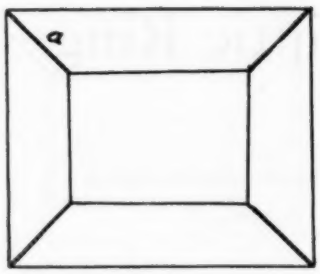


FIG. 1

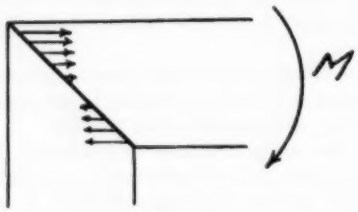


FIG. 2

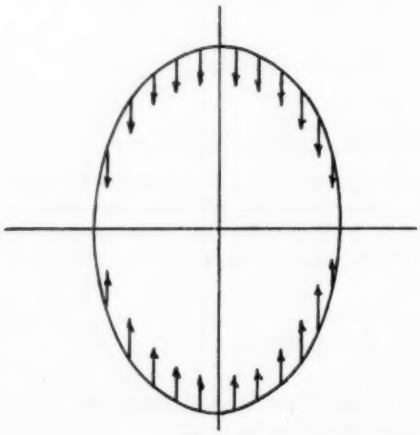


FIG. 4

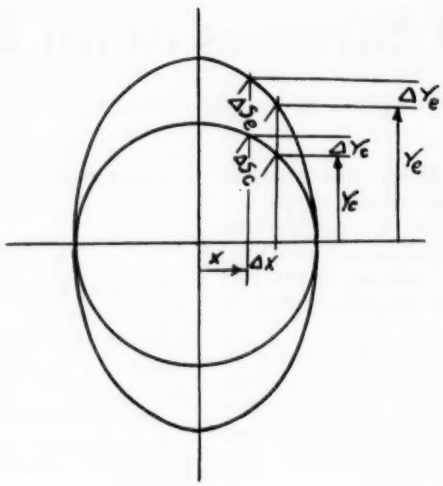


FIG. 3

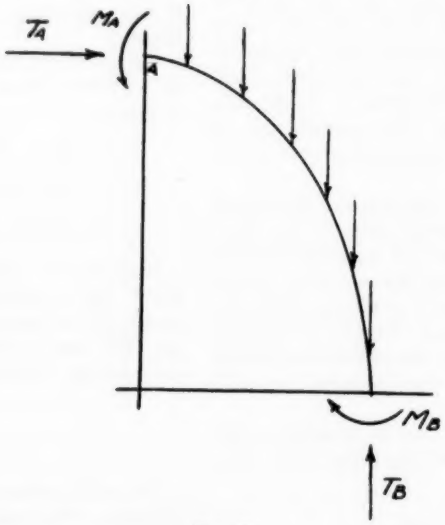


FIG. 5

on the developed length of the ellipse as an axis, there is constructed a new moment diagram (Fig. 6-*C*), whose ordinates are transferred directly from Fig. 6-*B*.

Then $M_A \times$ Developed Length = Area of Diagram, or M_A = average ordinate to moment diagram

Graphically, $M_A = 0.121M$
Analytically, $M_A = 0.1219M$

A series of concentrated loads or a change of cross-section (I) would not greatly complicate this process, but would make the analytic method almost impossible. For the simplest case, that of a uniform circular ring with continuous variation of load, the analytic method would be suitable.

If more than one equation from elasticity were needed—for example, if the moment of the moment diagram were also zero about some axis—this method could be extended. The moment diagram would be constructed as before and divided into areas with equal bases. Then the mid-ordinate of each area would be multiplied (algebraically) by its perpendicular distance to the axis about which moments are taken (obtained from Fig. 6-*A*). This would be done for the moment diagrams of M_A and V_A or H_A, expressed in terms of these unknowns. Finally, the algebraic sum of these moments would be set equal to zero, which would give another relation between the unknowns.

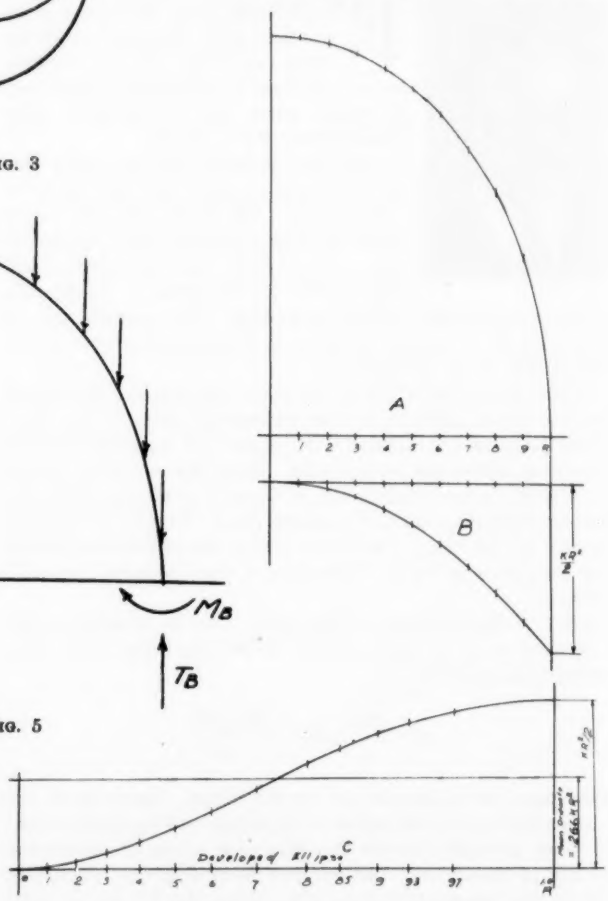


FIG. 6

APM-55-6

Stress Distribution in a Reentrant Corner

By J. H. A. BRAHTZ,[1] PASADENA, CALIF.

This paper deals with the stress distribution under plain strain in a corner of any angular magnitude, i.e., a plane with an angular incision or notch.

It begins with a brief statement of the method employed by Dr. Th. von Karman in his exact treatment of a beam in bending (Aachen Abhandlungen, Heft 7, 1927).

A generalization of this method is outlined which is applicable to the corner for any force distribution over the straight boundaries, and the solution is found for a concentrated load at any point of the boundary. The stresses are determined and shown to be infinite at the vertex of the corner.

The discussion points out the very interesting paradox that stresses may be finite for certain continuous loadings, but become infinite if a portion of the load is removed.



THE knowledge of two-dimensional stress distributions is useful for the solution of many problems in mechanical and civil engineering when the usual methods of strength of materials are not applicable or not sufficiently accurate. The stress distribution in the half-plane under the influence of a concentrated force acting on the straight boundary is of fundamental importance, and the solution of this problem is due to Boussinesq. Several extensions of his results have been made by Michell,[2] especially by means of the so-called method of inversion, the only conformal transformation of biharmonic solutions which leaves the boundary stresses undisturbed. By superposition of Boussinesq's solutions the stress distribution in the half-plane is easily obtained for an arbitrary load distribution along the straight boundary.

The stress distribution in a plane strip has been discussed by several authors, and special reference is made to a paper by Dr. Th. von Kármán[3] containing a general method for the determination of the stresses due to an arbitrary load along the boundaries. The paper is not yet available in the English language, therefore a brief description will be given of von Kármán's formulation of the problem.

A two-dimensional elastic system[4] is in equilibrium if the stresses σ_x, σ_y, and τ are the second partial derivatives of an arbitrary function $F(xy)$:

$$\sigma_x = \frac{\partial^2 F}{\partial y^2}; \quad \sigma_y = \frac{\partial^2 F}{\partial x^2}; \quad \tau_{xy} = -\frac{\partial^2 F}{\partial x \partial y} \ldots\ldots\ldots [1]$$

In order to be compatible with the stress-strain relations in accordance with Hooke's law, F must be a solution of the equation

$$\frac{\partial^4 F}{\partial x^4} + 2\frac{\partial^4 F}{\partial x^2 \partial y^2} + \frac{\partial^4 F}{\partial y^4} = 0 \ldots\ldots\ldots\ldots [2]$$

or in Gibbs's notation

$$\nabla^4 F = 0$$

F, known as Airy's function, and its first partial derivatives can be interpreted in terms of the quantities usually employed and defined in "strength of materials," namely, shear and moment of the external forces. By integrating [1] along the boundary between points A and B it is seen that the differences $(\partial F/\partial x)_B - (\partial F/\partial x)_A$ and $(\partial F/\partial y)_B - (\partial F/\partial y)_A$ represent, respectively, the resultant Y- and X-components of the load on the boundary between A and B, and, integrating once more, it is seen that the difference $(F)_B - (F)_A$ represents the increase in the bending moment in going from A to B.[5]

Hence, if all loads are assumed applied at the upper surface of the strip x-axis (see Fig. 1), the usual moment curve becomes identical with $F(x,0)$ for the strip in question and the problem of stress distribution becomes simply the determination of $F(x,y)$ for given boundary values.

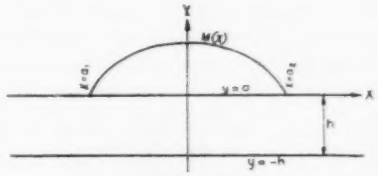


FIG. 1

It can easily be verified that the following functions are particular integrals of [2]:

$$A_1 \cos mx \cosh my,\ A_2 \cos mx \sinh my,\ A_3 \sin mx \cosh my,$$
$$A_4 \sin mx \sinh my,\ A_5 y \cos mx \cosh my,\ A_6 y \cos mx \sinh my,$$
$$A_7 y \sin mx \cosh my,\ A_8 y \sin mx \sinh my$$

and if we consider the coefficients A_1, etc. as functions of the parameter m, Airy's stress function can be written

$$F = \int_0^\infty \Big[(A_1 \cosh my + A_2 \sinh my + A_5 y \cosh my + A_6 y \sinh my) \cos mx + (A_3 \cosh my + A_4 \sinh my + A_7 y \cosh my + A_8 y \sinh my) \sin mx \Big] dm \ldots [3]$$

where A_1, etc. are to be determined by the boundary conditions.

The loads are assumed perpendicular to the upper surface of the strip or beam, and if the depth of the beam is h, the boundary conditions to be satisfied by F are:

$$\frac{\partial F}{\partial y} = 0, \quad F = M(x) \text{ at the upper boundary, } y = 0$$

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[2] A. E. H. Love, "Elasticity," p. 216.

[3] Aachen Abhandlungen, 1927, Heft 7.

[4] For explanation see J. Prescott, "Applied Elasticity," chapter xiii.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[5] See Riemann-Weber, "Differentialgleichungen der Physik," vol. ii.

$$\frac{dF}{dy} = 0, \quad F = 0 \text{ at the lower boundary, } y = -h$$

As in Fig. 1, the moment $M(x)$ is assumed to be zero except in the region $x = a_1$ to $x = a_2$, and we now express $M(x)$ as a Fourier integral:[6]

$$M_x = \int_0^\infty (a \cos mx + b \sin mx)\, dm$$

where a and b are determined by

$$a = \frac{1}{\pi}\int_{a_1}^{a_2} M(\alpha)\cos m\alpha d\alpha$$

$$b = \frac{1}{\pi}\int_{a_1}^{a_2} M(\alpha)\sin m\alpha d\alpha$$

We now form the four boundary equations which must hold for all values of x. Hence, by equating to zero the coefficients of $\cos mx$ and $\sin mx$ we obtain eight other equations which determine the coefficients A_1, etc. (later in the paper this process is carried out in detail), and by the substitution of these in [3] we finally obtain the stress function F. The stresses are then found by employing [1].

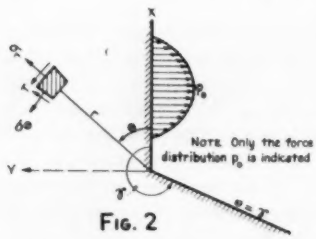


FIG. 2

In other words, if it is possible to represent the moment distribution $M(x)$ as a Fourier integral, the solution of the stress problem is immediately obtained.

Von Kármán and F. Seewald have applied this method to several important cases and obtained interesting results, among others the "corrected" relation between the bending moment and shear at a point of a beam and the curvature of the central line

$$K = \frac{M}{EI} + \left[\frac{6}{5G} - \frac{9\mu}{10E}\right]\frac{P}{A} \quad \ldots\ldots\ldots\ldots [4]$$

where K denotes the curvature; E and G, Young's modulus and the corresponding modulus in shear; μ, Poisson's ratio; A, area of cross-section; I, moment of inertia; and M and P, the bending moment and shear at the point in question. The second term corresponds to the influence of shear on the deflection.

A new and interesting conclusion of this theory is that the deflection of the central line of a long beam is wave shaped outside the loaded region.

Generalization. The method just described can be applied generally, not only to the half-plane, but to any angular portion (γ) of the plane with any distribution of normal and tangential forces over the straight boundaries.

Introducing polar coordinates the stress function must satisfy the equation

$$\nabla^4 F(r,\theta) = \left(\frac{\partial^2}{\partial r^2} + \frac{1}{r}\frac{\partial}{\partial r} + \frac{\partial^2}{r^2\partial\theta^2}\right)^2 F = 0 \quad \ldots\ldots [5]$$

with corresponding stresses

$$\sigma_r = \frac{\partial^2 F}{r^2\partial\theta^2} + \frac{1}{r}\frac{\partial F}{\partial r} \quad \ldots\ldots\ldots\ldots [6]$$

$$\sigma_\theta = \frac{\partial^2 F}{\partial r^2} \quad \ldots\ldots\ldots\ldots [7]$$

$$\tau = -\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial F}{\partial\theta}\right) \quad \ldots\ldots\ldots\ldots [8]$$

(See Fig. 2.)

The general solution of [5] is the real and/or imaginary parts of the complex function:

$$\varphi(z) = Af(z) + Bxg(z) + Cyh(z) \quad \ldots\ldots\ldots\ldots [9]$$

where $z = x + iy = r(\cos\theta + i\sin\theta) = re^{i\theta}$. A, B, and C are arbitrary complex constants; $f(z)$, $g(z)$, and $h(z)$ are arbitrary analytic functions.

For the present purpose the author will employ the functions obtained by taking real or imaginary parts of

$$\varphi_n = \frac{1}{2}A_n \begin{pmatrix} x \\ \text{and} \\ y \end{pmatrix} (z^{+im} \pm z^{-im}) \quad \ldots\ldots\ldots\ldots [10]$$

where m is a real and A_n an arbitrary constant to be determined by the boundary conditions.

Let $\xi = \log r$, and let $f(z)_r$ denote the real part of $f(z)$ or $Rf(z)$, and $f(z)_i$ the imaginary part of $f(z)$ or $If(z)$; then we obtain as particular solutions of $\nabla^4 F = 0$:

$$\left.\begin{aligned}
\varphi_1 &= \frac{1}{2}A_1 x(z^{im} + z^{-im})_r = +A_1 x\cos m\xi\cosh m\theta \\
\varphi_2 &= \frac{1}{2}A_2 x(z^{im} - z^{-im})_r = -A_2 x\cos m\xi\sinh m\theta \\
\varphi_3 &= \frac{1}{2}A_3 x(z^{im} + z^{-im})_i = -A_3 x\sin m\xi\sinh m\theta \\
\varphi_4 &= \frac{1}{2}A_4 x(z^{im} - z^{-im})_i = +A_4 x\sin m\xi\cosh m\theta \\
\varphi_5 &= \frac{1}{2}A_5 y(z^{im} + z^{-im})_r = +A_5 y\cos m\xi\cosh m\theta \\
\varphi_6 &= \frac{1}{2}A_6 y(z^{im} - z^{-im})_r = -A_6 y\cos m\xi\sinh m\theta \\
\varphi_7 &= \frac{1}{2}A_6 y(z^{im} + z^{-im})_i = -A_7 y\sin m\xi\sinh m\theta \\
\varphi_8 &= \frac{1}{2}A_8 y(z^{im} - z^{-im})_i = +A_8 y\sin m\xi\cosh m\theta
\end{aligned}\right\} \quad \ldots\ldots [11]$$

The coefficients A_n are now defined as functions of the parameter m. The stress function may then be expressed as

$$F = \sum_{n=1}^{n=8}\int_0^\infty \varphi_n dm = \int_0^\infty \left[\varphi_1 + \varphi_2 \ldots\ldots + \varphi_8\right] dm \quad \ldots\ldots [12]$$

from which the stresses are obtained by [6], etc.:

$$\sigma_r = \sum_{n=1}^{n=8}\int_0^\infty \left[\frac{\partial^2\varphi_n}{r^2\partial\theta^2} + \frac{1}{r}\frac{\partial\varphi_n}{\partial r}\right] dm \quad \ldots\ldots [13]$$

$$\sigma_\theta = \sum_{1}^{8}\int_0^\infty \frac{\partial^2\varphi_n}{\partial r^2}\, dm \quad \ldots\ldots\ldots\ldots [14]$$

[6] For explanation, see "Fourier Series," by W. E. Byerly.

$$\tau = -\sum_{1}^{8}\int_0^\infty \frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial \varphi_n}{\partial \theta}\right) dm \ldots\ldots\ldots [15]$$

Two typical cases of boundary loads must be considered:

(*a*) The load may be continuous (not necessarily uniform) over portions of the straight boundaries. Let it be assumed that the load is distributed over finite portions of the boundaries or, if the load extends to infinity, that the load per unit length becomes zero at infinity. Let the boundaries be the lines $\theta = 0$ and $\theta = \gamma$, and let the normal loads be, respectively, p_0 and p_γ and the tangential loads q_0 and q_γ. These quantities are functions of r, measured from the intersection of the boundaries, or functions of $\xi = \log r$. Using Fourier's double integrals we can write $p_0(\xi)$ in the following form:

$$p_0(\xi) = \frac{1}{\pi}\int_0^\infty \left[\cos m\xi \int_{\xi_1}^{\xi_2} p_0(\alpha)\cos m\alpha d\alpha + \sin m\xi \int_{\xi_1}^{\xi_2} p_0(\alpha)\sin m\alpha d\alpha\right] dm \ldots\ldots [16][7]$$

where the load $p_0(\xi)$ is assumed to exist only between the points ξ_1 and ξ_2. Similar expressions are obtained for p_γ, q_0, and q_γ.

Therefore the boundary conditions are

$$(\sigma_\theta)_{\theta=0} = p_0(\xi) \text{ expressed as in [16]} \ldots\ldots [17]$$

$$(\tau)_{\theta=0} = q_0 \ldots\ldots [18]$$

$$(\sigma_\theta)_{\theta=\gamma} = p_\gamma \ldots\ldots [19]$$

$$(\tau)_{\theta=\gamma} = q_\gamma \ldots\ldots [20]$$

where σ and τ are given by [14] and [15].

By now equating coefficients of $\cos m\xi$ and $\sin m\xi$ in the above equations, eight other equations are obtained which determine the eight functions $A(m)$ and consequently the stress function F by substituting $A(m)$ in [12].

(*b*) In the case of concentrated loads the boundary conditions are obtained in the following manner: It has already been shown that the differences $(\partial F/\partial x)_B - (\partial F/\partial x)_A$ and $(\partial F/\partial y)_B - (\partial F/\partial y)_A$ represent, respectively, the Y- and X-components of all the boundary forces between A and B. Therefore in the case of concentrated forces these differences become discontinuous (similar to the shear polygon for concentrated forces) and the jump in these values at a certain point is equal to the respective components of the concentrated force at that point. Consider for instance unit normal force acting at point $r = a$ of the boundary $\theta = 0$. Obviously

$$\left(\frac{\partial F}{\partial r}\right)_{a+\epsilon} - \left(\frac{\partial F}{\partial r}\right)_{a-\epsilon} = 1$$

where ϵ is a small positive quantity.

If there is no continuously distributed load on the boundary $\theta = 0$, then, by [17],

$$(\sigma_\theta)_{\theta=0} = \left(\frac{\partial^2 F}{\partial r^2}\right)_{\theta=0} = 0$$

Hence $\partial F/\partial r$ = constant on either side of $r = a$. These conditions can all be satisfied by writing

$$\frac{\partial F}{\partial r} = \frac{1}{\pi}\int_0^\infty \frac{\sin\left(m\log\frac{r}{a}\right)}{m} dm \ldots\ldots [21][8]$$

[7] See "Fourier Series," by W. E. Byerly.

[8] See Peirce, "Short Tables of Integrals," p. 484.

This integral has the value

$$-\frac{1}{2} \text{ for } \xi = \log(r/a) < 0 \quad \text{or} \quad r < a$$

$$+\frac{1}{2} \text{ for } \xi = \log(r/a) \gtrless 0 \quad \text{or} \quad r \gtrless a$$

The integral[9]

$$\frac{1}{\pi}\int_0^\infty \frac{n\cos m\xi - m\sin m\xi}{m^2+n^2} = \begin{cases} e^{n\xi} \text{ when } \xi < 0 \\ \frac{1}{2} \text{ when } \xi = 0 \\ 0 \text{ when } \xi > 0 \end{cases}$$

with its ramifications is often very useful in expressing boundary functions in Fourier integral form.

As an application of the foregoing, the stress function F corresponding to a unit normal traction at point $r = a$ of boundary $\theta = 0$ will be found for a reentrant right corner, i.e., $\gamma = 3\pi/2$. (See Fig. 3.)

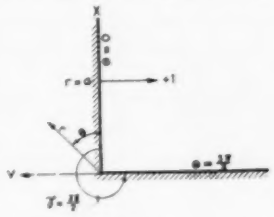


FIG. 3

The boundary conditions are:

$$\theta = 0 \begin{cases} \frac{\partial F}{\partial r} = 0 & \text{when } \xi = \log(r/a) < 0; \\ \frac{\partial F}{\partial r} = 1 & \text{when } \xi > 0, \\ \frac{\partial F}{\partial \theta} = 0 \end{cases} \qquad \theta = \gamma \begin{cases} \frac{\partial F}{\partial \theta} = 0 \\ F = 0 \end{cases}$$

Let $F = F_0 + F_1$. F_0 is determined by the boundary conditions

$$\left(\frac{\partial F_0}{\partial r}\right)_{\theta=0} = \begin{cases} -\frac{1}{2} \text{ for } \xi < 0 \\ +\frac{1}{2} \text{ for } \xi > 0 \end{cases}$$

Hence $\dfrac{\partial F_0}{\partial r} = \dfrac{1}{\pi}\displaystyle\int_0^\infty \frac{\sin m\xi}{m}\partial m$, by [21],

$$\left(\frac{\partial F_0}{\partial \theta}\right)_{\theta=0} = 0; \quad (F_0)_{\theta=\gamma} = 0; \quad \left(\frac{\partial F_0}{\partial \theta}\right)_{\theta=\gamma} = 0$$

In order to remove the concentrated force introduced at the origin (due to the discontinuity in $(\partial F/\partial x)$, F_1 is determined so that

$$\left(\frac{\partial F_1}{\partial r}\right)_{\theta=0} = \frac{1}{2} \text{ for } -\infty < \xi < +\infty$$

$$\left(\frac{\partial F_1}{\partial \theta}\right)_{\theta=0} = 0; \quad (F_1)_{\theta=\gamma} = 0; \quad \left(\frac{\partial F_1}{\partial \theta}\right)_{\theta=\gamma} = 0$$

F_1 is easily found by employing the simple solution to $\nabla^4 F_1 = 0$:

$$F_1 = A\theta r\cos\theta + B\theta r\sin\theta + Cr\cos\theta + Dr\sin\theta \ldots [22a]$$

Hence, by the boundary conditions for F_1:

$$\left(\frac{\partial F_1}{\partial r}\right)_{\theta=0} = C = \frac{1}{2}$$

[9] Riemann-Weber, "Differentialgleichungen der Physik," vol. I, p. 157.

$$\left(\frac{\partial F_1}{\partial \theta}\right)_{\theta=0} = Ar + Dr = 0$$

$$(F_1)_{\theta=\frac{3\pi}{2}} = -\frac{3\pi}{2} Br - Dr = 0$$

$$\left(\frac{\partial F_1}{\partial \theta}\right)_{\theta=\frac{3\pi}{2}} = \frac{3\pi}{2} Ar - Br + Cr = 0$$

Solving these and substituting in [22a],

$$F_1 = \frac{3\pi}{4-9\pi^2}\theta r\cos\theta + \frac{2}{4-9\pi^2}\theta r\sin\theta \quad \ldots\ldots [22b]$$

The first-degree terms have been omitted as they contribute no stress.

In order to determine F_0 we employ Equation [12] and formulate the given boundary conditions by differentiating the functions φ given by [11] and obtain:

$$\left(\frac{\partial F_0}{\partial r}\right)_{\theta=0} = \int_0^\infty \Big[A_1(\cos m\xi - m\sin m\xi) + A_4(m\cos m\xi + \sin m\xi)\Big]\,dm = \frac{1}{\pi}\int_0^\infty \frac{\sin m\xi}{m}\,dm$$

$$\left(\frac{\partial F_0}{r\partial \theta}\right)_{\theta=0} = \int_0^\infty \Big[-mA_2\cos m\xi - mA_3\sin m\xi + A_5\cos m\xi + A_8\sin m\xi\Big]\,dm = 0$$

$$(F_0)_{\theta=\gamma} = \int_0^\infty \Big[-A_5\cosh\gamma m\cos m\xi + A_6\sinh\gamma m\cos m\xi + A_7\sinh\gamma m\sin m\xi - A_8\cosh\gamma m\sin m\xi\Big]\,dm = 0$$

$$\left(\frac{\partial F_0}{r\partial \theta}\right)_{\theta=\gamma} = \int_0^\infty \Big[A_1\cosh\gamma m\cos m\xi - A_2\sinh\gamma m\cos m\xi - A_3\sinh\gamma m\sin m\xi + A_4\cosh\gamma m\sin m\xi - mA_5\sinh\gamma m\cos m\xi + mA_6\cosh\gamma m\cos m\xi + mA_7\cosh\gamma m\sin m\xi - mA_8\sinh\gamma m\sin m\xi\Big]\,dm = 0$$

where $\xi = \log(r/a)$ and $\gamma = 3\pi/2$.

By equating coefficients of $\cos m\xi$ and $\sin m\xi$:

$$A_1 + mA_4 = 0; \quad -mA_1 + A_4 = 1/m\pi; \quad -mA_2 + A_5 = 0; \quad -mA_3 + A_8 = 0$$

$$A_5\cosh\gamma m - A_6\sinh\gamma m = 0$$

$$A_7\sinh\gamma m - A_8\cosh\gamma m = 0$$

$$A_1\cosh\gamma m - A_2\sinh\gamma m - mA_5\sinh\gamma m + mA_6\cosh\gamma m = 0$$

$$-A_3\sinh\gamma m + A_4\cosh\gamma m + mA_7\cosh\gamma m - mA_8\sinh\gamma m = 0$$

and solving for A_n:

$$A_1 = -\frac{1}{\pi}\frac{1}{1+m^2}$$

$$A_2 = +\frac{1}{\pi}\frac{\cosh\gamma m\sinh\gamma m}{(1+m^2)(m^2-\sinh^2\gamma m)}$$

$$A_3 = -\frac{1}{\pi}\frac{\sinh\gamma m\cosh\gamma m}{m(1+m^2)(m^2-\sinh^2\gamma m)}$$

$$A_4 = +\frac{1}{\pi}\frac{1}{m(1+m^2)}$$

$$A_5 = +\frac{1}{\pi}\frac{m\cosh\gamma m\sinh\gamma m}{(1+m^2)(m^2-\sinh^2\gamma m)}$$

$$A_6 = +\frac{1}{\pi}\frac{m\cosh^2\gamma m}{(1+m^2)(m^2-\sinh^2\gamma m)}$$

$$A_7 = -\frac{1}{\pi}\frac{\cosh^2\gamma m}{(1+m^2)(m^2-\sinh^2\gamma m)}$$

$$A_8 = -\frac{1}{\pi}\frac{\sinh\gamma m\cosh\gamma m}{(1+m^2)(m^2-\sinh^2\gamma m)}$$

The stress function is then found simply by substituting the A's back into Equation [12] and the stresses are found from Equations [13], [14], and [15].

The stress function in the $^3/_4$-plane for unit tensile force concentrated at point $(r,\theta) = (a,0)$ after some reduction becomes: $F = F_0 + F_1 =$

$$F = \frac{1}{\pi}\int_0^\infty \frac{[xm^2 - (x\sinh\gamma m + ym\cosh\gamma m)\sinh m(\gamma-\theta)]\,(m\cos m\xi - \sin m\xi)}{m(1+m^2)(\sinh^2\gamma m - m^2)}\,dm + \frac{3\pi}{4-9\pi^2}\theta r\cos\theta + \frac{2}{4-9\pi^2}\theta r\sin\theta \quad \ldots\ldots [23]$$

where $\gamma = {}^3/_2\pi$; $\xi = \log(r/a)$; $x = r\cos\theta$; $y = r\sin\theta$.

The integral F may be evaluated by the method of contour integration in the complex plane,[10] making use of Cauchy's theorem of residues as outlined in the Appendix.

The stress functions valid for small and large values of r/a become: $F = F_0 + F_1 =$

$$F = a[0.101\,(r/a)^{1.545}\,g(\theta) + 0.125\,(r/a)^{1.909}\,h(\theta)] \quad \text{when } r/a \ll 1 \quad \ldots\ldots [24]$$

$$F = a\left[-0.340\,(r/a)^{0.455}\,g(\theta) - 2.629\,(r/a)^{0.091}\,h(\theta) + \frac{r}{a}\frac{6\pi}{4-9\pi^2}\theta\cos\theta + \frac{r}{a}\frac{4}{4-9\pi^2}\theta\sin\theta - \cos^2\theta\right] \quad \text{when } r/a \gg 1 \quad \ldots\ldots [25]$$

where

$$g(\theta) = \sin 0.545\,\theta\cos\theta - 0.545\cos 0.545\,\theta\sin\theta - 0.839\sin 0.545\,\theta\sin\theta$$

$$h(\theta) = \sin 0.909\,\theta\cos\theta - 0.909\cos 0.909\,\theta\sin\theta + 0.416\sin 0.909\,\theta\sin\theta$$

It is interesting to notice that the terms in [24], including those of higher order, are the biharmonics (solutions of $\nabla^4 F = 0$) which correspond to boundaries free of forces. This can also be seen by writing

$$F = Ar^n\cos n\theta + Br^n\cos(n-2)\theta + Cr^n\sin n\theta + Dr^n\sin(n-2)\theta \quad \ldots\ldots [26]$$

where n may be complex and finding the values of n which give possible solutions in case F and $\partial F/\partial\theta$ are both to be zero along both boundaries. The determinant of the four boundary equations gives exactly the roots corresponding to the exponents in [24]. It is, however, only possible to determine the ratios B/A, etc. in this way. The absolute values of the constants

[10] For explanation see MacRobert, "Functions of a Complex Variable."

naturally depend upon the location and magnitude of the concentrated forces, and these can only be specified by the use of Fourier integrals as above.

The writer is informed that Dr. H. M. Westergaard, professor of theoretical and applied mechanics, University of Illinois, in a communication to Dr. von Kármán, has pointed out the existence of these solutions, presumably obtained in the manner just described.

Check on boundary conditions:
By [24] and [25] it is easily seen that

$$\left.\begin{matrix} F = 0 \\ \dfrac{\partial F}{\partial \theta} = 0 \end{matrix}\right\} \text{ when } \theta = 0 \text{ and } \theta = \frac{3\pi}{2}$$

are satisfied. (See Fig. 4.)

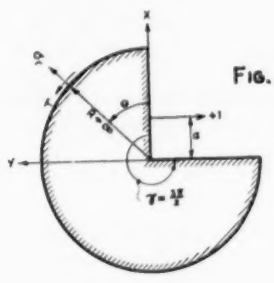


FIG. 4

In order to investigate if all boundary forces are in equilibrium we shall determine the stresses on a circle of large radius $r = R \to \infty$. By differentiating [25] and taking only such terms that contribute,

$$\sigma_r = -\frac{6\pi}{4 - 9\pi^2}\frac{2\sin\theta}{R} + \frac{4}{4 - 9\pi^2}\frac{2\cos\theta}{R}$$

$$\tau = \frac{a\sin 2\theta}{R^2}$$

Projecting on the X-axis and letting $R \to \infty$:

$$\Sigma X = \int_0^{\frac{3\pi}{2}} r\sigma_r \cos\theta\, d\theta = \left[\frac{3\pi}{4 - 9\pi^2}\cos 2\theta + \frac{4}{4 - 9\pi^2}\left(\theta + \frac{1}{2}\sin 2\theta\right)\right]_0^{\frac{3\pi}{2}} = 0$$

Projecting on the Y-axis:

$$\Sigma Y = \int_0^{\frac{3\pi}{2}} r\sigma_r \sin\theta\, d\theta = \left[-\frac{6\pi}{4 - 9\pi^2}\left(\theta - \frac{1}{2}\sin 2\theta\right) - \frac{2}{4 - 9\pi^2}\cos 2\theta\right]_0^{\frac{3\pi}{2}} = +1$$

Moments about the origin:

$$M_0 = \int_0^{\frac{3\pi}{2}} \tau r^2\, d\theta = -\left[\frac{a\cos 2\theta}{2}\right]_0^{\frac{3\pi}{2}} = +a$$

It will be seen that all equilibrium conditions are satisfied, the only other force being the unit tensile force at a. The stresses for large and small radii are now obtained from [24] and [25] by application of Equations [6], [7], and [8]. It will be noticed that stresses become infinite at the origin.

Evaluation of Stresses

If it is only desired to know the stresses at certain points or the distribution on a line, it is often simpler to specialize F (see [23]) before evaluating the integral. For example, to obtain the distribution of normal stress on the Y-axis, place $\theta = \frac{\pi}{2}$ in [23]:

$$F_{\theta=\pi/2} = \frac{r}{\pi}\int_0^\infty \frac{\cosh\gamma m \sinh\pi m}{(1 + m^2)(\sinh^2\gamma m - m^2)} \times (m\cos m\xi - \sin m\xi)dm + F_1 \quad [27]$$

$$\sigma_{\theta=\pi/2} = \frac{\partial^2 F}{\partial r^2} = \frac{1}{\pi r}\int_0^\infty \frac{m\cosh\gamma m \sinh\pi m}{\sinh^2\gamma m - m^2}\cos m\xi\, dm \quad [28]$$

Notice that F_1 contributes nothing to $\sigma_{\theta=\pi/2}$.

By [24] and [25] we obtain the correct stresses for small and large radii which usually are the most interesting, and only a few additional terms of higher powers in r would be needed to give the stresses in the region $r = a$. These can be obtained by the method of residues applied to [28] as explained in the Appendix. However, if only a plot of the stresses is desired, it is simpler to evaluate [28] for values of r in the neighborhood of a, by more primitive methods. The simplest of these would perhaps be to plot the integrand for various values of m and r and apply Simpson's rule. This need only be done for $m \leq 1$ because for larger values of m, [28] can be further simplified by placing

$\cosh\gamma m = \sinh\gamma m = \varepsilon^{\frac{3\pi m}{2}}$ and ignoring m^2 in the denominator. The integral then becomes

$$\frac{1}{\pi r}\int_0^\infty m\varepsilon^{-\frac{\pi m}{2}}\cos m\xi\, dm$$

The normal stresses on the line $\theta = \pi/2$ for large and small r/a are easily found by [24] and [25]:

$$\sigma_{\theta=\pi/2} = \frac{d^2F}{dr^2} = \frac{1}{a}\left[-0.084\left(\frac{r}{a}\right)^{-0.455} + 0.062\left(\frac{r}{a}\right)^{-0.091}\right]; \quad r \ll a \quad [29]$$

$$\sigma_{\theta=\pi/2} = \frac{d^2F}{dr^2} = \frac{1}{a}\left[-0.084\left(\frac{r}{a}\right)^{0.455} + 0.062\left(\frac{r}{a}\right)^{0.091}\right]\left(\frac{r}{a}\right)^{-2}; \quad r \ll a \quad [30]$$

The stresses $\sigma_{\theta=\pi/2}$ obtained by evaluating [28], [29], and [30] are plotted in Fig. 5. The corresponding stresses for the half-plane, $\gamma = \pi$, are also shown. It is interesting to notice the effect of the additional quarter-plane by comparing the two stress distributions, I and II.

If several concentrated forces are applied on the boundary $\theta = 0$, the stress functions and stresses are obtained by superposing the functions computed for the various values of a. For a concentrated force acting on the boundary $\theta = 3\pi/2$, the stresses are obtained from the present formulas by substituting $(3\pi/2) - \theta$ for θ, maintaining the same coordinate system.

It is of interest to note that the function $g(\theta)$ in [24] is symmetric and $h(\theta)$ is antisymmetric about the bisector of the angle γ. This holds also for the corresponding functions for higher powers in r/a. This would mean that concentrated forces of equal magnitude located on each boundary at equal distance from the vertex would have stress functions (valid near the vertex):

$$F = 0.202a(r/a)^{1.545}g(\theta) \quad [24a]$$

if the unit forces are both tension

and

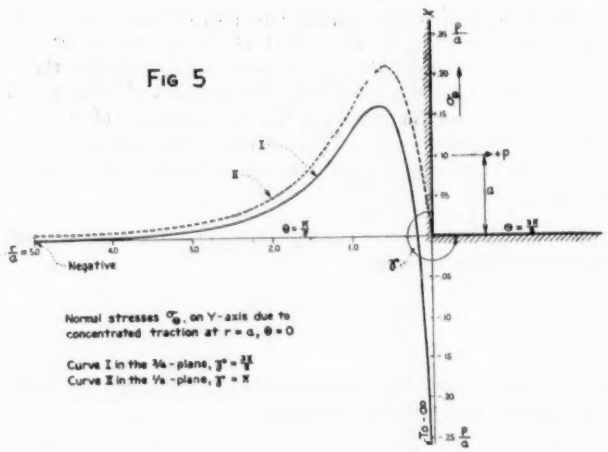
$$F = 0.250a(r/a)^{1.909}h(\theta) \quad [25a]$$

with traction on the boundary $\theta = 0$ and pressure on $\theta = 3\pi/2$. This shows that the stress at the vertex is always infinite for

partial loading of the boundaries. It is now also clear why there should be two terms producing infinite stresses.

It will be noticed that Equation [37] of the Appendix has no real roots, except $m = 0$, when γ is less than π. This means that in general a partial loading of the boundaries of a corner will produce finite stresses at the origin (vertex) if $\gamma < \pi$, but infinite stresses if $\gamma > \pi$, and the greater γ is, the sharper the infinity. The half-plane, $\gamma = \pi$, thus is the critical case, and the stresses at the origin are zero if it is chosen outside the loaded region.

Special cases of stress distribution in an infinite wedge have been treated by a number of investigators. We find J. H.



Michell who generalizes Boussinesq's solution for a concentrated force on the half-plane to a concentrated force at the vertex of an infinite wedge, M. Levy[11] who finds a solution for a linear load distribution over the boundaries, S. D. Carothers[12] who gives solutions for a couple at the vertex and uniform loads on the boundaries.

The stress function for continuous uniform pressure on the boundaries is easily determined by Equation [26]:

$$F = -\frac{p}{2} r^2 \cos^2 \theta \ldots\ldots\ldots\ldots\ldots [31]$$

corresponds to uniform pressure on the boundary $\theta = 0$. From this it will be seen that the stresses are finite at the vertex for any value of γ. We have seen that a partial loading produces infinite stress at the vertex if $\gamma > \pi$, so that we have the curious result that in this case the removal of a portion of the load raises the stress to infinity.

If uniform pressure acts on the boundary $\theta = 3\pi/2$ (as in the case of a dam with a full reservoir), the corresponding stress function is

$$F = -\frac{p}{2} r^2 \sin^2 \theta \ldots\ldots\ldots\ldots\ldots [32]$$

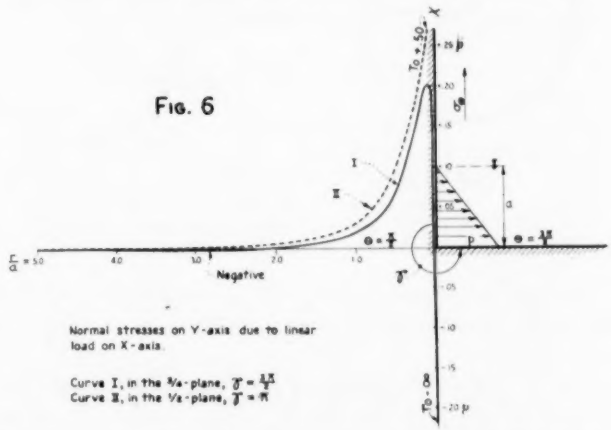
This is important in the design of dams, as it shows that the base will be under compression due to this load. In Fig. 6 is plotted the normal stress on the base, $\theta = \pi/2$, due to linear pressure distribution on the boundary, $\theta = 0$. It is important to notice that near the vertex the base is under high tension, i.e., opposite to the stress due to the uniform pressure on the boundary, $\theta = 3\pi/2$.

Levy's solution for linear normal pressure on the boundary of an infinite wedge (or triangle) shows linear distribution of all stresses along any straight line through the wedge. In practice this result is conveniently applied to masonry dams. However, the application is dubious because the height of a dam is of the same order of magnitude as the base width, so that great deviations from the linear stress distribution can be expected (at any rate near the base) due to the influence of the unyielding, or rather, infinite, foundation.

Engineers are well aware of this fact, and the present study grew out of an attempt to determine the stress distribution in triangular dams on infinite, rigid, and elastic foundations by the Ritz method. However, it was found that the stresses in the up- and down-stream faces at the base did not converge when more parameters were included in the solution. At points only slight distant from the up- and down stream corners Ritz's method gave reasonable results, which will be published in the near future.

The present method enables us to compute the stresses in dams of triangular and other shapes for any load, and far more accurate solutions can be reached in this way than by any previously employed.

Photoelastic experiments are at present being carried out at the California Institute by the author and Mr. Ralph M. Watson



for the purpose of verifying the theoretical results of this study. It is of special interest to find out how closely the equation $\nabla^4 F = 0$ holds for plates of finite thickness. The stress distribution is also being investigated in plates of arbitrary shapes, especially applicable to masonry dams, as also are the stress concentrations around openings corresponding to the inspection and drainage galleries which usually exist in high dams.

Appendix

Cauchy's theorem of residues

$$\int_S \psi(m)dm = 2\pi i \sum \text{Residues}$$

enables us to compute the definite integral in [23]

$$m \text{ is complex } = \alpha + i\beta$$

The residues are found by integrating around each pole in the positive direction.

F_0 may be written as

$$F_0 = \frac{1}{2\pi}\int_0^\infty \frac{mf(m)\cos m\xi\, dm}{m(1+m^2)(\sinh^2 \gamma m - m^2)} - \frac{1}{2\pi}\int_0^\infty \frac{f(m)\sin m\xi dm}{m(1+m^2)(\sinh^2 \gamma m - m^2)} = M_1 - M_2 \ldots [33]$$

[11] A. E. H. Love, "Elasticity," p. 212.
[12] Proc. Roy. Soc., Edinburgh, vol. 53 (1913), p. 292.

where the meaning of $f(m)$ is evident by comparing with Equation [23].

Consider the complex integrals:

$$G + iH = \frac{1}{2\pi}\int_S \frac{e^{im}f(m)dm}{m(1+m^2)(\sinh^2\gamma m - m^2)} \quad [34]$$

and

$$K + iL = \frac{1}{2\pi}\int_S \frac{m^{im}f(m)dm}{m(1+m^2)(\sinh^2\gamma m - m^2)} \quad [35]$$

taken along the contours S consisting of the real axis and the infinite half-circle above or below the real axis.

If $\xi = \log r/a < 0$ the half-circle below the real axis must be chosen because in this case the integral along this part of the path converges to zero as $|m| \to \infty$. For the same reason the upper infinite half-circle is chosen when $\xi > 0$, i.e., $r > a$. Consider, for example, the lower half-plane:

$$m = \alpha - i\beta$$

where β is real and positive; therefore, if $0 > \xi = -c$, where c is real and positive,

$$e^{im\xi} = e^{i(\alpha - i\beta)(-c)} = e^{-i\alpha c}e^{-c\beta} = \frac{\cos\alpha c - i\sin\alpha c}{e^{c\beta}}$$

which becomes zero in the limit, $\beta \to \infty$.

The singularities of the integrand are all simple poles and are the roots of:

$$\sinh^2\gamma m - m^2 = 0 \quad [36]$$

or if m is pure imaginary $= i\beta$

$$\beta = \pm\sin(\gamma\beta) \quad [37]$$

The complex roots may be found with very good approximation (except $n = 2$) by:

$$m = \alpha_n + i\beta_n = \pm\frac{1}{\gamma}\log\frac{\pi(2n+1) - \epsilon}{\gamma} \pm i\frac{\pi(2n+1) - \epsilon}{2\gamma} \quad [38]$$

where

$$\epsilon = \frac{4\log\dfrac{\pi(2n+1)}{\gamma}}{\pi(2n+1)} \quad [39]$$

and $n = 2, 3, 4, 5 \ldots$

$$\gamma = \frac{3\pi}{2}$$

For large values of n:

$$m = \alpha_n + i\beta_n = \pm\frac{2}{3\pi}\log\frac{4n+2}{3} \pm i\frac{2n+1}{3} \quad [40]$$

The lower roots are:

α	β
0	$\pm i0.539$
0	$\pm i0.909$
0	$\pm i$
± 0.231	± 1.629, corresponds to $n = 2$ in [38].

For $\alpha_{n=3}$, etc., use Equations [38] and [39].

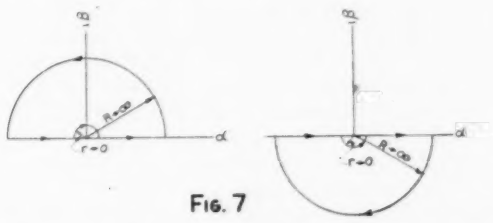


FIG. 7

It will be seen that for $|m| \leq 1$ the roots are located on the imaginary axis, i.e., are purely imaginary. For $|m| > 1$ all the roots are located near the imaginary axis.

Now returning to [34] and [35], integrating in the positive directions and observing [33], we find for: $\xi > 0$, i.e., $r > a$:

$$M_2 = H = \mathbf{I}\frac{1}{2\pi}\int_S \frac{e^{im\xi}f(m)dm}{m(1+m^2)(\sinh^2\gamma m - m^2)} = [i\,\Sigma\,\text{Residues}]_i = [+\,\Sigma R_2]_r \quad [41]$$

$$M_1 = K = \mathbf{R}\frac{1}{2\pi}\int_S \frac{me^{im\xi}f(m)dm}{m(1+m^2)(\sinh^2\gamma m - m^2)} = [i\,\Sigma R_1]_r = [-\,\Sigma R_1]_i \quad [42]$$

Due to $m = 0$ being a simple pole of the integrand in [41], the contour must be indented and half the residue at the origin must be included in ΣR_2. In [42], $m = 0$ is not a pole, so no indentation is necessary.

It should be noted that $R_1 = mR_2$, which facilitates the computations greatly.

For $\xi < 0$, i.e., $r < a$, and now integrating in negative direction:

$$M_{(-2)} = H = [-i\Sigma\,\text{Residues}]_i = [-\Sigma R_{(-2)}]_r \quad [43]$$

$$M_{(-1)} = K = [-i\Sigma\,\text{Residues}]_r = [+\,\Sigma R_{(-1)}]_i \quad [44]$$

The same remarks as above hold here and $R_{(-1)} = mR_{(-2)}$, where the subscript indicates integration in the negative half-plane.

The convergence of the power series is rapid, and even for values of r near a, only a few of the poles need be considered outside the unit circle $|m| = 1$. For very large $|\xi|$ only the poles inside the circle $|m| = 1$ contribute.

The stress function valid for small and large values of r/a was obtained by this method and is given in Equations [24] and [25] of the paper.

APM-55-7

Concentration of Stress Around Spherical and Cylindrical Inclusions and Flaws

By J. N. GOODIER,[1] TORONTO, ONT., CANADA

Solutions of the equation of elasticity are applied to investigate the disturbing effect of small spherical and cylindrical inclusions on an otherwise uniform stress distribution. Numerical results are found for gaseous inclusions (flaws), perfectly rigid inclusions, and for systems roughly representing slag globules in steel and reinforcing rods in concrete.



THE determination of the effects of holes and cavities, or flaws, in the elastic solid when the stress at points remote from the flaw is uniform, has been achieved for holes in the form of circular and elliptical cylinders,[2] and for cavities of spherical form.[3] These problems are special cases of the more general one involving a solid inclusion instead of a hole or cavity. Solutions for circularly cylindrical and spherical inclusions are derived in this paper. They yield confirmation of the known results for flaws, and afford a means of discussing such questions as the effects of foreign inclusions in metal, and the influence of a reinforcing rod in concrete on tensions perpendicular to its length.

For the purposes of calculation it is assumed at first that the solid surrounding the inclusion is infinite in extent, and subjected to uniform applied stress at infinity, besides having the ideal properties of elasticity, isotropy, and homogeneity. It is well, therefore, to lay down at once the considerations underlying the application of the results to finite bodies of actual material.

We know beforehand, from Saint Venant's principle, that the disturbance will be confined to the neighborhood of the inclusion. It appears in fact, from the solutions themselves, that at a distance of some four diameters away from the inclusion the stress distribution, which would be uniform without the inclusion, is in no case modified by more than about one per cent. In consequence the results are valid for inclusions which are, say, four diameters and more away from a boundary, or other inclusions or flaws. They are also valid, for the same reason, in regions where the stress, but for the inclusion, would not vary appreciably over distances measuring several diameters of the inclusion. In general this will be so if the inclusion is small in comparison with the ruling dimensions of the body in which it occurs. On the other hand, in seeking to interpret the behavior of actual solids, no conclusions can be taken over from the theory of elasticity unless it is legitimate to regard the material as *statistically* isotropic and homogeneous, as it is only if the inclusion is very large in comparison with the grain size of the crystalline aggregate. Then, provided the proportional limit be not exceeded, the "stress at a point" indicated by the theory is an estimate of the average stress on an area comprising a great number of crystals, and this average is the "stress" measured by the experimental methods of elasticity, and commonly referred to in engineering discussion. Its significance in relation to the initiation and propagation of fatigue cracks is discussed by Southwell and Gough.[4] Once started, the general direction of the crack follows the surface on which the greatest tensile stress (in the sense defined above) occurs.

In all the results obtained, the factor of stress concentration—the ratio of the maximum stress to the uniform stress at a distance—is independent of the absolute size of the inclusion. This is not peculiar to the spherical and cylindrical boundaries to which the detailed theory is confined, but is necessarily true for any given form of inclusion boundary, for dimensional reasons. We may then state the result thus: The intensification of applied stress due to a small inclusion not near a boundary depends on its geometrical form but not on its absolute size. The limitations just described apply to this result, since the dimensional argument by which it is demonstrated below depends on the theory of elasticity.

The solutions for spherical and cylindrical cavities and holes have been employed to elucidate the phenomenon of spiral fracture in shafts subject to alternating torsion.[5] They offer also an explanation of the manner of occurrence of fatigue cracks in weld metal,[6] which contains gaseous inclusions (blowholes) of spheroidal form and has a surface pitted with "pinholes."[7] The solutions for solid inclusions may be of interest in the study of non-metallic inclusions in metals. The effects of such inclusions, as measured by fatigue strength, are not very definite. There are instances where the endurance limit is only about one-half of, and others where it is somewhat higher than, that of the clean metal. As was to be expected, consideration of stresses gives no indication of anything but weakening effects, due to stress concentration, from any kind of inclusion which is large compared with the individual crystals, except that a metal containing only strong solid inclusions would have smaller stress concentrations than one containing cavities.

B. P. Haigh[8] suggests that the dominant effect of the inclusions is the stress concentrations they induce. The destructive action

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[1] Ontario Research Foundation. Mr. Goodier was educated in England at Downing College, Cambridge University, where he studied mathematics and mechanical sciences. He took the degree of B.A. through the Mechanical Sciences Tripos, with the Rex Moir Prize for general distinction and the Ricardo Prize for thermodynamics. For two years he worked as a research student under Prof. C. E. Inglis, F.R.S., on problems of elasticity and applied mechanics, stress distributions, bridge vibrations, and the phenomenon of rail corrugation, and, together with H. S. Sayles, B.A., was awarded the John Steddy Winbolt Prize of Cambridge University for a dissertation on "Some Problems of Plane Stress." He was elected a Commonwealth Fellow in 1929. He then went to Ann Arbor to continue work under the supervision of Professor Timoshenko at the University of Michigan, remaining there until August, 1931, when he assumed his present duties.

[2] See "Strength of Materials," S. Timoshenko, vol. 2, pp. 616–617.

[3] R. V. Southwell and H. J. Gough, *Phil. Mag.*, Jan., 1926, p. 71.

Presented at the National Applied Mechanics Meeting, New Haven, Conn., June 23–25, 1932, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society

[4] Loc. cit.

[5] R. V. Southwell and H. J. Gough, loc. cit.

[6] Illustrated in the paper by R. E. Peterson, Proc. Am. Soc. Test. Matls., vol. 29, pt. II, p. 371.

[7] Ibid., vol. 31, pt. II, p. 194.

[8] B. P. Haigh, Trans. Faraday Soc., vol. 20 (1924–5), p. 153.

of these may be aided by two other effects, chemical embrittlement of the metal by the inclusion, and local residual stress due to different contractions of metal and inclusion in cooling. We shall see that whether the inclusion be more or less rigid than the metal, the concentration of tension arising from the different rigidities will generally augment directly any residual tension due to the different thermal contractions.

The stress concentrations for several examples of especial interest are given in the following paragraphs.

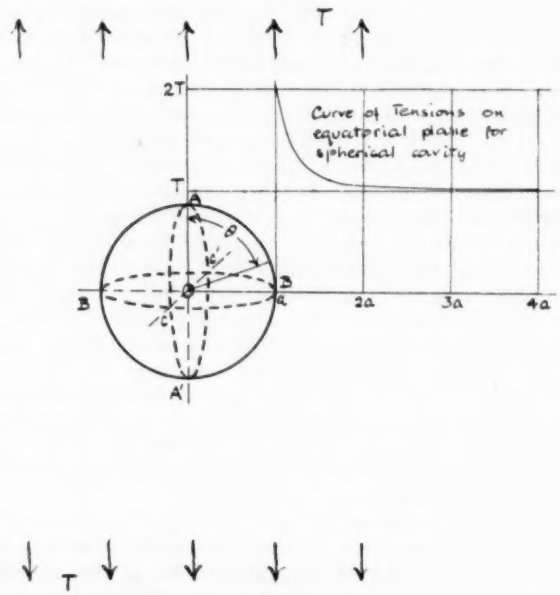


FIG. 1

CALCULATED STRESS CONCENTRATIONS

A uniform tension T, in one direction, applied at a distance from a spherical cavity produces a tensile hoop stress ($\widehat{\theta\theta}$, θ as in Fig. 1) on the "equator" $BCB'C'B$, in the same direction as the applied tension, of magnitude $\dfrac{27 - 15\sigma}{14 - 10\sigma}\,T$ (σ is Poisson's ratio) which is $^{45}/_{22}T$ for steel ($\sigma = 0.3$). The curve in Fig. 1 shows how rapidly the tension drops to the average value T within the solid. It falls away rapidly also as θ departs from 90 deg, so that the stress concentration is confined to a small belt about the "equator." There is a compressive "hoop" stress at A $\left(= -\dfrac{3 + 15\sigma}{14 - 10\sigma}\,T\right)$, and the hoop stress at the equator, at right angles to $\widehat{\theta\theta}$ ($\widehat{\psi\psi}$ in the notation of the analysis to follow) is tensile in steel, being $-\dfrac{3 - 15\sigma}{14 - 10\sigma}\,T$, or $^{3}/_{22}T$.

Combining these stresses with those due to an equal applied compressive stress at a distance, and perpendicular to the applied tension just considered, we obtain the effect of a spherical cavity in a region of pure shear T. There are two stress concentrations—a concentration of shear at C and C' (Fig. 2) of magnitude $\dfrac{15(1 - \sigma)}{7 - 5\sigma}\,T$, or $^{21}/_{11}T$ for steel—and a concentration of hoop tension at B and B' of magnitude $\dfrac{15}{7 - 5\sigma}\,T$, or $^{30}/_{11}T$ for steel. Since the endurance limit in shear is usually about one-half of the endurance limit in direct stress, it appears that the greatest shear, $^{21}/_{11}T$, may exceed the endurance limit for shear while the greatest tension still falls short of the endurance limit in direct stress. Once a crack is formed, however, it is to be expected that its general direction of propagation will be at 45 deg to the direction of the "distant" applied shear, for it is across planes in this direction that the greatest tension acts. This is illustrated by the spiral fracture of twisted shafts.[9]

Two equal principal stresses, of the same sign, are not so greatly affected by a spherical cavity, the greatest tension being $^{15}/_{11}T$. For three equal principal stresses we have a check in the Lamé solution for fluid pressures on thick spherical shells, which gives the value of the hoop tension at the surface of the cavity, when the stress at a distance is hydrostatic tension T, as $^{3}/_{2}T$.

In general, by superposing the effects of three simple tensions or compressions of magnitudes corresponding to any three principal stresses at a distance, we can obtain the factors of stress concentration due to a cavity in any field of uniform or, relative to the cavity, slowly varying stress.

The corresponding results for a circular hole in a plate, or cylindrical bore in a solid subject to plane strain, are well known. For simple tension T in one direction there is a maximum tension $3T$ at the hole, and for pure shear T, a maximum tension $4T$ and a maximum shear $2T$. Hydrostatic tension T induces a hoop tension $2T$. The tensions due to a cylindrical hole in shear

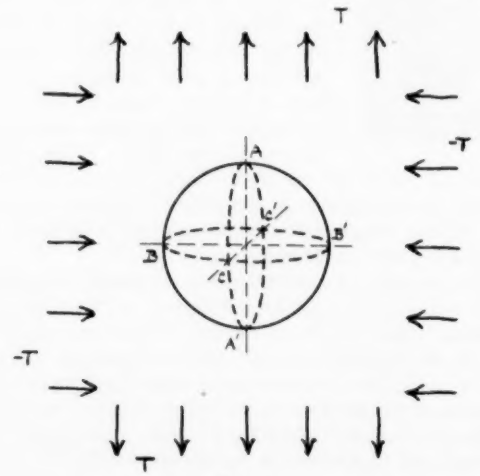


FIG. 2

are thus considerably greater than those due to a spherical cavity, but the shear stress is intensified to about the same degree by both.

We consider next a perfectly rigid inclusion, representing the worst possible case of an inclusion more rigid than the surrounding material. Such an inclusion, which adheres to the surrounding material, produces a different type of stress concentration. In simple tension, a perfectly rigid spherical inclusion intensifies the tension (adhesion stress) at A, A' (Fig. 3), in the same direction as the applied tension T, to $\left(\dfrac{2}{1 + \sigma} + \dfrac{1}{4 - 5\sigma}\right)T$, which is about $2T$ for values of Poisson's ratio ordinarily met with (0 to 0.5). The tension at the "equator," in the same direction is reduced to $\left(\dfrac{\sigma}{1 + \sigma} - \dfrac{5\sigma}{8 - 10\sigma}\right)T$. If $\sigma > {}^{1}/_{5}$, we have the curious result that this stress is compressive.

The rigid inclusion produces no appreciable intensification of shear. The radial (adhesion) stress in a region where the stress remote from the inclusion is a hydrostatic tension T, is a tension $3\dfrac{1 - \sigma}{1 + \sigma}\,T$, which will usually be nearly $2T$.

[9] R. V. Southwell and H. J. Gough, loc. cit.

A perfectly rigid cylindrical inclusion in simple tension T induces an adhesion tension at A, A' (Fig. 4) of magnitude $\frac{1}{2}\left(3 - 2\sigma + \frac{1}{3 - 4\sigma}\right) T$ (plane strain), about $1.5T$. The tension at B, B' is reduced to $\frac{1}{2}\left(1 + 2\sigma - \frac{3}{3 - 4\sigma}\right) T$. It becomes a compression if $\sigma > 1/4$. Two simple tensions T, at right angles, i.e., uniform all-around tension, produce an adhesion stress $2(1 - \sigma)T$. If the inclusion is spherical and consists of material having the same Poisson's ratio, but a Young's modulus one-quarter of that of the surrounding metal—taken as a plausible value for slag in steel—the stress concentrations have the same character as for a cavity, but are less in magnitude by about 20 per cent.

A cylindrical rod of steel set in cement, or in concrete with a fine aggregate, will induce a radial adhesion stress of about $1.5T$ when subject to simple transverse tension, and about $1.7T$ when subject to two such tensions at right angles. Poisson's ratio for the steel is taken as 0.3, for the concrete as 0.13, and the ratio of the Young's moduli as 15.

From the results for cavities and holes we conclude that, in

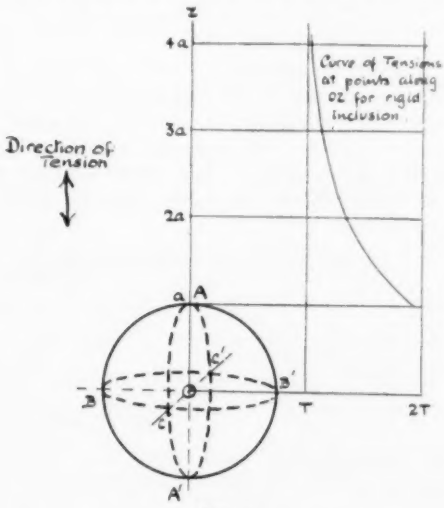


FIG. 3

simple tension, an inclusion appreciably less rigid than the surrounding material may be expected to intensify the tension at the ends of transverse diameters B, B'. Such an inclusion (e.g., slag in steel) would generally have the smaller coefficient of thermal expansion, and the residual stresses set up in cooling would involve tension at B, B' in the same direction. B. P. Haigh[10] has estimated that cooling through a temperature range of no more than 300 C would be sufficient to set up residual tensions of the same order as the yield stress, around an inclusion of slag in steel.

An inclusion which is appreciably more rigid than the surrounding material will behave like a perfectly rigid inclusion in intensifying the radial tension at the "poles" A, A'. Such an inclusion would probably have the greater thermal expansion, and the radial stresses, at A, A' and all other points on the boundary, which it would set up in cooling would be tensile. Hence the conclusion that residual cooling stresses will usually be directly augmented by the concentrations of applied tension.

AXIALLY SYMMETRICAL SOLUTIONS OF THE EQUATIONS OF ELASTICITY, IN POLAR COORDINATES

The Cartesian components of equilibrium displacement in the elastic solid free from body forces satisfy the three equations

$$\left(\frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial z}\right) \Delta + (1 - 2\sigma)\nabla^2(u, v, w) = 0 \quad \ldots\ldots [1]$$

where $\Delta \equiv \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}$ and ∇^2 is the Laplacian operator $\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$.

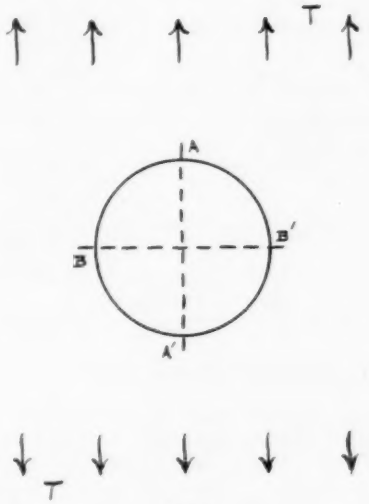


FIG. 4

A type of solution is given by

$$u, v, w = \left(\frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial z}\right) \phi \quad \ldots\ldots [2]$$

where ϕ is any harmonic function.[11] Such solutions correspond to distributions of the singularity "center of pressure" or "center of compression,"[12] for which the displacement is purely radial and inversely proportional to the square of the distance from the singularity. This displacement can thus be regarded as derived from a potential $1/r$, and the displacement due to a distribution of such singularities will therefore be the gradients of a Newtonian potential function, as in [2]. The displacements will be everywhere normal to the equipotentials and will be proportional to the density of electric charge on an isolated conductor coinciding with the equipotential. The dilatation vanishes.

A second type of solution of Equation [1] is

$$u, v, w = r^2\left(\frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial z}\right) \omega_n + \alpha_n(x, y, z)\omega_n \quad \ldots\ldots [3]$$

where ω_n is a homogeneous solid harmonic of degree n. This satisfies [1] in virtue of the properties

$$\nabla^2(x\omega_n) = 2\frac{\partial \omega_n}{\partial x}, \text{ etc.}, \quad \nabla^2(r^m\omega_n) = m(m + 2n + 1)r^{m-2}\omega_n$$

and the Euler relation for any homogeneous function

$$x\frac{\partial \omega_n}{\partial x} + y\frac{\partial \omega_n}{\partial y} + z\frac{\partial \omega_n}{\partial z} = n\omega_n$$

provided that α_n is given the value

[10] Loc. cit.

[11] A. E. H. Love, "Mathematical Theory of Elasticity," 4th Ed., art. 172.

[12] Ibid., p. 187.

$$\alpha_n = -2\,\frac{3n+1-2(2n+1)\sigma}{n+5-4\sigma} \qquad [4]$$

The dilatation is $\Delta = [2n + (3+n)\alpha_n]\omega_n$.

To obtain solutions symmetrical about an axis OZ, Fig. 5, we may use zonal harmonics for the functions ϕ and ω_n, and transform [2] and [3] to polar coordinates. The displacements will be confined to meridian planes, having a component u_r along the radius r, and a component u_θ at right angles to it, in the direction θ increasing. From [2], since the displacement is simply the gradient of ϕ,

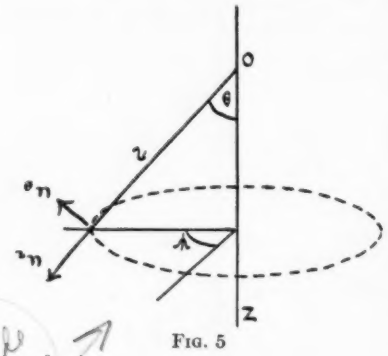


FIG. 5

$$u_r = \frac{\partial\phi}{\partial r}, \quad u_\theta = \frac{1}{r}\frac{\partial\phi}{\partial\theta} \qquad [5]$$

The general formulas for the strains are:

$$\left.\begin{aligned} e_{rr} &= \frac{\partial u_r}{\partial r}, \quad e_{\theta\theta} = \frac{1}{r}\frac{\partial u_\theta}{\partial\theta} + \frac{u_r}{r} \\ e_{\psi\psi} &= \Delta - e_{rr} - e_{\theta\theta}, \quad e_{r\theta} = \frac{1}{r}\frac{\partial u_r}{\partial\theta} + r\frac{\partial}{\partial r}\left(\frac{u_\theta}{r}\right) \end{aligned}\right\} \qquad [6]$$

which give, substituting [5],

$$e_{rr} = \frac{\partial^2\phi}{\partial r^2}, \quad e_{\theta\theta} = \frac{1}{r^2}\frac{\partial^2\phi}{\partial\theta^2} + \frac{1}{r}\frac{\partial\phi}{\partial r}, \quad e_{r\theta} = 2\,\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial\phi}{\partial\theta}\right)$$

The general stress-strain formulas are:

$$\widehat{rr},\ \widehat{\theta\theta},\ \widehat{\psi\psi} = 2\mu\left[\frac{\sigma}{1-2\sigma}\,\Delta + (e_{rr}, e_{\theta\theta}, e_{\psi\psi})\right], \quad \widehat{r\theta} = \mu e_{r\theta}. \qquad [7]$$

and since Δ vanishes for the ϕ-solutions, we have

$$\left.\begin{aligned} \widehat{rr} &= 2\mu\,\frac{\partial^2\phi}{\partial r^2}, \quad \widehat{\theta\theta} = 2\mu\left(\frac{1}{r^2}\frac{\partial^2\phi}{\partial\theta^2} + \frac{1}{r}\frac{\partial\phi}{\partial r}\right), \\ \widehat{\psi\psi} &= -(\widehat{rr} + \widehat{\theta\theta}), \quad \widehat{r\theta} = 2\mu\,\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial\phi}{\partial\theta}\right) \end{aligned}\right\} \qquad [8]$$

where μ is the modulus of rigidity.

For the second type of solution we observe from [3] that the displacement in any direction consists of the gradient of ω_n in that direction, multiplied by r^2, together with the component in the same direction of a vector $\alpha_n r\omega_n$ directed along the radius r. Hence

$$u_r = r^2\frac{\partial\omega_n}{\partial r} + \alpha_n r\omega_n, \quad u_\theta = r\,\frac{\partial\omega_n}{\partial\theta} \qquad [9]$$

The strains can be calculated by [6] and the stresses obtained from the stress-strain formulas [7]. Taking for ϕ the zonal spherical harmonics

$$\frac{1}{r} \quad \text{and} \quad \frac{1}{2r^3}(3\cos^2\theta - 1)$$

and for ω_n the second of these (ω_{-3}), each with an arbitrary multiplier, and superposing the resulting displacements and stresses, we find the following system for a region outside a spherical boundary:

$$\left.\begin{aligned} u_r &= -\frac{A}{r^2} - \frac{3B}{r^4} + \left[\frac{5-4\sigma}{1-2\sigma}\frac{C}{r^3} - 9\frac{B}{r^4}\right]\cos 2\theta \\ u_\theta &= -\left[\frac{2C}{r^3} + 6\frac{B}{r^4}\right]\sin 2\theta \\ \widehat{rr} &= 2\mu\left\{\frac{2A}{r^3} - \frac{2\sigma}{1-2\sigma}\frac{C}{r^3} + 12\frac{B}{r^5} + \left[-\frac{2(5-\sigma)}{1-2\sigma}\frac{C}{r^3} + 36\frac{B}{r^5}\right]\cos 2\theta\right\} \\ \widehat{\theta\theta} &= 2\mu\left\{-\frac{A}{r^3} - \frac{2\sigma}{1-2\sigma}\frac{C}{r^3} - 3\frac{B}{r^5} + \left[\frac{C}{r^3} - 21\frac{B}{r^5}\right]\cos 2\theta\right\} \\ \widehat{\psi\psi} &= 2\mu\left\{-\frac{A}{r^3} - \frac{2(1-\sigma)}{1-2\sigma}\frac{C}{r^3} - 9\frac{B}{r^5} + \left[3\frac{C}{r^3} - 15\frac{B}{r^5}\right]\cos 2\theta\right\} \\ \widehat{r\theta} &= 2\mu\left\{-\frac{2(1+\sigma)}{1-2\sigma}\frac{C}{r^3} + 24\frac{B}{r^5}\right\}\sin 2\theta \end{aligned}\right\} \qquad [10]$$

where A, B, and C are arbitrary.

The analogous system for the region inside a spherical boundary is obtained by taking for ϕ and ω_n ($= \omega_2$) the zonal spherical harmonic $\frac{1}{2}r^2(3\cos^2\theta - 1)$ with an arbitrary multiplier in each case, and superposing the "hydrostatic solution" for stress uniform in all directions (which corresponds to $\omega_n = \omega_0 =$ constant). We find

$$\left.\begin{aligned} u_r &= Hr + Fr + 2\sigma Gr^3 + [3Fr + 6\sigma Gr^3]\cos 2\theta \\ u_\theta &= -[3Fr + (7-4\sigma)Gr^3]\sin 2\theta \\ \widehat{rr} &= 2\mu\left\{\frac{1+\sigma}{1-2\sigma}H + F - \sigma Gr^2 + [3F - 3\sigma Gr^2]\cos 2\theta\right\} \\ \widehat{\theta\theta} &= 2\mu\left\{\frac{1+\sigma}{1-2\sigma}H + F - 5\sigma Gr^2 - [3F + 7(2-\sigma)Gr^2]\cos 2\theta\right\} \\ \widehat{\psi\psi} &= 2\mu\left\{\frac{1+\sigma}{1-2\sigma}H - 2F - (15-7\sigma)Gr^2 - (7+11\sigma)Gr^2\cos 2\theta\right\} \\ \widehat{r\theta} &= -2\mu\,\{3F + (7+2\sigma)Gr^2\}\sin 2\theta \end{aligned}\right\} \qquad [11]$$

where F, G, and H are arbitrary.

Uniform tension T (in the direction $\theta = 0$) in a *complete* elastic solid produces on a spherical surface of radius a the displacements and stresses

$$\left.\begin{aligned} u_r &= \frac{Ta}{2E}\left[(1-\sigma) + (1+\sigma)\cos 2\theta\right] \\ u_\theta &= -\frac{Ta}{2E}(1+\sigma)\sin 2\theta \\ \widehat{rr} &= \frac{T}{2}(1+\cos 2\theta) \\ \widehat{r\theta} &= -\frac{T}{2}\sin 2\theta \end{aligned}\right\} \qquad [12]$$

We can find values of the constants A, B, and C in [10] which give stresses $\widehat{rr}$ and $\widehat{r\theta}$ at $r = a$ equal and opposite to those of [12].

The corresponding system [10] superposed on the uniform tension T then gives the stresses for a spherical cavity of radius a. We can find another set of constants A, B, and C which give displacements at $r = a$ equal and opposite to those of [12], and the corresponding stresses from [10], superposed on the uniform tension, give the stresses for a rigid spherical inclusion.

For the elastic inclusion we distinguish the elastic constants in [10] by the subscript 1 (medium) and those in [11] by the subscript 2 (inclusion). The displacements and stresses outside the inclusion ($r > a$) are taken as those of [10] together with those due to uniform tension, and the displacements and stresses within the inclusion ($r < a$) are taken as those of [11]. Equating displacements and tractions at the common boundary $r = a$, enough equations are obtained to determine the six quantities A, B, C, F, G, H. The values of A, B, and C—which determine the stresses in the medium—are

$$\frac{A}{a^3} = -\frac{T}{8\mu_1}\frac{\mu_1 - \mu_2}{(7-5\sigma_1)\mu_1 + (8-10\sigma_1)\mu_2} \times \frac{(1-2\sigma_2)(6-5\sigma_1)2\mu_1 + (3+19\sigma_2-20\sigma_1\sigma_2)\mu_2}{(1-2\sigma_2)2\mu_1 + (1+\sigma_2)\mu_2} + \frac{T}{4\mu_1}\frac{\left[(1-\sigma_1)\dfrac{1+\sigma_2}{1+\sigma_1} - \sigma_2\right]\mu_2 - (1-2\sigma_2)\mu_1}{(1-2\sigma_2)2\mu_1 + (1+\sigma_2)\mu_2}$$

$$\frac{B}{a^5} = \frac{T}{8\mu_1}\frac{\mu_1 - \mu_2}{(7-5\sigma_1)\mu_1 + (8-10\sigma_1)\mu_2}$$

$$\frac{C}{a^3} = \frac{T}{8\mu_1}\frac{5(1-2\sigma_1)(\mu_1-\mu_2)}{(7-5\sigma_1)\mu_1 + (8-10\sigma_1)\mu_2}$$

The stresses in the medium are obtained by superposing the stresses in [10] with these values of the constants on the uniform tension.

The spherical cavity and rigid inclusion occur as limiting cases. For the cavity ($\mu_2 = 0$),

$$\frac{A}{a^3} = -\frac{T}{8\mu_1}\frac{13-10\sigma_1}{7-5\sigma_1}, \quad \frac{B}{a^5} = \frac{T}{8\mu_1}\frac{1}{7-5\sigma_1}, \quad \frac{C}{a^3} = \frac{T}{8\mu_1}\frac{5(1-2\sigma_1)}{7-5\sigma_1}$$

For the rigid inclusion ($\mu_2 = \infty$),

$$\frac{A}{a^3} = \frac{T}{4\mu_1}\left[\frac{1-\sigma_1}{1+\sigma_1} + \frac{3}{16-20\sigma_1}\right], \quad \frac{B}{a^5} = -\frac{T}{8\mu_1}\frac{1}{8-10\sigma_1},$$

$$\frac{C}{a^3} = -\frac{T}{8\mu_1}\frac{5(1-2\sigma_1)}{8-10\sigma_1}$$

Plane Strain and Plane Stress in Polar Coordinates

An analogous theory can be developed for two-dimensional systems. Taking the case of plane strain ($w = 0$), we shall have from [1]

$$\left(\frac{\partial}{\partial x}, \frac{\partial}{\partial y}\right)\Delta + (1-2\sigma)\nabla^2(u, v) = 0 \quad \ldots\ldots [13]$$

where $\Delta \equiv \dfrac{\partial u}{\partial x} + \dfrac{\partial v}{\partial y}$ and $\nabla^2 \equiv \dfrac{\partial^2}{\partial x^2} + \dfrac{\partial^2}{\partial y^2}$

The solutions of the ϕ-type are

$$u\ v = \left(\frac{\partial}{\partial x}\ \frac{\partial}{\partial y}\right)\phi$$

where ϕ is a plane harmonic function. The dilatation vanishes. The solutions of the ω-type are

$$u, v = r^2\left(\frac{\partial}{\partial x}, \frac{\partial}{\partial y}\right)\omega_n + \alpha_n(x, y)\omega_n$$

where $r^2 = x^2 + y^2$ and ω_n is a plane harmonic which is homogeneous, of degree n. These satisfy [13] in virtue of the properties

$$\nabla^2(x\omega_n) = 2\frac{\partial\omega_n}{\partial x}, \nabla^2(r^m\omega_n) = m(m+2n)r^{m-2}\omega_n$$

and the Euler relation for a homogeneous function of two variables

$$x\frac{\partial\omega_n}{\partial x} + y\frac{\partial\omega_n}{\partial y} = n\omega_n$$

provided that α_n is given the value

$$\alpha_n = -2n\frac{3-4\sigma}{n+4-4\sigma}$$

The dilatation Δ is

$$[2n + (2+n)\alpha_n]\omega_n$$

The stresses can be found from the stress-strain relations appropriate to plane strain when the strains have been calculated by the usual formulas from the displacements.

In polar coordinates we have, for the ϕ-type,

$$u_r = \frac{\partial\phi}{\partial r}, \quad u_\theta = \frac{1}{r}\frac{\partial\phi}{\partial\theta}$$

and for the ω-type,

$$u_r = r^2\frac{\partial\omega_n}{\partial r} + \alpha_n r\omega_n, \quad u_\theta = r\frac{\partial\omega_n}{\partial\theta}$$

For displacements and stresses which can be represented as Fourier series in θ we may use as plane harmonics of degree n the functions

$$r^n\cos n\theta, \quad r^n\sin n\theta$$

when n is not zero. As harmonics of degree zero we have $\log r$ and θ.

Taking for ϕ, $\log r$ and $(1/r^2)\cos 2\theta$ and for ω_n, $\omega_2 = (1/r^2)\cos 2\theta$ with arbitrary multipliers, we obtain, for the region outside a circular boundary,

$$\left.\begin{aligned} u_r &= \frac{A}{r} + \left[-\frac{B}{r^3} + \frac{2C}{r}(1-\sigma)\right]\cos 2\theta \\ u_\theta &= -\left[\frac{B}{r^3} + \frac{C}{r}(1-2\sigma)\right]\sin 2\theta \\ \widehat{rr} &= 2\mu\left[-\frac{A}{r^2} + \left(\frac{3B}{r^4} - \frac{2C}{r^2}\right)\cos 2\theta\right] \\ \widehat{\theta\theta} &= 2\mu\left[\frac{A}{r^2} - \frac{3B}{r^4}\cos 2\theta\right] \\ \widehat{r\theta} &= 2\mu\left[\frac{3B}{r^4} - \frac{C}{r^2}\right]\sin 2\theta \end{aligned}\right\} \ldots [14]$$

where A, B, and C are arbitrary.

The analogous system for the region within a circular boundary is derived from $\log r$ (ω_0) and $r^2\cos 2\theta$, and is

$$
\left.\begin{aligned}
u_r &= F_r + (Gr + 2\sigma Hr^3)\cos 2\theta \\
u_\theta &= -[Gr + (3 - 2\sigma)Hr^3]\sin 2\theta \\
\widehat{rr} &= 2\mu\left[\frac{F}{1-2\sigma} + G\cos 2\theta\right] \\
\widehat{\theta\theta} &= 2\mu\left[\frac{F}{1-2\sigma} - (G + 6Hr^2)\cos 2\theta\right] \\
\widehat{r\theta} &= -2\mu(G + 3Hr^2)\sin 2\theta
\end{aligned}\right\} \quad [15]
$$

Uniform tension T (in the direction $\theta = 0$) gives on the cylindrical surface $r = a$ the displacements and stresses (in plane strain)

$$
\left.\begin{aligned}
u_r &= \frac{Ta}{2E}(1+\sigma)[1 - 2\sigma + \cos 2\theta] \\
u_\theta &= -\frac{Ta}{2E}(1+\sigma)\sin 2\theta \\
\widehat{rr} &= \frac{T}{2}(1 + \cos 2\theta) \\
\widehat{r\theta} &= -\frac{T}{2}\sin 2\theta
\end{aligned}\right\} \quad [16]
$$

We can find the constants A, B, and C so that either the tractions $(\widehat{rr}, \widehat{r\theta})$, or the displacements of [14] eliminate those of [16], and so obtain solutions for the cylindrical hole or the perfectly rigid inclusion. For the elastic inclusion we combine [14] and the simple tension giving the boundary values [16], distinguishing the elastic constants by the subscripts 1 (medium) and 2 (inclusion). Equating the resultant boundary tractions and displacements to those of [15], representing the inclusion, we obtain equations for the six constants A, B, C, F, G, H. The stresses in the medium are determined by A, B, and C, which take the values

$$
\frac{A}{a^2} = \frac{T}{4\mu_1}\frac{(1-2\sigma_2)\mu_1 - (1-2\sigma_1)\mu_2}{(1-2\sigma_2)\mu_1 + \mu_2}
$$

$$
\frac{B}{a^4} = \frac{T}{4\mu_1}\frac{\mu_1 - \mu_2}{\mu_1 + (3-4\sigma_1)\mu_2}
$$

$$
\frac{C}{a^2} = \frac{T}{2\mu_1}\frac{\mu_1 - \mu_2}{\mu_1 + (3-4\sigma_1)\mu_2}
$$

For the cylindrical hole ($\mu_2 = 0$),

$$
\frac{A}{a^2} = \frac{B}{a^4} = \frac{T}{4\mu_1}, \quad \frac{C}{a^2} = \frac{T}{2\mu_1}
$$

For the perfectly rigid inclusion ($\mu_2 = \infty$),

$$
\frac{A}{a^2} = -\frac{T}{4\mu_1}(1-2\sigma_1), \quad \frac{B}{a^4} = -\frac{T}{4\mu_1}\frac{1}{3-4\sigma_1}
$$

$$
\frac{C}{a^2} = -\frac{T}{2\mu_1}\frac{1}{3-4\sigma_1}
$$

Dimensional Analysis for Inclusions of Any Shape

Consider the process of obtaining from the general equations of the theory of elasticity a formula for any stress p near an inclusion of given geometrical form, when the stress at a distance is of a given type, say, simple tension T in one direction. The formula consists of a relation between the following quantities

$$
p, T, E_1, E_2, \sigma_1, \sigma_2, r, a, \theta, \psi, \alpha, \beta, \gamma, \ldots
$$

where E_1, σ_1, E_2, σ_2 are elastic constants for the medium and the inclusion, a is a representative length giving the absolute size of the inclusion, α, β, γ . . . are the ratios of the other necessary dimensions of the inclusion to a, and r, θ, ψ are polar coordinates specifying the point at which the stress p is to be found.

There are only two fundamental units involved, and they may be taken as those of stress and length.[13] The dimensionless arguments obtainable from the quantities concerned may be taken as

$$
\frac{p}{T}, \frac{E_1}{T}, \frac{E_1}{E_2}, \sigma_1, \sigma_2, \frac{r}{a}, \theta, \psi, \alpha, \beta, \gamma, \ldots
$$

The stress p must be proportional to T, by the principle of superposition. It follows that the formula for p takes the form

$$
p = T f\left(\frac{E_1}{E_2}, \sigma_1, \sigma_2, \theta, \psi, \frac{r}{a}, \alpha, \beta, \gamma \ldots\right)
$$

where $f()$ denotes an undetermined function.

Since r and a appear only in the ratio r/a, the stresses in two systems with inclusions of the same form but different sizes, other things being the same, are the same at similarly situated points (r/a, θ, ψ, the same). The greatest stress will occur at such points and will clearly be the same whatever the size of the inclusion.

[13] The basis of the procedure followed is discussed in "Dimensional Analysis," by P. W. Bridgman.

APM-55-8

The Elastic Behavior of Vulcanized Rubber

By H. HENCKY,[1] CAMBRIDGE, MASS.

Up to extensions of about 270 per cent the elastic behavior of vulcanized rubber can be represented analytically by a very simple function. The stress-strain diagrams for tension and compression are quite different; nevertheless this asymmetrical behavior is satisfactorily taken care of by employing only two elastic coefficients.



THE mental attitude of the engineer to an organic compound such as rubber is a very skeptical one if its elastic behavior has to be compared with that of metals; nevertheless experiments with rubber belts furnished by the Goodyear Tire and Rubber Company to the Massachusetts Institute of Technology have convinced the author of this paper that a comparison of the elastic properties of rubber with those of metals is justified and at the same time very instructive as regards the mechanics of finite deformations in general.

Naturally rubber is more easily influenced by temperature and by surrounding liquid and gaseous substances which eventually cause chemical changes. However, vulcanized rubber can now be manufactured of such excellent quality that the way is opened to uses hitherto not exploited. Of these may be mentioned the use of rubber tires for the wheels of steam locomotives, enabling the latter to climb steep grades, and the wide field of application on damping vibrations and changing unfavorable critical speeds, already exemplified in the automotive industry.

Such applications can only be industrially exploited if the elastic stresses in rubber obey a known, mathematically defined law. In this paper the author therefore develops the law for large deformations and shows its agreement with the results of experiments made with the two rubber compounds that were furnished.

General Law of the Ideal Elastic Body[2]

We call a material ideally elastic if, even after a considerable deformation, it assumes its original shape after removal of the constraints introduced and if the process of loading and unloading is not accompanied by transformations of mechanical energy into heat. Nearly all materials are ideally elastic if the deformation is infinitesimal, but in the case of most substances finite deformations are accompanied by losses of mechanical energy. It is not the aim of this paper to study these losses, but it may be mentioned that there is even in the realm of ideal elasticity an unexplored region indicated by the experimental fact that the law for infinitesimal elastic deformations does not hold good for finite deformations. It is only necessary to subject a rubber cylinder to tension and compression at comparatively small strains, when the lack of agreement becomes obvious.

At the foundation of all elastic theories lies the definition of strain, and before introducing a new law of elasticity we must explain how finite strain is to be measured.

Infinitesimal strain is measured by the ratio of the increase in length to the length itself. In defining a considerable strain λ of a fiber we assume this strain to be divided into small portions $d\lambda$. Each of these portions we define, as in the case of an infinitesimal strain, by the ratio:

$$d\lambda = \frac{dx}{x} \qquad [1]$$

where x is the length of a fiber in a certain state of strain. Integrating and taking the length of the fiber before stressing as a and after stressing as b, we get

$$\lambda = \log_e \frac{b}{a} \qquad [1a]$$

Our measure of finite strain is therefore the natural logarithm of the ratio of the two lengths.

On the basis of the general theory developed in the Appendix we obtain the law of the simple *uniaxial* tension and compression test of rubber in the following two forms:

Stress per Unit Area Referred to the Deformed State:

$$\sigma = 2G\lambda(e_\lambda + \frac{1}{2}e^{-\lambda/2}) \qquad [2a]$$

Stress per Unit Area Referred to the Undeformed State:

$$\sigma_0 = 2G\lambda\left(1 + \frac{1}{2}e^{-3\lambda/2}\right) \qquad [2b]$$

If P is the load, A_0 the area of cross-section of the unloaded specimen, and A the area after the load is applied, we have, for incompressible material, $A = A_0 e^{-\lambda}$. The stress σ_0 is the observed stress, because all testing machines measure directly the force applied and not the true physical stress σ.

The coefficient G is about 67 lb. per sq. in. for the material examined and is the same as the modulus of shear for infinitesimal deformations. For such deformations both Equations [2a] and [2b] yield

$$\sigma = \sigma_0 = 3G\lambda$$

On the basis of Equations [2a] and [2b] and taking $2G = 134$ lb. per sq. in., Table 1 has been calculated and the values plotted in Fig. 1.

TABLE 1

$+\lambda$	$e^\lambda + \frac{1}{2}e^{-\lambda/2}$	$\lambda(e^\lambda + \frac{1}{2}e^{-\lambda/2})$	σ	σ_0
0.0	1.500	0.0	0	8
0.2	1.674	0.3348	45	37
0.4	1.901	0.7604	102	60
0.6	2.192	1.3152	176	97
0.8	2.561	2.0488	275	124
1.0	3.021	3.021	405	149
1.2	3.594	4.32	580	174
1.4	4.303	6.03	810	200
1.6	5.178	8.29	1110	224

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Before the war he spent a few years with the States Railways in Alsace-Lothringen in Strasbourg and later in Russia. At the outbreak of the war he was taken prisoner in Charkow and interned in the Ural district. Afterward he became a lecturer in mechanics at the Universities of Darmstadt, Dresden (Germany), and Delft (Holland).

[2] See *Journal of Rheology*, vol. 2, no. 2, pp. 169–176.

Presented at the National Applied Mechanics Meeting, New Haven, Conn., June 22–25, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

An analogous table for the negative values of λ is given in Table 2.

TABLE 2

$-\lambda$	$e^{-\lambda} + \frac{1}{2} e^{\lambda/2}$	$\frac{1}{3}\lambda(e^{-\lambda} + \frac{1}{2} e^{\lambda/2})$	σ	σ_0
0.0	1.500	0.0	0	0
0.2	1.371	0.2742	37	45
0.4	1.281	0.5124	69	102
0.6	1.224	0.7344	98	179
0.8	1.195	0.9560	128	285
1.0	1.192	1.192	160	435
1.2	1.212	1.455	195	645
1.4	1.253	1.750	235	951
1.6	1.315	2.100	282	1397

Experiments

The rubber with which the experiments were made was delivered in form of closed belts of 2 in. × 0.32 in. cross-section and 38 in. circumference. Two different compounds (Nos. 4183 and 4799) were used and there were 12 belts in all. The values of the elastic constants are about the same for both compounds, but slight variations in the elastic constants occur even at different parts of one and the same specimen.

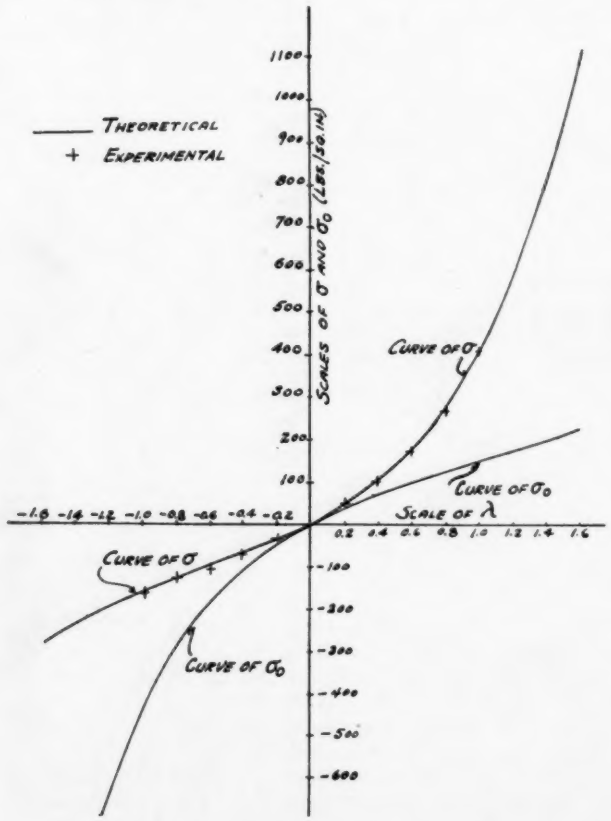


Fig. 1

The Bulk Modulus. To measure the bulk modulus, rubber cylinders 1 in. in diameter and 1.5 in. long were placed in a compression chamber and the distance between the pressure heads measured with Ames dials. The bulk modulus K was found to be 387,000 lb. per sq. in. or 27,200 kg. per sq. cm. (average of three experiments). This is somewhat larger than the modulus of water, which is 23,000 kg. per sq. cm., but agrees well with the fact that the specific gravity of rubber is about the same as that of water.

Considering the great difference between the shear and bulk moduli (their ratio is $G/K = 0.211 \times 10^{-3}$), it is justifiable to consider rubber as incompressible in the range for which validity is claimed for the elastic law [2]. The commonly used modulus of elasticity E is in that case equal to $3G = 200$ lb. per sq. in.

The Tensile Test. The tensile test under uniaxial stress is technically very easy to carry out. However, by an increase in length of 200 per cent or more, time effects and elastic strain hardening play an important role. The strain hardening is due to the microstructure of rubber and can only be taken into account by employing a statistical theory. This strain hardening has in some respects the character of internal friction; therefore the unloading curve coincides nearly with the author's theoretical curve.

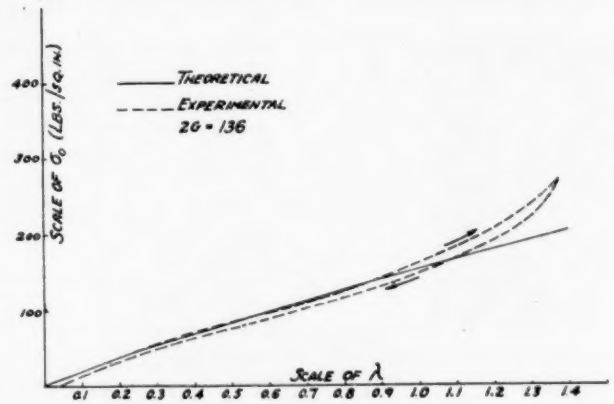


Fig. 2

The curves of Fig. 2 show the loading and unloading at a constant rate of applying strain and removing stress (time of loading and unloading, 4 hr. in all) with the stresses related to the original area of cross-section. Table 3 gives the values of the true stresses and strains for very slow loading and unloading. The observed values of λ are reduced to equal differences by interpolation, because it would not have been possible otherwise to take the average of the experiments with three specimens.

TABLE 3

$+\lambda$	Calculated stress σ (Equation [2a]), lb. per sq. in.	Measured stress σ, lb. per sq. in.
0.0	0.0	0.0
0.2	44.8	44.5
0.4	101.8	100.7
0.6	176.2	171.0
0.8	275.0	270.8
1.0	405.0	410.0

The Compression Test. The compression test is very difficult to perform and almost impossible to carry out for higher pressures if uniaxial stress has to be used. Not only is the state of uniform uniaxial stress unstable even for comparatively short rubber cylinders, but it is even difficult to realize such a state of stress. Despite careful greasing of the planes of contact with the pressure heads of the testing-machine, dry friction was detected. Evidently the grease is removed under too much pressure. Table 4 gives the calculated and measured stresses.

TABLE 4

$-\lambda$	Calculated stress σ (Equation [2a]), lb. per sq. in.	Measured stress σ, lb. per sq. in.
0.0	0.0	0.0
0.2	36.8	37.3
0.4	68.7	72.7
0.6	98.5	103.7
0.8	128.0	129.8
1.0	160.0	162.4

The change in the elastic behavior in going from positive to negative strains is very marked, but is satisfactorily represented by the theoretical formula. Fig. 2 shows the deviations from the theoretical curve at the larger strains.

Conclusions

As a consequence of the work described the author arrives at the conclusion that in the deformation range from 0 to 270 per cent only one elastic constant is needed for a full mathematical description, and that this constant is the modulus of shear defined for infinitely small deformations.

If models could be made of rubber at less expense, this material could be used for the integration of the equations of elasticity. The solutions could be easily corrected for the lack of compressibility and for the change in the common law of elasticity.

Appendix

Mathematical Derivations

Taking an element of a body and stressing it in three directions perpendicular to each other, x_1, x_2, x_3, we get the new lengths $\overline{dx_1}$, $\overline{dx_2}$, $\overline{dx_3}$ from the old ones dx_1, dx_2, dx_3. If we denote the principal strains by ϵ_1, ϵ_2, ϵ_3, the corresponding strains are, according to [1] and [1a],

$$\epsilon_i = \log_e \left(\frac{\overline{dx_i}}{dx_i} \right) \quad \ldots\ldots [3]$$

It is very convenient to take the average of these strains namely,

$$\epsilon = \frac{1}{3} (\epsilon_1 + \epsilon_2 + \epsilon_3) \quad \ldots\ldots [3a]$$

whence

$$3\epsilon = \log_e \left(\frac{\overline{dx_1}}{dx_1} \frac{\overline{dx_2}}{dx_2} \frac{\overline{dx_3}}{dx_3} \right) = \log_e \left(\frac{\overline{dV}}{dV} \right)$$

ϵ arises from that part of the strain which is equal in all directions. Consequently if we subtract ϵ from ϵ_1, ϵ_2, ϵ_3, and denote the result by φ_1, φ_2, φ_3,

$$\varphi = \epsilon_i - \epsilon \quad \ldots\ldots [3b]$$

will represent the pure deformation without the change in volume. We can resolve the stresses acting at any point in the same manner. $S = \frac{1}{3} (S_1 + S_2 + S_3)$ is that part of the stress which can exist in an ideal liquid. This hydrostatic part of the stress S is connected with the change of volume, the remainder of the stress, $S_i - S$, is connected with the change of form.

Now in the case of infinitely small strains it is not necessary to state that the stresses in the deformed states are to be employed when we formulate the elastic law. It does not make any difference for which state we give the stresses. With finite deformations, however, only the true physical stresses after equilibrium is attained can be used. For any homogeneous and isotropic material we can write down easily the simplest form the elastic energy can assume. We introduce into this two moduli, the modulus of shear G and the bulk modulus K, and choose the simplest possible analytical form, which becomes the expression of the classical theory of elasticity when we assume infinitely small displacements:

$$2W = 2G (\varphi_1^2 + \varphi_2^2 + \varphi_3^2) + 9K\epsilon^2 \quad \ldots\ldots [4]$$

W is the elastic energy per unit volume in the undeformed state. The energy must be related to the undeformed state because it is a physical entity connected with the mass of the body. Equation [4] is naturally an assumption and has to be checked by experiment.

But there is yet another doubt. Must we not relate the formula for the energy to the real bearer of the energy, the molecule? If the substance is built up from such units, should we not assume expression [4] as the energy of the molecule? Our experimental results indeed point in this direction. But before we study this question we shall formulate another expression for the work done which is independent of any law of elasticity.

If dV is the volume before and $\overline{dV}$ the volume after the deformation,

$$\delta W dV = S_1 \overline{dx_2}\, \overline{dx_3}\, \delta\, \overline{dx_1} + S_2 \overline{dx_1}\, \overline{dx_3}\, \delta\, \overline{dx_2} + S_3 \overline{dx_1}\, \overline{dx_2}\, \overline{dx_3}$$

or

$$\delta W = \frac{\overline{dV}}{dV} (S_1\delta\epsilon_1 + S_2\delta\epsilon_2 + S_3\delta\epsilon_3) = e^{3\epsilon} (S_1\delta\epsilon_1 + S_2\delta\epsilon_2 + S_3\delta\epsilon_3) \quad \ldots\ldots [5]$$

$e^{3\epsilon}$ is the relative volume, and will be denoted by v. Assuming now that Equation [4] is applied to isotropic and continuous matter, we can immediately derive the law of elasticity.

By differentiating [4],

$$\delta W = 2G [\varphi_1 (\delta\epsilon_1 - \delta\epsilon) + \varphi_2 (\delta\epsilon_2 - \delta\epsilon) + \varphi_3 (\delta\epsilon_3 - \delta\epsilon)] + 3K\epsilon (\delta\epsilon_1 + \delta\epsilon_2 + \delta\epsilon_3)$$

$$= \delta\epsilon_1(2G\varphi_1 + 3K\epsilon) + \delta\epsilon_2(2G\varphi_2 + 3K\epsilon) + \delta\epsilon_3(2G\varphi_3 + 3K\epsilon)$$

Comparing this expression with Equation [5],

$$S_i v = 2G\varphi_i + 3K\epsilon \quad \ldots\ldots [6a]$$

or in another form,

$$\left. \begin{aligned} S_i - S &= \frac{2G}{v} \varphi_i \\ S &= \frac{K}{v} \log_e v \end{aligned} \right\} \quad \ldots\ldots [6b]$$

The apparent increase of the modulus of shear for high compression measured by Bridgman is accounted for in Equation [6b], because v is in that case less than unity.

For rubber v differs but little from unity so long as S is in the range of 0–4000 lb. per sq. in.

Under these circumstances we can neglect the change in volume entirely, writing

$$S_i = 2G\varphi_i + S \quad \ldots\ldots [7]$$

for an ideal incompressible material.

But having considered an element dx_1, dx_2, dx_3 of continuous matter which led us to Equation [7], we now assume an element containing thousands of molecules. As it does not matter which form we assume them to be in, let us take them as small spheres. The true shape of the molecule of rubber is not known, but it probably is of a prismatic form. We must keep clearly in mind, therefore, that we are neglecting the true microstructure of rubber in assuming spherical molecules.

Here we no longer apply Equation [7] to the stresses, but to the forces with which the molecules act on one another, writing

$$P_i - P = 2G'\varphi_i \quad \ldots\ldots [8]$$

where $P = \frac{1}{3} (P_1 + P_2 + P_3)$.

If N is the number of molecules lying in the unit of area of a certain cross-section in the undeformed state, then after deformation there will be in the $\overline{dx_2}$, $\overline{dx_3}$ plane the number Ne^{φ_1} molecules, since

$$\frac{\overline{dx_2}}{dx_2} \cdot \frac{\overline{dx_3}}{dx_3} = \frac{\overline{dx_1}}{dx_1} = e^{-\varphi_1}$$

for $\epsilon = 0$.

Multiplying Equation [8] by Ne^{φ_1} gives the true stresses

$$Ne^{\varphi_1}(P_1 - P) = 2G'N\varphi_1 e^{\varphi_1}$$

$$Ne^{\varphi_2}(P_2 - P) = 2G'N\varphi_2 e^{\varphi_2}$$

$$Ne^{\varphi_3}(P_3 - P) = 2G'N\varphi_3 e^{\varphi_3}$$

These stresses now comprise an amount of hydrostatic stress which has to be subtracted. As the hydrostatic stresses in incompressible material are not elastically but statically determined, we can so determine them and, putting $G'N = G$, obtain

$$S_i - S = 2G\left[\varphi_i e^{\varphi_i} - \left(\frac{\varphi_1 e^{\varphi_1} + \varphi_2 e^{\varphi_2} + \varphi_3 e^{\varphi_3}}{3}\right)\right] \ldots [9]$$

Putting $\varphi_1 = \lambda$, $\varphi_2 = \varphi_3 = -\frac{1}{2}\lambda$, $S_1 = \sigma$, and $S = \frac{1}{3}\sigma$ in [9] we get Equation [2a], and then, after dividing by e^{λ}, Equation [2b].

It is easy to generalize the law [9] so that it can be applied to other substances.

We wish, however, to conserve under all circumstances the condition that the stresses can be derived from an elastic potential.

If we put

$$\left.\begin{aligned} S_i - S &= \frac{2G}{v}\left\{f(\varphi_i) - \frac{1}{3}[f(\varphi_1) + f(\varphi_2) + f(\varphi_3)]\right\} \\ S &= \frac{3K}{v}\psi(\epsilon) \end{aligned}\right\} \ldots [10]$$

the work done by the stresses, with $\epsilon_1 = \epsilon + \varphi_1$, $\epsilon_2 = \epsilon + \varphi_2$, $\epsilon_3 = \epsilon + \varphi_3$, will be, according to [5],

$$\delta W = [(S_1 - S)\delta\varphi_1 + (S_2 - S)\delta\varphi_2 + (S_3 - S)\delta\varphi_3 + 3S\delta\epsilon]v$$

Introducing the value of $S_i - S$ from [10],

$$W = 2G\left[\int_0^{\varphi_1} f(\varphi_1)\delta\varphi_1 + \int_0^{\varphi_2} f(\varphi_2)\delta\varphi_2 + \int_0^{\varphi_3} f(\varphi_3)\delta\varphi_3\right] + 9K\int_0^{\epsilon} \psi(\epsilon)\delta\epsilon \ldots\ldots [11]$$

The functions $f(\varphi_i)$ and $\psi(\epsilon)$ must therefore be obtained experimentally.

The use of Equation [11] is more satisfactory than that of expressions which can be considered only as interpolation formulas of the experiments, because, according to [11], the work done is independent of the manner in which the material is loaded.

In the study of the behavior of rubber under deformations greater than 270 per cent we cannot neglect the influences of the time effects comprising relaxation and creeping (see Fig. 2). The shape of the rubber molecules also plays an important part in the behavior under very large deformation.

The study of these deformations is not important for technical purposes because in practice such large strains are never used, but the subject is very interesting from the standpoint of physical chemistry, because it enables definite conclusions to be drawn concerning the microstructure of rubber.

Acknowledgments

Grateful acknowledgment is due to Prof. H. W. Hayward of the Massachusetts Institute of Technology for valuable aid and support, to the Goodyear Rubber and Tire Co. for furnishing the specimens, and to I. Silverman for his assistance in carrying out the tests.

Discussion

William Hovgaard.[3] The curves of stress and strain given in Fig. 1 show that the proportionality between these quantities existing in ordinary elastic materials, and expressed by the constancy of the modulus of elasticity E, does not exist in rubber. Hence all attempts at using the ordinary law of elasticity to rubber have failed, it being found that E was not constant. Now the author has shown that rubber nevertheless follows a very definite law, which he derives in the Appendix and which is expressed by the simple formulas [2a] and [2b]. This law differs radically from that given by Hooke, in that a new expression for the strain λ is introduced, being a logarithmic function of the length before and after strain at every point. In other words, instead of expressing the elongation or compression as a fraction of unit length, say ϵ_1 in uniaxial stress, he takes the logarithm of $1 + \epsilon_1$ as a measure of the strain. In the limit, when ϵ_1 is very small, this makes $\lambda = \epsilon_1$. He shows, moreover, that the volume strain in rubber is so small, relative to the deformation, that it can be neglected; Poisson's ratio becomes equal to 2, and the relation between stress and strain can be described by one elastic constant, G, within a deformation range of up to 270 per cent.

The curves given in Fig. 1 are, it is supposed, the result of a series of tests. It would be of interest to know more about the number of tests, the kind of material used, and its source. Was the material of one specially selected quality or did it represent various grades?

The statement that vulcanized rubber is suitable for tires for the wheels of steam locomotives, enabling them to climb steep grades, is very interesting and important. The writer would ask whether tests of this kind have been made and whether the qualities of rubber revealed in this paper explain the suitability of rubber for this purpose.

In the Appendix the mathematical development of the formulas is given. Dr. Hencky first writes down the elastic potential energy W in terms of the principal and average strains, dividing it into two parts, one representing pure deformation and the other pure volume strain. Differentiating this expression, he obtains a value for δW, which he equates to that obtained by applying the principle of virtual velocities, involving the stresses S_i. He thus obtains Equation [7], which gives an expression for the stress S_i acting on a continuous particle of matter or on a single molecule, which he conceives as a sphere capable of great deformation, but of constant volume. Equation [7] gives a linear relation between the stress and strain, which obviously does not conform to the behavior of rubber, but the author now passes to the consideration of the forces with which the molecules act on one another and introduces stress forces P_i acting on a great number of molecules. He thus, through a development which is somewhat difficult to follow, arrives at Equation [9], and applying this to the case of uniaxial stress, which he denotes by σ, he finally arrives at formulas [2a] and [2b]. It is suggested that the author elaborate and explain the development connecting Equation [7] with Equations [8] and [9] more fully; also, it is not seen clearly why $G'N = G$. This paper is of great practical importance. It brings to light remarkable and unexpected qualities of vulcanized rubber and furnishes a simple equation which represents its behavior with great accuracy.

A. V. Karpov[4] and R. L. Templin.[5] The interesting theo-

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retical and experimental work by the author has to be limited to the particular rubber compound fabricated under a particular manufacturing procedure that was investigated. It is probable that by way of further experiments it can be extended over a number of rubber compounds, but it is fairly certain that it cannot be applied to vulcanized rubber in general, as the title of the paper suggests.

The numerous products that are at present used commercially under the general classification of vulcanized rubber have in common the fact that among other ingredients they contain a substantial admixture of natural rubber, but differ very considerably in their elastic behavior. The writers were confronted recently with the rather unusual problem of finding a material that would meet the following set of specifications:

1 Specific gravity of 2.4
2 Uniform elastic properties without any pronounced directional qualities
3 Absence of skin or internal stresses
4 Low modulus of elasticity
5 Reasonably uniform values of the modulus of elasticity at various unit stresses
6 Low permanent set
7 Low plastic properties
8 Properties that will permit the material to be easily furnished in the required irregular shapes
9 Poisson's ratio close to 0.20
10 Uniform coefficient of thermal expansion.

This material was to be used in building a model of an arch dam, to be tested to determine the deflections and stresses.[6] The range of stresses of the material in the model was from 6 lb per sq in. tension to 20 lb per sq in. compression.

A search made among materials that are available commercially or any known materials that were investigated brought out the fact that no material will satisfy the specifications entirely. Very many materials can be found that will satisfy a single requirement of the specifications, but the number decreases with astounding rapidity with each additional requirement. The known rubber compounds gave good promises, but all of them had much lower specific gravities and rather uncertain elastic properties. The whole problem resolved itself into a research proposition of developing a proper rubber compound.

Directional Properties

It is a well-established fact that the shape of the particles of the fillers is to a large extent responsible for the directional properties, and in order to exclude rubber compounds with pronounced directional properties, only known isotropic fillers or fillers unknown so far as their isotropic properties are concerned were used. Besides that, a certain amount of responsibility is due to the manufacturing process that may introduce directional properties or that makes them more pronounced. That the milling process is the one that introduces the directional properties or exaggerates them, due to a definite orientation of the particles of the fillers, is a fairly well-accepted theory. It is clear that no final judgment can be passed on the elastic properties of a rubber compound unless its directional properties are investigated. This was done by the preparing and testing of three specimen cylinders.

The first specimen cylinder was made by cementing together a number of rubber-compound sheets and cutting out a cylinder with its axis at right angles to the surface of the sheets. Such cylinders were designated as "across the slabs" and are indicated on the tables and figures as X.

The second specimen cylinder was made by cutting it out of a number of sheets cemented together, with the axis of the cylinder parallel to the surface of the sheets, and also to the direction in which the uncured sheets were milled. These cylinders were designated "with slabs 1" and are indicated as W1.

The third specimen cylinder was made by cutting it out of a number of sheets cemented together with the axis of the cylinder parallel to the surface of the sheets and at right angles to the direction in which the uncured sheets were milled. These cylinders were designated "with slabs 2" and are indicated as W2.[7]

Discussion of the Tests

The tests made may be divided into the following groups:

1 Tests of different compounds
2 Tests to determine directional properties
3 Repeated tests.

Data pertaining to these three groups of tests are given in Tables 5, 6, and 7.

The stress-strain curves for some of these various compounds are shown in Figs. 3, 4, and 5. These curves are drawn in the usual way, but the closing line of each curve connects two observed points—the largest deformation and the permanent deformation observed after the removal of the loads.

The solid lines are the strains in the axial directions and the dotted in the lateral directions. The ratio between these two strains at any of the loadings gives the corresponding values of Poisson's ratio.

1 *Tests of Different Compounds.* Each of the compounds listed in Table 5 was tested in one direction. Specimens of the same composition which were fabricated at approximately the same time are numbered with the same number, followed by a letter, while specimens of the same composition which were fabricated and tested at a later date are numbered differently.

Specimen 3 showed a much lower Poisson's ratio than either 1 or 2, which was taken as an indication that by the use of a proper filler, Poisson's ratio might be decreased and a material obtained which would more nearly represent concrete in its elastic behavior. A study of the data given in Table 5 shows that while there may be a tendency for some fillers to decrease Poisson's ratio, there seems to be no consistent tendency unless it may be for zinc oxide and microcell.

The investigated compounds cover a wide range, starting with a modulus of elasticity as low as 490 lb per sq in. for the compound 1 and up to 10,400 lb per sq in. for the compound 22. The permanent sets range from 1.4 per cent for 1 and up to 27.5 per cent for 30. The Poisson's ratio varied from 0.26 for 22 up to 0.50 for a number of compounds. The samples with a favorable low Poisson's ratio did show high permanent set.

Graphical data for the more important compounds are given in Fig. 3. The compound 1 shows a practically straight line for the strain-stress curve in both axial and lateral directions and a negligible permanent set. The rest of the compounds show more or less pronounced curvature of the strain-stress curves and different values of the permanent set.

2 *Tests to Determine Directional Properties.* Table 6 shows that only a few compounds are satisfactorily uniform so far as the values of moduli of elasticity in the three directions are concerned. If the uniformity of the values of the Poisson's

[6] See discussion by A. V. Karpov of the paper of Benjamin F. Groat, M. Am. Soc. C. E., entitled "Theory of Similarity and Models," Trans. A.S.C.E., vol. 96, 1932, p. 308.

[7] For a description of the testing procedure see R. L. Templin and R. G. Sturm, "Methods for Determining the Physical Properties of Certain Rubber Compounds at Low Stresses," Proc. A.S.T.M., vol. 31, part 2, p. 882 (1931).

TABLE 5 TESTS OF DIFFERENT COMPOUNDS

Serial No. R-924-V	1	1	2	3	3	3c	3d	3e	5	12	13	14	15	18	20	21	22	22a	22b	23	25	27	28	28a	30
Time of curing, min	75	75	75	50	50	75	75	150	150	100	60	60	100	60	75	75	75	150	75	75	75	75	150	75	75
Temperature of curing, deg F	285	285	285	285	285	285	285	285	285	260	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285
Shape	Slab	Slab	Cyl.	Slab	Slab	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.	Cyl.
Filler by volume per 100 volumes of rubber:																									
Activated alumina	..	..	..	..	..	..	..	..	..	56.0	34.6	..	61.2	..	..	..	..	..	..	..	..	32.8	..	..	..
Barytes	..	..	71.5	..	..	..	..	..	89.0	..	59.0	67.2	..	..	..	..	..	..	..	..	..	..	..	..	..
Calcium aluminate	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	30.0	..	..	..	..
Carbon black, P. 33	..	..	..	..	..	..	..	..	53.0	..	..	..	..	..	32.0	..	..	..	..	..	..	..	..	..	..
Ground silica	..	..	..	..	..	..	..	..	..	..	..	36.0	..	..	..	..	..	..	..	..	..	..	..	..	..
Litharge	22.0	22.0	..	..	..	..	..	..	..	..	..	..	13.1	20.5	..	..	..	..	..	..	..	..	..	..	..
Microcell	..	..	..	..	..	..	..	..	..	..	..	..	..	40.0	..	..	22.0	22.0	22.0	..	..	..	..	..	41.0
Thermatonic	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	32.0	..	..	..	..	..	..	..	..	..
Zinc oxide	..	..	4.6	48.0	4.80	48.0	48.0	48.0	4.6	38.7	4.6	4.6	4.6	4.6	56.0	56.0	51.0	51.0	51.0	..	46.5	..	48.0	48.0	48.0
Zinc oxide (American)	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	48.0	..	48.0	..	..	..
Internal or surface stresses noted	None	None	None	None	None	None	None	None	Slight	Some	None	None	Very much	Slight	Very slight	Slight	Slight	Very slight	Slight	None	None	None	None	None	None
Permanent set (% max deformation)	1.4	1.4	5.4	5.5	5.5	6.4	8.4	7.5	15.5	23.0	17.0	7.5	24.6	9.0	11.0	6.9	23.4	9.5	13.1	4.2	11.6	11.2	8.3	11.2	27.5
Modulus of elasticity E, lb/sq in.	530	490	1050	1020	884	700	1570	1050	6710	2370	1130	1500	1610	1590	7610	2810	7410	4330	10400	1240	5040	800	1370	756	7700
Poisson's ratio, μ	0.47	0.46	0.47	0.33	0.29	0.49	0.51	0.47	0.34	0.29	0.48	0.51	0.40	0.44	0.34	0.43	0.26	0.48	0.43	0.51	0.48	0.49	0.50	0.50	0.42
Remarks	..	Tens. test	..	..	Tens. test	..	..	..	..	..	..	..	..	..	..	..	Time load	..	..	..	Time load	..	..	..	..

NOTE: Specific gravity of all compounds = 2.40 ± 0.08.

TABLE 6 TESTS TO DETERMINE DIRECTIONAL PROPERTIES

Serial No. R-924-V		1a	3a	3b	30b	31	32	33	34	35	36	41	42
Time of curing, min		75	75	75	75	75	75	75	75	75	75	75	75
Temperature of curing, deg F		285	285	285	285	285	285	285	285	285	285	285	285
Shape		Slab	Slab	Slab	Slab	Disk	Disk	Disk	Disk	Disk	Disk	Disk	Slab
Filler by volume per 100 volumes of rubber	Litharge	22.0	....	....	....	....	....	....	....	....	....	....	....
	Microcell	....	....	....	41.0	41.0	41.0	41.0	41.0	41.0	41.0	36.0	36.0
	Zinc oxide	....	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0
Internal or surface stresses noted		None	None	None	Slight	Some	Slight	Slight	Slight	Slight	Slight	Much	Slight
Permanent set (% of max deformation)		0.5	7.0	8.3	22.0	23.5	21.1	14.7	23.1	22.2	16.2	14.8	18.8
Modulus of elasticity E in lb per sq in. and Poisson's ratio, μ	W1 E	580	1480	1550	16200	7170	4570	3510	5100	2900	2390	7130	6140
	W1 μ	0.50	0.49	0.48	0.49	0.33	0.42	0.42	0.40	0.51	0.45	0.39	0.46
	W2 E	550	2110	1110	29000	7550	6220	3510	10500	6140	2630	8500	9210
	W2 μ	0.46	0.52	0.51	0.27	0.38	0.33	0.42	0.11	0.32	0.47	0.49	0.46
	X E	475	1660	950	10200	4100	3170	970	4860	3930	2360	[a]	6110
	X μ	0.48	0.19	0.47	0.17	0.14	0.54	0.40	0.20	0.24	0.37	[a]	0.44
Average modulus of elasticity		535	1750	1200	18470	6270	4650	2660	6820	4320	2460	7820	7180
Deviation of modulus of elasticity from average (%)	W1	+8.4	−15.4	+29.2	−12.3	+14.4	−2.6	+32.0	−25.2	−32.9	−2.8	−8.8	−14.2
	W2	+2.8	+20.6	−7.5	+57.0	+20.4	+33.8	+32.0	+54.0	+42.2	+6.9	+8.8	+28.8
	X	−11.2	−5.2	−20.8	−44.5	−34.6	−31.8	−63.5	−28.8	−9.1	−4.1	[a]	−14.5

NOTE: Specific gravity of all compounds = 2.40 ± 0.08. [a] Sample badly warped; no test made.

TABLE 7 REPEATED TESTS

Serial No. R-924-V	1	1	1a	1a	21	21	22	22	3b	3b	3b	30b	30b	30b	30b	34	34	36	36	36	36	15	15	12	12
Time of curing, min	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	100	100	100	100
Temperature of curing, deg F	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285	285	260	260
Shape	Slab	Slab	Slab	Slab	Cyl.	Cyl.	Cyl.	Cyl.	Slab	Slab	Slab	Slab	Slab	Slab	Slab	Disk	Disk	Disk	Disk	Disk	Disk	Cyl.	Cyl.	Cyl.	Cyl.
Filler by volume per 100 volumes of rubber: Activated alumina	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	61.2	61.2	56.0	56.0
Litharge	22.0	22.0	22.0	22.0	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	13.1	13.1	..	..
Microcell	..	..	..	..	..	..	22.0	22.0	..	..	..	41.0	41.0	41.0	41.0	41.0	41.0	41.0	41.0	41.0	41.0	..	..	..	..
Thermatonic	..	..	..	..	32.0	32.0	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
Zinc oxide	..	..	..	..	56.0	56.0	51.0	51.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	4.6	4.6	38.7	38.7
Internal or surface stresses noted	None	None	None	None	Slight	Slight	Slight	Slight	None	None	None	Slight	None	None	None	Slight	Slight	Slight	Slight	Slight	Slight	Much	Much	Some	Some
Permanent set (% max deformation)	1.4	1.8	0.5	1.2	6.9	12.5	23.4	9.0	8.3	8.0	6.2	22.0	16.7	20.9	19.6	23.1	19.0	16.2	14.5	14.3	13.5	24.6	8.1	23.0	22.5
Modulus of elasticity E in lb per sq in. and Poisson's ratio μ W1 E	..	..	580	560	..	..	..	..	1550	1570	1680	16200	22650	17300	15090	5100	5830	2390	..	..	..	..	..	..	..
W1 μ	..	..	0.50	0.49	..	..	..	..	0.48	0.48	0.46	0.49	0.42	0.42	0.42	0.40	0.33	0.45	..	..	..	..	..	..	..
W2 E	..	..	550	560	..	..	..	..	1110	1180	1210	29000	14400	15600	15500	10500	10900	2630	2660	2740	..	..	..	..	..
W2 μ	..	..	0.46	0.50	..	..	..	..	0.51	0.50	0.49	0.27	0.39	0.36	0.41	0.11	0.46	0.47	0.47	0.45	..	..	..	..	..
X E	530	505	475	480	2810	2550	7410	11220	950	1070	1130	10200	..	14550	20700	4860	4990	2360	2280	2610	2320	1610	1570	2370	2130
X μ	0.47	0.51	0.48	0.51	0.43	0.51	0.26	0.54	0.47	0.53	0.53	0.17	..	0.26	0.20	0.20	0.32	0.37	0.35	0.46	0.54	0.40	..	0.29	0.36
Average modulus of elasticity	..	..	535	530	..	..	..	..	1200	1270	1340	18470	18530	15820	17100	6820	7240	2460	2470	2680	..	..	..	..	..
Deviation of modulus of elasticity from average (%) W1	..	..	+8.4	+5.1	..	..	..	..	+29.2	+23.6	+25.4	−12.3	+22.2	+9.4	−11.7	−25.2	−19.5	−2.8	..	..	..	..	..	..	..
W2	..	..	+2.8	+5.1	..	..	..	..	−7.5	−7.1	−9.7	+57.0	−22.2	−1.4	−9.4	+54.0	+50.6	+6.9	+7.7	+2.4	..	..	..	..	..
X	..	..	−11.2	−10.0	..	..	..	..	−20.8	−15.8	−15.7	−44.5	..	−8.0	+21.0	−28.8	−31.1	−4.1	−7.7	−2.4	..	..	..	..	..
Remarks	[a]	[b]	[a]	[b]	[a]	Slow retest	[a]	Retest	[a]	[b]	[c]	[a]	[b]	[c]	[d]	[a]	[b]	[a]	Retest	Third test	Fourth test	[a]	Retest	[a]	Re-test

NOTE: Specific gravity of all compounds = 2.40 ± 0.08.
[a] Original test. See Tables 5 and 6. [b] After first application of hydrostatic pressure. [c] After second application of hydrostatic pressure. [d] After third application of hydrostatic pressure.

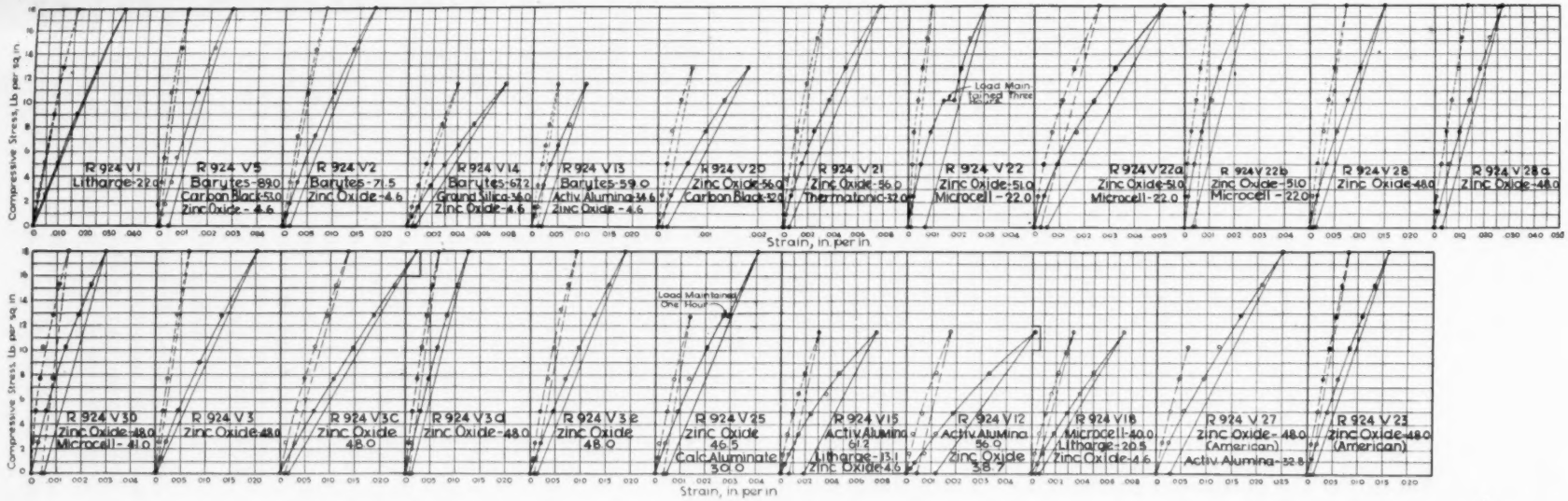

Compressive Stress, Lb per sq in
Strain, in. per in.
R 924 V1 Litharge-22.0
R 924 V5 Barytes-89.0 Carbon Black-5.0 Zinc Oxide - 4.6
R 924 V2 Barytes - 71.5 Zinc Oxide-4.6
R 924 V14 Barytes-67.2 Ground Silica-14.0 Zinc Oxide-4.6
R 924 V13 Barytes-59.0 Activ. Alumina-14.6 ZINC Oxide - 4.6
R 924 V20 Zinc Oxide-56.0 Carbon Black-5.0
R 924 V21 Zinc Oxide-56.0 Thermatomic-32.0
R 924 V22 Zinc Oxide-51.0 Microcell - 22.0
Load Maintained Three Hours
R 924 V22a Zinc Oxide-51.0 Microcell - 22.0
R 924 V22b Zinc Oxide-51.0 Microcell - 22.0
R 924 V28 Zinc Oxide-48.0
R 924 V28a Zinc Oxide-48.0
R 924 V3b Zinc Oxide-48.0 Microcell - 41.0
R 924 V3 Zinc Oxide-48.0
R 924 V3c Zinc Oxide 48.0
R 924 V3d Zinc Oxide-48.0
R 924 V3e Zinc Oxide 48.0
Load Maintained One Hour
R 924 V25 Zinc Oxide 46.5 Calc.Aluminate 30.0
R 924 V15 Activ.Alumina 61.2 Litharge-13.1 Zinc Oxide-4.6
R 924 V12 Activ.Alumina 56.0 Zinc Oxide 38.7
R 924 V18 Microcell-40.0 Litharge-20.5 Zinc Oxide-4.6
R 924 V27 Zinc Oxide - 48.0 (American) Activ. Alumina-32.8
R 924 V23 Zinc Oxide-48.0 (American)

FIG. 3

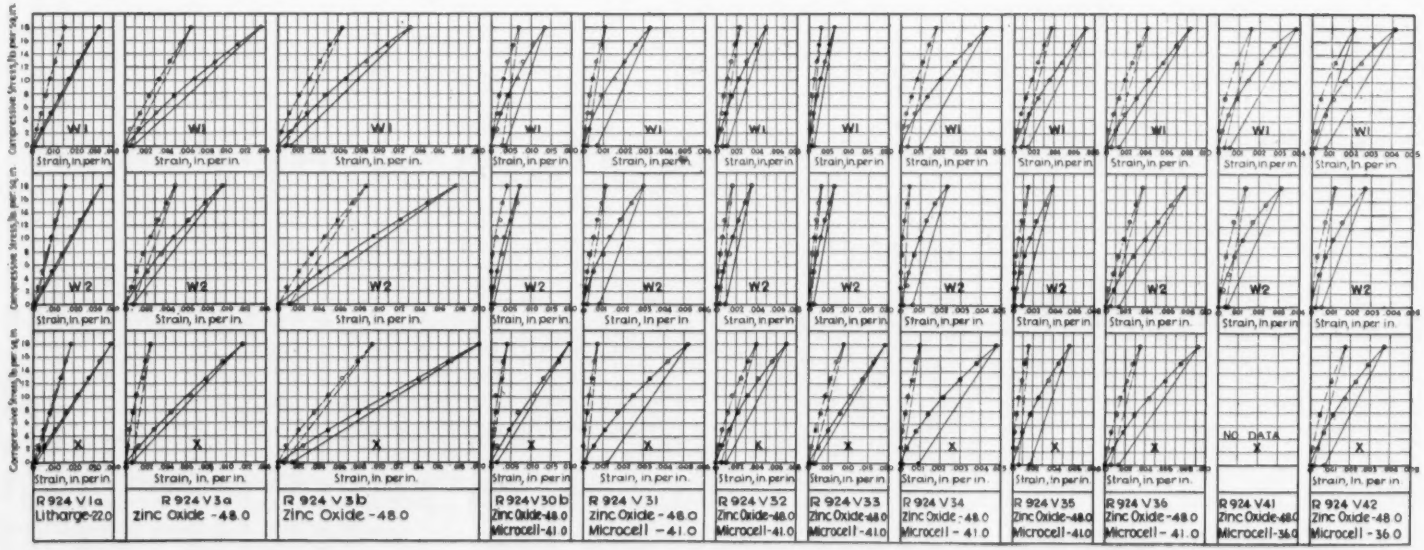

Compressive Stress, lb per sq in
Strain, in per in
W1
W2
X
R 924 V1a Litharge-22.0
R 924 V3a Zinc Oxide - 48.0
R 924 V3b Zinc Oxide - 48.0
R 924V30b Zinc Oxide-48.0 Microcell-41.0
R 924 V31 Zinc Oxide - 48.0 Microcell - 41.0
R 924 V32 Zinc Oxide-48.0 Microcell-41.0
R 924 V33 Zinc Oxide-48.0 Microcell - 41.0
R 924 V34 Zinc Oxide - 48.0 Microcell - 41.0
R 924 V35 Zinc Oxide-48.0 Microcell-41.0
R 924 V36 Zinc Oxide - 48.0 Microcell - 41.0
NO DATA
R 924 V41 Zinc Oxide-48.0 Microcell-36.0
R 924 V42 Zinc Oxide-48.0 Microcell - 36.0

FIG. 4

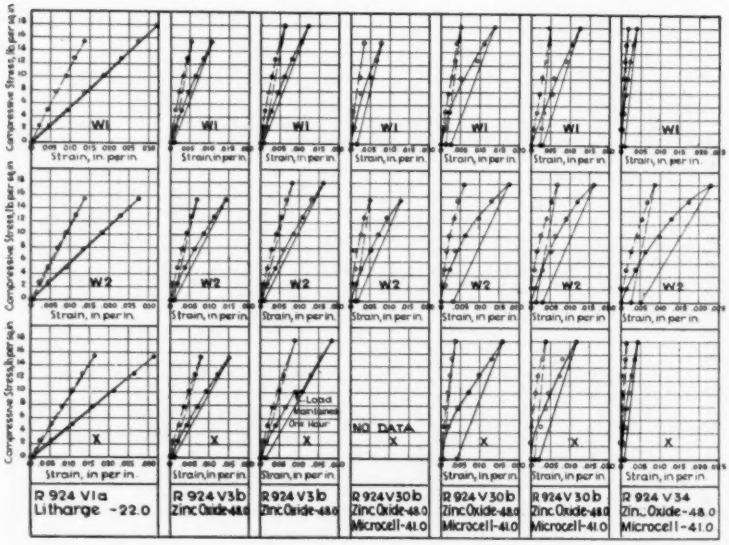


FIG. 5

ratio is also considered, then only the compound 1a shows satisfactory results.

Fig. 4 gives the graphical picture for a number of these compounds. The diagram is drawn in such a way that in every vertical column there are shown the stress-strain curves of a single compound in three directions. Here again it can be seen that the compound 1a shows the straight line strain-stress curve very similar for all three directions. The rest of the compounds show quite a variation in the three directions. The compounds 30, 31, 32, 33, 34, 35, 36, 41, and 42 contain practically the same fillers but show very different elastic properties that indicate the difficulties that may be expected if an attempt is made to obtain uniform material using this composition.

3 *Repeated Tests.* Two kinds of repeated tests were made: (1) To check the original tests, and (2) to determine the influence of high pressures on the elastic properties. The first kind of retests was made on compounds 12, 15, 21, 22, and 36, and the second was made on compounds 1, 1a, 3b, 30b, and 34.

The last-mentioned specimens were subjected to 5000 lb per sq in. hydrostatic pressure for a period of from 8 to 10 hours, after which they were again tested in the regular manner. On some of the specimens this treatment was repeated a few times. Fig. 5 shows the stress-strain curves of some of these specimens.

It may be noted by comparing the specimen 1a on Figs. 4 and 5 that this specimen, which was originally very uniform in all three directions, did not show any appreciable changes.

The specimens 3b, 30b, and 34, that originally did show very pronounced directional properties, changed rather considerably. It seems that in general the application of hydrostatic pressure resulted in a material with less-pronounced directional properties.

CONCLUSIONS

The conclusions based on this research can be summarized as follows:

1 The required specific gravity of 2.4 can be easily satisfied, and using commercially available fillers a large number of compounds can be formed with very different elastic properties. For each filler there seems to be a definite maximum amount of filler that can be added without impairing the workability of the compound in milling.

2 The heavily loaded rubber compounds as a general rule are unstable, cannot be exactly duplicated using the usual manufacturing procedure, show uncertain elastic, highly developed directional and plastic properties, and a large value for the permanent set.

3 It seems that fillers which do not have any chemical affiliation with the rubber and which enter into the compound merely as an ingredient which is mechanically mixed with the rubber are inferior to the compounds with such fillers, as, for instance, litharge, which enter into a chemical combination with the rubber.

4 The compounds containing some fillers in large volumetrical proportions show low values of Poisson's ratio, accompanied with pronounced directional and plastic properties. The satisfactory duplication of these compounds, keeping the value of Poisson's ratio constant, is impossible if the usual manufacturing procedure is used.

5 Compounds having directional properties can be improved by the application of sufficiently high hydrostatic pressure.

6 From the different compounds investigated, the rubber litharge compound was the one that showed the most remarkable uniform elastic properties, not very pronounced directional properties, a practical absence of plastic properties, and a negligible permanent set. This compound can be satisfactorily duplicated using the customary manufacturing procedure and is stable, being influenced very little by aging and repeated loadings. At the same time it possesses a high value of Poisson's ratio.

7 A material that would satisfy the specification should have properties similar to the rubber litharge compound but with a low Poisson's ratio. Such a material seems to be feasible but would require the discovery of the proper filler or fillers. Since no indications are available that would show in what direction a search of such kind is to be made, the possibilities are that a considerable amount of time will be necessary before such material can be found, because of the necessity of preparing and testing a large number of specimens using different fillers at random. Such research cannot be limited to the composition of the compounds only and necessarily have to include an investigation and possibly consequent changes of the manufacturing procedure.

8 The compound 1 was chosen as the material for the model[8] in spite of the high value of the Poisson's ratio in recognition of its remarkable elastic properties, uniformity, and ease of manufacturing.

Very recently, the United States Bureau of Reclamation used some 40,000 lb of this compound for the construction of a water-loaded model of Hoover dam. The bureau already had built and tested very successfully a mercury loaded model made of plaster of paris and celite material. The building and testing of a new type of model made of an entirely different material and the comparison of the test results of both models should give an excellent independent check, which is in line with the highly commendable and unusually thorough procedure established by the bureau in the designing and building of the highest dam ever attempted in engineering history.

AUTHOR'S CLOSURE

In answering Professor Hovgaard's remarks and questions I can remove some possibilities of misunderstanding overlooked at the time the paper was preprinted. In mentioning the modern possibilities of the use of rubber in machinery, as for instance

[8] For the description of this model see A. V. Karpov and R. L. Templin, "Building and Testing an Arch Dam Model," *Civil Engineering*, January, 1932.

for the construction of tires for locomotives, the intention was to stimulate research. The reference contained only the fact that such uses are contemplated in France; results were not given. With the law of elasticity given by Equation [9] of the paper, the practical value of such suggestions can be investigated without costly experiments. A technical description of a rubber tired car is given in *Railway Age* for November 12, 1932.

Considering the way in which I came to the elastic law given in Equation [9], the separation of the results from the mathematical appendix may have caused some confusion. A great number of tests were made with different kinds of rubber. The results showed that a striking agreement between theory and experiment could be reached only if compounds were used nearly isotropic in the unstrained state and free from excessive permanent set by small deformations. The elastic law, Equation [9], is not simply an interpolating formula to reproduce the experiments, but is based on the findings of colloid chemists.

It occurred to the investigators of the elastic properties of colloidal substances that only such materials are capable of undergoing finite strain in an almost reversible manner which possess long principal valence chains of a spiral form. Now we do not wish to say that such long molecules are just the same as spiral springs, but surely in stretching such a molecule an elastic resistance is encountered. Before the cohesive properties of the principal valence chain make themselves felt, the elastic resistance is due entirely to the resistance of the molecules against stretching the spiral form. Assuming now arbitrarily (it is true) for the elastic behavior of an individual molecular spring, the simplest possible elastic law given in Equation [7], and taking a statistical average over all springs, Equation [9] is reached. The difficulties in understanding the deduction for the general case are connected with the impossibility of separating the hydrostatic part of the stress if we give up the continuum conception. To illustrate this law the author took 12 rubber belts and 12 rubber cylinders and subjected them to the same tension and compression test; these experiments only were published in the preprint.

However, I am very glad indeed that in the meantime some remarkable papers appeared confirming my results.[9]

[9] Rudolf Weise, "Die elastischen Konstanten des Kautschuks bei grossen Spannungen," Kautschuk, 1932, pp. 106–110. J. R. Sheppard and W. J. Clapson, "Compressional Stress Strain of Rubber," *Engineering Chemistry*, vol. 24, 1932, pp. 782–790.

The work of Sheppard and Clapson is especially important, because they have found a way to overcome the difficulties of the compression test by testing thin rubber balloons under internal gas pressure. Although they did not deduce an elastic law, they stated that such a law must exist and gave the experimental records so completely that a comparison of formula [9] and the result of one typical experiment of Sheppard and Clapson disclosed a coincidence that is very satisfactory.

In taking a more precise law for the individual molecule, formula [9] could be modified so as to include even the elastic behavior of rubber close to fracture. I did not do that because, for the strains free from excessive permanent set, the simpler formula serves as well.

In answering Messrs. Karpov and Templin, I must say that the individual properties of a rubber compound are not worth while studying, at least not by applied mechanics, unless the results do not show that the behavior is typical. In saying that my results are not applicable to any other compound, the discussors assert indirectly that the elastic behavior of rubber is not a problem of applied mechanics. However, in reading their discussion I could not only not detect a proof of this statement, but also no relation to my own work. They investigate only such small deformations that the classical law of Hooke is good enough. But perhaps the objection is that the mere coincidence of experiment and theory cannot prove the existence of a general law. If the discussors intended to make this objection in reducing the validity of formula [9] to the rubber compound investigated, I can answer with the results of an experiment that I did not publish. The experiment can be performed, with the same results, with any compound free from admixtures lowering the cohesive resistance. A rubber string is loaded, and the period of the small vibrations caused by disturbing the equilibrium somewhat is observed. From the weight, the length of the string, and the stress the apparent modulus of elasticity can be calculated. The modulus of elasticity according to Equation [2a], which is only a special case of Equation [9], can be calculated by differentiating these equations with respect to the strain. The coincidence for small strains shows that the author's mechanical explanation of the possibility of undergoing finite deformations is correct. We maintain therefore that formula [9] is applicable not only to rubber compounds, but to all colloidal substances of like structure.

APM-55-9

An Experimental and Analytical Investigation of Creep in Bending

By GLEASON H. MacCULLOUGH,[1] WORCESTER, MASS.

This paper describes an experimental and analytical study of the phenomenon of creep as it occurs in bending. Creep tests upon a lead beam were made to determine the behavior of originally plane sections during pure bending accompanied by creep. From the results of these tests and of creep tests in tension and compression, a method is devised whereby the results of the usual tensile creep test may be utilized to predict the rate of creep for transverse sections of the beam. The paper concludes with several analytical examples involving creep as it occurs in pure bending.



1—Statement of the Problem

CREEP may be defined as the deformation of a material which occurs with time and which is caused by an externally applied stress. The existing analytical solutions of problems involving creep phenomena, as far as we know, are confined to cases involving tension and compression where neither bending nor actual twist (with one exception[2]) is present, and appear to be based on sound theory.

When one approaches the problem where creep is associated with pure bending, a natural starting point lies in making an assumption concerning the deformation of cross-sections of the beam.

Perhaps the most logical hypothesis is that made in the usual theory of elastic and plastic bending: namely, that cross-sections remain plane during bending. Because we were unwilling to accept this hypothesis as a basis for calculation without further verification when creep was involved, what has proved perhaps to be the major problem of this investigation arose; that is, to ascertain whether cross-sections do remain plane during bending when creep is taking place. The results, it will be seen, indicate that such is practically the case. A second experimental verification which has been attempted is to determine whether, from the results of tensile and compressive creep tests on the beam material, the creep rate of the beam may be predicted.

That the foregoing assumption concerning plane sections is a logical one for the case of pure bending we may see by the following argument. Suppose we take a beam subject to pure bending caused by a stress distribution like that shown at the ends [Fig. 1(*a*)]. The beam may be divided into a number of shorter lengths with the same type of stress distribution on each, of course, as in Fig. 1(*b*). With exactly the same stress distribution on each free body, there is no reason why one should be distorted differently than another. Yet, if all are distorted in the same manner [Fig. 1(*c*)], they cannot possibly fit together *unless all of the end sections have remained plane.*

2—Description of Apparatus

The equipment used in these tests consisted of four principal parts: a lead beam to be tested; a frame and knife edges for mounting the beam; means for applying a constant bending moment over the middle portion of the beam; and an accurate means of measuring the changes in length of the imaginary fibers of the beam. A general view of the set-up is shown in Fig. 2.

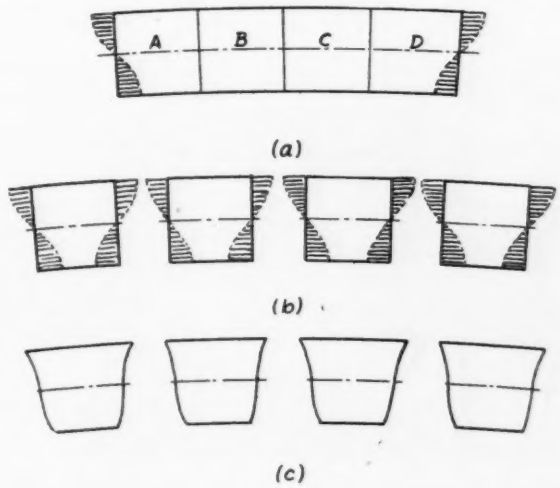


Fig. 1

It was early decided to carry out the experimental problem with lead as the working material, since lead at room temperatures behaves under loads producing creep about as steel and other metals at elevated temperatures. The use of lead therefore allows much more direct experimental methods to be employed than could be used in the case of steel at the necessary high temperatures, and should yield satisfactory qualitative results.

The lead beam was made from a slab of lead, 21 in. long, $7\frac{1}{2}$ in. wide, and $2\frac{1}{8}$ in. thick. A close-up view of the beam is shown in Fig. 3, while the dimensions of the beam and holders as originally designed are given in Fig. 4.

The actual determination as to whether plane sections remained plane was accomplished by measuring the extensions of longitudinal gaged lengths on the surface of the beam which took place under the action of creep in bending. These extensions, when plotted against distance from the center, should give a straight line. This necessitated the use of a strain-measuring device, which proved capable of giving readings under average conditions that varied by about 2 parts in 100,000 for a 10-cm gage length.

The tests were conducted in a room which was entirely free

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[1] Professor of Engineering Mechanics, Worcester Polytechnic Institute. Assoc-Mem. A.S.M.E. Professor MacCullough received the degree of B.S. in 1918 and of M.S. in 1931 from the Worcester Polytechnic Institute. Since 1918, he has served as instructor, assistant professor, and professor at the Institute, and at the same time has been engaged in considerable outside work in engineering. During the year 1931–32, he was granted a sabbatical leave to pursue graduate work in engineering mechanics at the University of Michigan.

[2] See article by R. W. Bailey in *Engineering*, vol. 129, June, 1930, p. 819.

From a dissertation submitted in partial fulfilment of the requirements for the degree of Doctor of Science, in the University of Michigan, 1932.

from vibrations and in which the temperature variations could readily be controlled to within 0.2 deg. C.

3—Data and Results

Three complete series of tests of the lead beam under creep conditions were made. In series No. 1, seven equally spaced gage points were located in each of two vertical lines which were $12^1/_2$

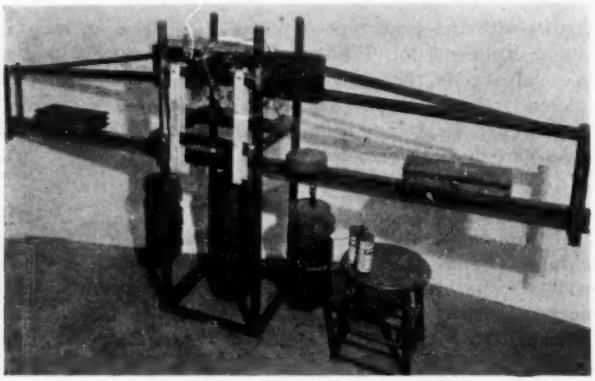


Fig. 2 The Flexure-Test Apparatus

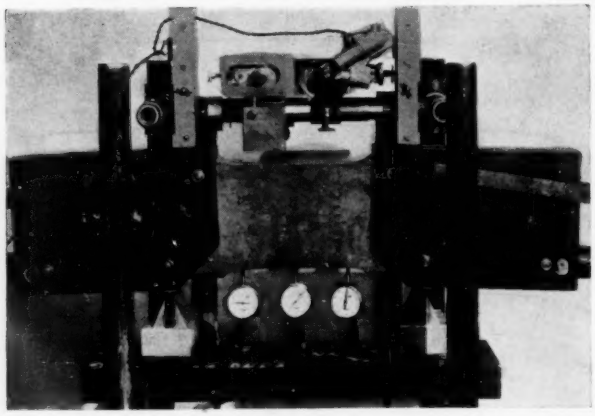


Fig. 3 The Lead Beam and Strain-Measuring Instrument

cm apart. Under the action of creep, the horizontal distances between the three top pairs were increased and those between the bottom pairs decreased. These extensions and contractions were compared with those if the straight and originally vertical lines had remained straight and always passed through the three middle points of each line. The top gage point lagged some 11 per cent behind this line, while the bottom point lagged some 5 per cent, showing that the plane sections did not remain truly plane.

It was suspected early in the course of series No. 1 that the gage sections unfortunately were too close to the supports—only 2 in. from the lower corner of the channel, which must have acted partially as a concentrated reaction—and that localized stress concentration extended even as far as the gage section. These questionable corners were therefore cut back 2 in. as shown in Fig. 3, before the tests of series No. 2 were made. It was also felt that perhaps the gage sections were too near the fillets and that the effect of stress concentration was being evidenced, so the gage distance was reduced from $12^1/_2$ cm to 10 cm in series No. 2. The beam was therefore straightened, remachined, and also given a much longer anneal (of three days in boiling water) before the next series was begun.

The results of this second series of flexure tests are given in Fig. 5. Plots are here shown of four sets of readings taken at various elapsed times. It will be observed that there is less variation from the straight line than in series No. 1, the top gage point now lagging but 7 per cent behind the straight line.

A third series of tests was made after the beam had again been straightened, each fillet cut back 2 in., the faces of the beam relieved so that they did not touch the channel holders until 1 in. back from the edges of the holders nearest the center of the beam, and given a much longer annealing—for seven days.

In this third series of tests two complete sets of gage points were used on each face of the lead beam. Each set consisted of eleven pairs of gage points (instead of seven as in the two previous series); the horizontal distance between the gage points in one set was 8 cm and in the second set 11 cm. All of the gage points in the vertical sections originally 8 cm apart, and all but the very top gage point in the set 11 cm apart, remained on their respective straight lines. This seems to prove rather conclusively that,

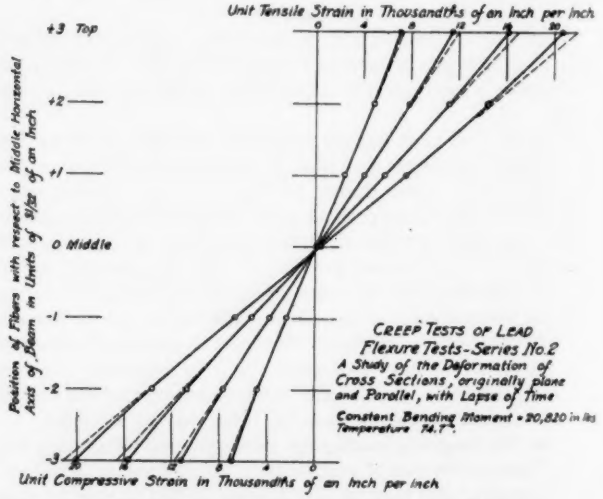


Fig. 5 Creep Tests of Lead (Flexure Tests)

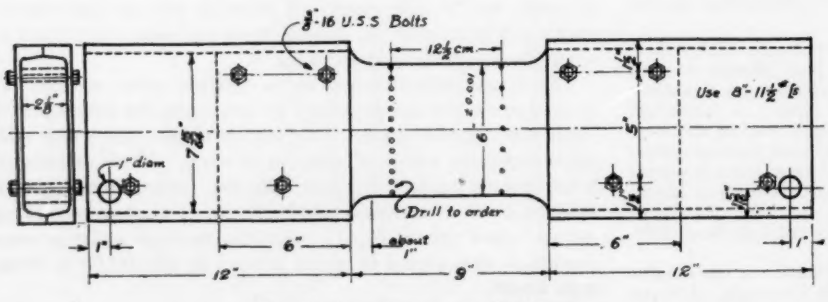


Fig. 4 Lead Beam and Holders

when the gage sections are sufficiently far removed from the clamped ends with their localized strains, plane sections remain plane in creep associated with pure bending.

Relation Between Fiber Extension and Deflection. The ordinary elastic theory shows that the elastic curve of a beam, acted upon by terminal couples only, is bent into the shape of a circular arc. This should also be true for pure bending accompanied by creep since transverse

slices at various places in the beam are all stressed alike. This test shows that the deflection obtained by experiment agrees within less than 1 per cent with that obtained from calculation, based on the assumption that the beam when bent assumes the form of a circular arc.

4—The Tension Tests

The specimens used in these tests were cut from the same lead slab as the beam used in the flexure tests. They were about $13^1/_2$ in. in overall length with a cross-section approximately $^1/_4$ in. sq. The pieces were carefully annealed for some three days in boiling water.

The method of applying the load was simple. Some inexpensive pliers were fitted up with a sort of toggle joint at the handle end of each pair (see Fig. 6) and were used as grips.

The strain-measuring apparatus used in the flexure tests was adapted to the tension tests, and the complete set-up is shown in Fig. 6.

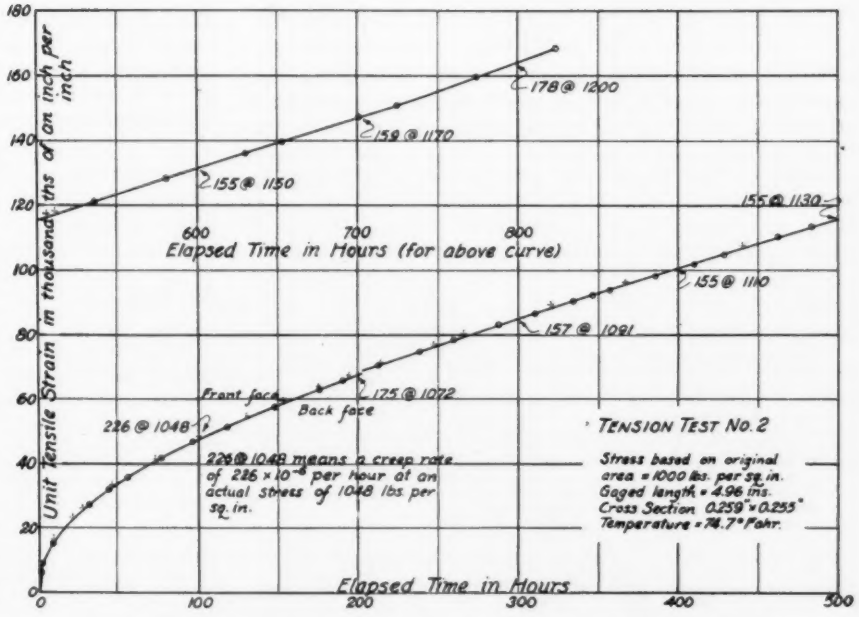


Fig. 7 Results of Tension Tests on a Highly Stressed Specimen (No. 2)

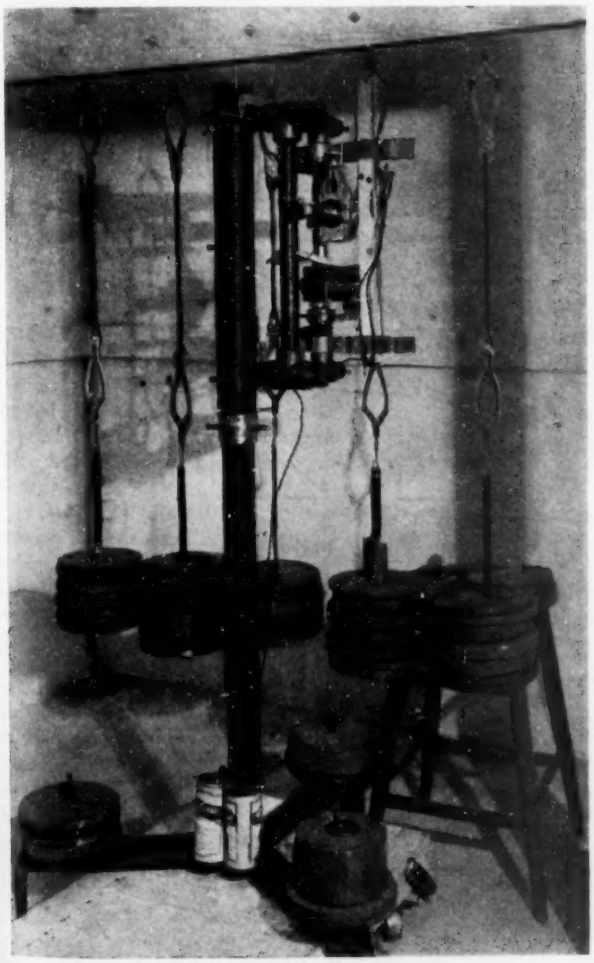


Fig. 6 The Tension-Test Apparatus

The results of the six tension tests run were plotted as in the sample curve shown in Fig. 7. This curve represents the results for a highly stressed specimen, No. 2. Three distinct stages are visible here: a stage of increasing creep rate, an intermediate stage where the creep rate is constant at 155×10^{-6} in. per in. per hr, and a final stage of increasing creep rate, doubtless showing the effect of the greatly reduced area, which is some 15 or 16 per cent less than at the beginning. The actual stresses indicated in Fig. 7 at the end of 100-hr intervals are computed on the basis that there is no volume change.

Another observed phenomenon is worth noting. Specimens Nos. 1 and 6 were subjected to approximately the same stress, 735 lb per sq in., but No. 6 had been overstrained initially so that it had been stretched about 2.2 per cent of its gaged length. By an examination of the two curves for these specimens (not shown), it was found that the creep rates of the two for equal amounts of strain are nearly identical. This agrees with a phenomenon observed by Dr. Everett[3] in his work on creep in torsion: namely, that the same minimum rate of strain could be reached more quickly by first overstraining the test specimen.

Fig. 8 shows the actual tensile stress, based on the reduced area, plotted against creep rate after various elapsed times upon semi-logarithmic paper. The resulting curves more or less assume the straight-line form which has sometimes been assumed by other experimenters, at least for the plot of the minimum creep rates. This means that the creep rate is related to the actual stress by an equation of the form

$$c = \epsilon^{a + b\sigma} \text{ or } \log_\epsilon c = a + b\sigma$$

where c = creep rate, σ = actual stress, ϵ = Napierian base of logarithms, and a and b are empirical constants.

5—The Compression Tests

Out of some five tests made, two were regarded as rather satisfactory, the amount of buckling produced not being excessive. These two tests gave creep rates in fair agreement with those obtained in tension.

[3] See F. L. Everett, "Strength of Materials Subjected to Shear at High Temperatures." Trans. A.S.M.E., APM-53-10, vol. 53 (1931), p. 126.

6—Correlation of the Results of the Flexure and Tension Tests

We are now in a position to study whether there is a possible way whereby the results of the common tensile creep tests may be utilized to predict the results that would be obtained in pure bending. In this particular study it is to be remembered that we

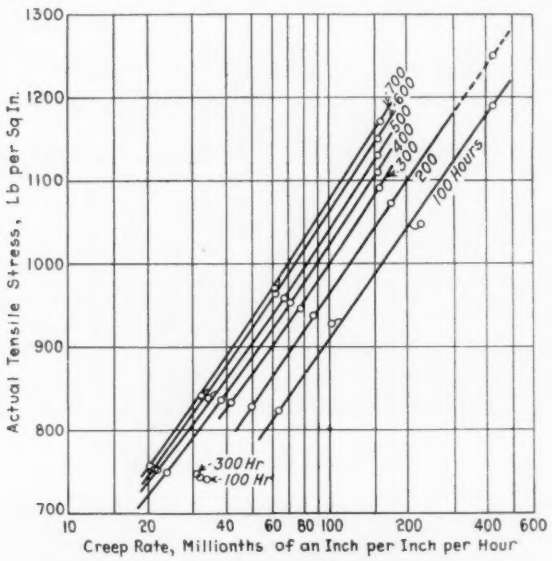


Fig. 8 Creep Tests of Lead (Tension Tests)

are restricting ourselves to the case of pure bending—bending unaccompanied by shear.

The general method of attack is this: Knowing the creep rate at the outside fiber of the beam and the manner of deformation of the cross-sections, we can find the creep rates for various fibers within the beam. If we should consider that the stresses producing these creep rates were equal to the stresses that would cause the same minimum rate of creep in tension or compression, then from a plot of "stress versus minimum rate of creep in tension," we could determine the corresponding stress in bending. Knowing the stress distribution for a few points in the cross-section of the beam, it is an easy matter to compute the moment of resistance of the bending moment acting on the section.

But the validity of this "minimum rate of creep" simplification is not settled. Perhaps the most puzzling problem which occurs in this attempted correlation, therefore, is that of deciding what creep stress to select from the tensile creep results corresponding to any rate of creep that we may select. Unfortunately the rate of tensile creep is not only dependent upon the tensile stress acting but also upon either the amount of strain that has occurred or upon the elapsed time since loading. Thus there are four variables mixed up in the problem: stress, strain, time, and a derived variable, the rate of creep. Consequently any one is dependent upon two others, not upon a single one. An expedient frequently used for reducing the number of variables is to tie up the *minimum rate of creep* with the stress, because then to every stress there corresponds but one (minimum) rate of creep. It is to be observed further that this minimum rate of creep occurs wherever the strain-time curve remains straight (which is the same as having a constant creep rate) or at a point of inflection.

From plots of creep for the outside fiber in bending against elapsed time, made by the author and not shown in this paper, it is seen that such creep-time curves assume nearly a constant rate of creep at the end of some 200 hr. Similarly at the end of 600 or 700 hr in the case of tensile creep, it is seen from a study of Fig. 8 that there is not much change in creep rate with increasing lapses of time. In fact, in series No. 2 of the flexure tests, the creep rate diminishes by the same percentage between 100 and 200 hr as between the 600th and 700th hour in the tension test. Since at both 150 or 200 hr elapsed time in bending and 700 hr elapsed time in tension, approximate minimum-rate-of-creep conditions prevail, it will therefore be assumed in the following calculations that the stress set up in bending for a given rate of creep equals the stress in tension which corresponds to the same rate of creep at an elapsed time of 700 hr.

7—Computation of the Bending Moment as Predicted From the Results of the Tensile Tests

Let us now endeavor to compute the moment acting on the beam in series No. 3 of the flexure tests at the end of 165 hr, using the creep rate observed for the top fiber of the "west" face with an initial gage length of 8 cm and the results of the tension tests as plotted in Fig. 8. This creep rate is observed to be 115.7×10^{-6} in. per in. per hr, but is really for a point 2.7 per cent of the half-depth below the top. At the top it would be about 119×10^{-6} in. per in. per hr. The creep rates at other fibers can be found by proportion on the assumption that displacements vary as the distance from the center.

Distance of fiber above center in inches	Creep rate at this fiber	Corresponding tensile stress, from "700-hour" curve, Fig. 8	Multiplied by distance from center
2.97	119×10^{-6}	1110	3290
2.475	99×10^{-6}	1070	2650
1.98	79×10^{-6}	1025	2030
1.485	$59\frac{1}{2} \times 10^{-6}$	970	1440
0.99	$39\frac{1}{2} \times 10^{-6}$	885	875
0.495	20×10^{-6}	755	375
0.00	0	0	0

Knowing the stress-distribution curve as of (*a*), Fig. 9, the moment can readily be computed for a beam of unit width by use of the σy distribution curve of (*b*), Fig. 9, where σ is the unit

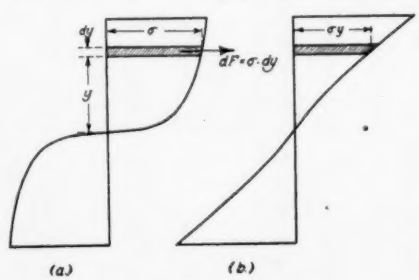


Fig. 9

stress. For, by reference to (*a*) it is seen that the total moment will be given by

$$M = \int (\sigma dy)y = \int (\sigma y)dy$$

and is nothing other than the area under a curve, such as Fig. 9(*b*), whose ordinates are y and whose abscissas are σy.

This area is readily computed by the use of Simpson's rule. Applying this rule, the area is

$$A = \frac{1}{3}\frac{2.97}{6}\left[3290 + 4\,(2650 + 1440 + 375) + 2\,(2030 + 875)\right]$$

$$= 4450 \text{ lb}$$

When multiplied by the width of the beam, 2.094 in., and doubled to care for the lower half of the section, the total moment is found

to be 18,650 in-lb, a value which is 97.2 per cent of the 19,170 in-lb actually acting on the beam.

A similar computation for series No. 2 gave a corresponding percentage of 89.8. The improvement in series No. 3 is attributed to the effect of the longer annealing upon softening the lead prior to this series.

It is to be noted that these calculations are based on the use of the 700-hr curve for stress versus rate of creep in tension. If the 200-hr curve had been used, for instance, the computed moment would be about 87 per cent of the actual moment. Therefore further investigation is desirable to settle just where to enter the family of "stress versus rate of creep" curves for those cases where the beam has not settled down to a steady rate of creep. In such a study it may even be found advisable to plot "stress versus rate of creep" curves with strain as the parameter, rather than time, in the family of curves.

In most practical problems, however, the elapsed time is sufficiently long for a steady rate of creep to have been attained, and the relation between stress and rate of creep to be used in such problems is that given by the "stress versus minimum rate of creep" curve.

Practically, the inverse of the problem just discussed is of greater interest: namely, knowing the moment acting on a beam which is in a state of creep, to determine the rate at which it is creeping, having as working data the results of tension and compression tests. This problem is discussed as an example of analysis in Art. 8, and the reader is referred to the method of solution there presented. Suffice it to say that if this method is applied to the problem discussed above, the computed and experimental creep rates will differ by some 14 per cent, the reason for the discrepancy being that creep rate is very sensitive to changes of load.

8—The Problem of the Stress Distribution and Rate of Creep in a Beam Subjected to Pure Bending

Several examples which have to do with creep in bending will now be discussed. They are all based on the assumption that plane cross-sections remain plane.

Case I—For a Rectangular Cross-Section and the Same Creep Characteristics in Tension and Compression. As an illustrative example, the beam of the experimental problem will be dealt with.

Given the lead beam of 5.940 × 2.094 in. cross-section acted on only by a bending couple of magnitude 19,170 in-lb (for stress-creep relations, use the 700-hr line of Fig. 8), to find the stress distribution and the rate of creep and rate of "deflection" of the beam.

The method of solution is an indirect one—the inverse of that used in Art. 7. It is based on the theory that plane cross-sections remain plane. The solution is obtained by trial, but the labor involved may be much reduced by starting with a good estimate of the value of the outside-fiber stress in the beam. A first estimate may be obtained by choosing a trial stress which is some 73 per cent of that obtained by the usual elastic theory, using the common formula, $\sigma = Mc/I$. The factor 73 per cent is arrived at from a study of the problem of Art. 7, and really assumes that the stress distribution is always of the form obtained in Art. 7. The *correct* choice of outside-fiber stress is obtained when the bending moment computed from this choice agrees with the given moment.

Solution. The "elastic" stress is

$$\sigma = \frac{Mc}{I} = \frac{M}{S} = \frac{19{,}170}{12.33} = 1555 \text{ lb per sq in.}$$

The assumed "creep" stress at the outside fiber is

$$\sigma = 0.73 \times 1555 = 1135 \text{ lb per sq in.}$$

The corresponding tensile creep-rate from the 700-hr curve of Fig. 8 is 134×10^{-6} per hr.

If plane cross-sections remain plane, the creep rates at fibers which divided the half-depth into six equal parts are, beginning at the outside, 134, 112, 89, 67, 45, 22, and 0. The corresponding stresses from Fig. 8 are 1135, 1100, 1050, 990, 915, 770, and 0. As in the problem of Art. 7, the best way to compute the moment is to multiply these values by their distance from the neutral axis and then plot as in Fig. 9(*b*). The corresponding products are 3370, 2720, 2080, 1470, 905, 380, and 0. Application of Simpson's rule gives

$$\text{Area} = \frac{1}{3}\frac{2.97}{6}\Big[(3370 + 0) + 4(2720 + 1470 + 380) + 2(2080 + 905)\Big] = 4560 \text{ lb}$$

Since this area represents only the moment of a section 1 in. wide and for the top half, the total moment is

$$M = 4560 \times 2.094 \times 2 = 19{,}080 \text{ in-lb}$$

If this is not in close enough agreement with the actual or given moment of 19,170 in-lb, a second approximation may be made using a new outside stress of magnitude (19,170/19,080) × 1135 or 1140 lb per sq in. A repetition of the above procedure using this new trial stress will doubtless give the correct moment of 19,170 in-lb. For this final outside stress, the corresponding creep rate is 136×10^{-6} in. per in. per hr for the outside fiber. This is the creep rate for the outer fiber which we set out to calculate. The stress distribution desired is practically that computed above.

Knowing the rate of creep of the outside fiber, it is a simple matter to compute the deflection, or the amount the beam is bowed away from a chord of length l. Assuming $l = 6$ in., and a creep rate of 136×10^{-6}, the relative rotation of plane sections 6 in. apart would be

$$\frac{136 \times 10^{-6} \times 6}{2.97} = 275 \times 10^{-6} \text{ radians per hr}$$

It is readily shown that the rate of change of deflection from the middle of a chord of length l under the action of a constant moment over the length l is

$$\frac{dy}{dt} = \frac{l}{8}\frac{d\vartheta}{dt}$$

where $d\vartheta$ is the angle of rotation between plane sections. For the assumed chord length, the deflection works out to be

$$\frac{dy}{dt} = \frac{6}{8} \times 275 \times 10^{-6} = 206 \times 10^{-6} \text{ in. per hr}$$

or 0.00495 in. per day.

If a known analytical relation exists between the stress and the resulting minimum rate of creep, such, for instance, as the familiar form, $\sigma = a + b \log_e C$, where C is the linear creep rate in tension or compression and the expression itself is valid for both tension and compression, then the problem may be solved analytically in some cases.

Case II—For a Rectangular Cross-Section and Unlike Creep Characteristics in Tension and Compression. This problem is somewhat more difficult than that of case I in that the position of the neutral axis is not known. In the following, a method of solution is outlined which is analogous to that[4] in pure bending when the material does not follow Hooke's law.

[4] For this method of solution, see, for instance, Timoshenko's "Strength of Materials," vol. 1, p. 233.

In this case we start as in Case I by assuming a reasonable stress in the tension fiber, greater than 73 per cent of the elastic stress if the creep rate in compression is more than in tension, and less than 73 per cent if the opposite is true. Under the condition that the compressive creep rate is the higher, the neutral axis will move away from the compressive fibers and the plane section will appear straight as in Fig. 10(*b*). Let us find the strain e_1 which corresponds to this stress, using the plot determined experimentally between tensile stress and minimum rate of creep.

Fig. 10(*c*) is similar but represents *any* strain-distribution curve following the linear law. In this last figure, let us locate the strain e_1. The stress-distribution curve corresponding to this linear deformation is plotted alongside it in Fig. 10(*d*) with both

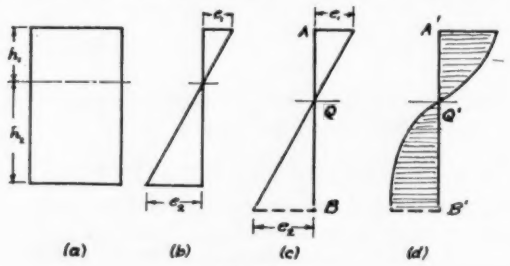


FIG. 10

the tensile and compressive portions shown. To locate correctly the neutral axis, assuming of course that no axial stresses are present, the condition that must be fulfilled is that the resultant force due to the stress distribution below the neutral axis must be equal and opposite to that above. Since these forces are proportional to the areas under the curves below and above the neutral axis, we first find the area between Q' and the top and then establish the position of the strain e_2 so that area between it and Q' shall be equal to that above. This can be done accurately enough by graphical means. The position of the neutral axis in the beam can evidently be located by the proportion $e_1/e_2 = AQ/BQ = h_1/h_2$.

Knowing the location of the neutral axis, the bending moment can be readily computed as in the preceding case. In this way one gets a moment that corresponds to the assumed stress in the top fiber. This moment of resistance probably will not agree with the given moment. A couple of further repetitions of this procedure will establish two more moments corresponding to other assumed stresses. If our choice of stresses for the top fiber has been anywhere near correct, these three points will enable us to plot a curve showing the relation between moment and stress in top fiber, and this curve can then be used to spot the top-fiber stress corresponding to the given moment. A final calculation based on this stress will undoubtedly give the correct stress distribution and creep rate.

Case III—For a Cross-Section Symmetrical With Respect to the Vertical Axes but Unsymmetrical With Respect to the Horizontal Axis. The creep characteristics in tension and in compression may or may not be alike.

This problem is even more complicated than the foregoing. The procedure will be about as follows:

The cross-section of the beam may be something like that of Fig. 11(*a*). Select arbitrarily a neutral axis, below the center of gravity of the cross-section if the creep rates in tension (the top fiber) are higher than in compression, otherwise above. Draw the linear strain relation of Fig. 11(*b*). Alongside this, plot the corresponding stress-distribution curve obtained from the experimentally determined "tensile stress versus minimum rate of creep" curve. Since the section is of non-uniform width, multiply each of these stresses by the width of the section at the corresponding point. The area under this curve represents the total force, and, as before, the resultant force or area above the neutral axis must equal that below. If the two areas work out to be equal, then the neutral axis is correctly located. If not, repeat with a slightly different assumed location for the neutral axis. When the neutral axis is correctly determined, then the moment can be computed as in case I.

A repetition of this procedure a couple of times for different

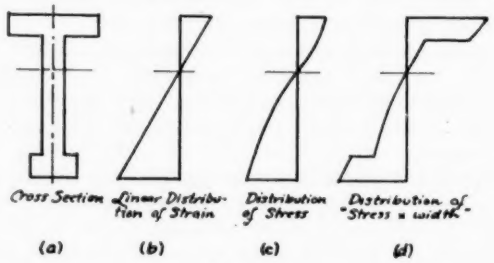


FIG. 11

assumed stresses in the outside fiber will enable one to get the corresponding moments and to interpolate between these moments and find the stress that will correspond to the given moment.

9—A SUMMARY OF CONCLUSIONS

From this investigation, the following conclusions may be drawn.

1 That transverse sections remain plane and normal to the axis of the beam during pure bending.

2 That deflections may be computed on the basis that the beam is bent into the form of a circular arc and that, conversely, deflection readings may be used for a rough determination, at least, of the creep of the fibers in pure bending.

3 That the minimum rate of creep in tension may be more quickly and just as accurately reached by first overstressing the specimen.

4 That the way is pointed toward a method of analysis whereby the results of tensile creep tests may be used to predict the behavior of a beam subjected to the action of creep under pure bending.

5 That the rate of creep is greatly affected by the previous mechanical and thermal treatment of the lead.

6 That the gage sections must probably be taken far away from fillets and other changes of section.

7 That the method of clamping the ends in pure bending probably plays a part in the deformation produced on cross-sections in the neighborhood of the clamping device.

ACKNOWLEDGMENTS

The author is particularly indebted to Prof. S. Timoshenko, who suggested this topic of investigation, and to the University of Michigan for providing the necessary equipment. Others, too numerous to mention, have also been of great assistance.

APM-55-10

The Creep of Metals

By A. NADAI,[1] EAST PITTSBURGH, PA.

In the following, a number of idealized cases of plastic flow of a more or less simple nature are discussed with primary intention of throwing light on how the speed of yielding influences the stresses under which metallic bars are permanently stretched or twisted at normal or at elevated temperatures.

I—Introduction

DURING the last few years considerable work has been done in the research institutions of the machine industry and in the laboratories of universities in the endeavor to obtain, mainly for important practical purposes, more reliable experimental data regarding the mechanical and physical properties of metals at high temperatures. The boiler tubes and heavy drums of the steam plant, the containers for the chemical and cracking industries, and various parts in steam turbines or internal-combustion engines have to withstand the prolonged action of large stresses or high pressures while they are at the same time heated to increasingly high temperatures. This has not only brought about an urgent demand to develop new materials of construction—and particularly new alloys—which would better resist the severe service conditions encountered in the modern boiler and engine plants than the materials formerly used, but has also put before the engineer a new task, namely, to search for facts which would help him to establish quantitative measuring scales for a comparison of the mechanical properties on which the behavior of the materials seems to depend at elevated temperatures. This has also largely served to draw more recently the attention of a number of investigators to certain experimental facts, which, although well known in connection with other physical phenomena concerning the slow movements of continuously distributed masses, were perhaps not so clearly conceived at first, and which for good reasons did not need to be considered in previous instances where the temperatures were low and where rates of flow could be neglected.

In comparing low and high temperatures for various metals definite ranges of temperature are not considered but rather a relative scale in which the absolute temperature in question is compared with the melting point of the material investigated, the ratios of the two being, according to P. Ludwik, the quantities to be compared ("homologous temperatures").

On the other hand, it is known that the forces necessary to produce plastic deformations at low temperatures may also depend to a certain extent on the speed of flow. Although at comparatively low temperatures this influence may in many cases be neglected, in a few it may become more pronounced. The rapid deformations resulting from an impact are such a case. The following statements may therefore also possess a certain value in dealing with rapid deformations at comparatively low temperatures.

Some of these facts concerning the behavior of materials which apparently are becoming increasingly important under present conditions, had already attracted the attention of early observers of the mechanical properties of solid matter, such as Clerk Maxwell and Lord Kelvin. These investigators created terms and introduced names for certain of these properties, which may prove still useful at the present time.

In connection with these phenomena, it may perhaps also be of interest to mention another set of facts where the extremely slow permanent changes of shape of solid bodies have produced effects. In nature, their consequences are evident on a large scale in the extremely slow movements of the strata of the upper earth crust, and, taking into consideration the extended period in which these movements occur, extremely small differences of stress, and in consequence infinitely small disturbances in the equilibrium conditions of the upper layers of the earth, must have been sufficient to produce some of the gigantic and magnificent effects visible in the signs of the uplifting of mountain chains.

Among the more recent investigations on the flow of metallic materials under stress must be mentioned particularly the work by H. J. Tapsell reported in his recently published book "Creep of Metals,"[2] by R. W. Bailey[3] and G. I. Taylor[4] in England, by P. G. McVetty[5] and by the collaborators of S. Timoshenko[6] in this country, by R. von Mises, L. Prandtl, T. von Kármán, and H. Hencky and by the K. W. Institute for Steel Research and by others in Germany. Further data on the experimental work on creep of metals are available in the volume published by the A.S.T.M. and A.S.M.E. in 1931 and embodying a report on the symposium on the effect of temperature on the properties of metals held in Chicago, June 23, 1931, to which further reference must be made. The phenomena of rapid and slow movements in solids have recently been the subject of most interesting investigations in their applications to geophysics by H. Jeffreys[7] and by B. Gutenberg,[8] to which special attention is called. With reference to the phenomena of elastic after-effects, internal friction, etc., which will, however, be excluded from treatment in what follows, the review by H. Fromm in his two articles on the limits of elasticity and elastic hysteresis

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Oxford University Press, London, 1931. 279 pp.

[3] R. W. Bailey's work is reported frequently in Tapsell's book mentioned above. Among some of his more recent papers reference must be made here, particularly to: "Creep of Steel Under Simple and Compound Stresses," World Power Conference, Tokyo, 1929; "Strength and Creep of Thick-Walled Metal Tubes," and "Testing of Materials for Service in High-Temperature Steam Plant," Proc. Inst. M.E., 1932.

[4] G. I. Taylor and H. Quinney, "The Plastic Distortion of Metals," Phil. Trans. Roy. Soc., London, Sec. A, vol. 230, p. 323, Nov. 1931; cf. also review of this paper in *The Metallurgist* (supplement to *The Engineer*), Feb., 1932, London, p. 25, and following issues.

[5] "Creep of Metals at Elevated Temperatures," Proc. A.S.T.M., vol. 28 (1928), part II, pp. 60–79. P. G. McVetty and N. L. Mochel, "The Tensile Properties of Stainless Iron, etc.," Trans. A.S.S.T., vol. 11 (1927), pp. 73–100, and paper by T. D. Lynch, N. L. Mochel, and P. G. McVetty in Proc. A.S.T.M., vol. 25 (1925), part II.

[6] F. L. Everett, Trans. A.S.M.E., Applied Mechanics Division, 1931.

[7] "The Earth, Its Origin, History, and Physical Constitution," Univ. Press, Cambridge, England, 2nd ed., 1929, p. 263.

[8] B. Gutenberg and H. Schlechtweg, "Viskosität und innere Reibung fester Körper," *Phys. Zeit.*, vol. 31 (1930), p. 745.

in the "Handbuch der physikalischen and technischen Mechanik" should be mentioned.[9] Finally, a number of papers have been published in the *Journal of Rheology*[10] in the last few years which mostly deal with the plasticity of non-metallic materials and cannot be referred to here in detail.

Although the phenomena indicated in solids have thus been observed very carefully and studied from various sides, including also the flow of rocks in nature, it seems that their mechanical laws need further and perhaps more detailed analysis, an introductory contribution to which is offered in the present paper.

II—Phenomenological Distinction Between Speed-Independent and Speed-Dependent Plasticity

1 As already indicated, a variety of cases of the permanent or plastic deformations of solids can be treated with sufficient accuracy without paying any attention to the variable time. If, for example, a tensile test with a ductile metal, such as steel or copper, is made on a testing machine and a stress-strain diagram is taken at a comparatively low speed and at normal temperatures, it will not make a difference of more than a few per cent in the yield stress whether the bar is stretched at one speed or at another.[11]

Under the comparatively low speeds used in testing materials at room temperature, very little effect will be exhibited, at least for materials having high melting points. For lower-melting metals, which start to recrystallize or to lose stress after permanent deformation at normal temperatures, this is, however, not the case. Ludwik[12] probably first called attention to this fact and suggested more quantitative relations. He tested tin wires, stretching them under constant speeds of the moving heads of a testing machine and found that the *true* stresses σ obtained by dividing the load P by the actual cross-sectional area A of the bar, while the load was acting, under which a given extension ϵ per unit of length (defined as $\epsilon = (l - l_0) : l_0$, where l_0 is the original length of the wire or bar and l the actual length) was produced, depended on the rate of flow $u = d\epsilon/dt$ or the unit extension per unit of time. With increasing u the yield stress σ becomes larger for the same ϵ. Ludwik suggested a logarithmic expression for σ as depending on u at a given value of ϵ. In a slightly changed form this is

$$\sigma = \sigma_1 + \sigma_2 \log_e \frac{u}{u_0} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where σ_1, σ_2, and u_0 are constants. H. Cassebaum,[13] while investigating the mode of yielding of mild-steel bars at room temperature, arrived independently at similar conclusions. His results, as far as they refer to mild steel, must, however, be taken today with precaution, because when computing ϵ or $d\epsilon/dt$ from his observations, he tacitly assumes that the whole length of the bar flows uniformly, which could not well be the case probably in many of his tests, on account of the effect of the upper and lower yield points characteristic for this material. These tests were repeated at room temperature by H. Deutler[14] with improved apparatus and with rates of flow u ranging from the ordinary speeds of a slow tensile test up to such as can be produced by impact (also using tensile bars). Within a range of speeds of from $u = 1$ to $u = 1{,}000{,}000$, Deutler verified the same relation, which also might be expressed by saying that if at a given unit elongation ϵ the stress ordinates $\sigma = \sigma_1, \sigma_2, \sigma_3 \ldots$ increase according to the terms of an arithmetical progression then the corresponding rates of flow u required to produce such stresses will satisfy a geometric progression of terms $u_1, u_2, u_3 \ldots$. To obtain the stresses $\sigma_1 = \sigma_0 + \Delta\sigma$, $\sigma_2 = \sigma_0 + 2\Delta\sigma$, $\sigma_3 = \sigma_0 + 3\Delta\sigma$, . . . for a given ϵ, the speeds u or rates of flow required must be chosen according to the relations $u_1 = u_0 c$, $u_2 = u_0 c^2$, $u_3 = u_0 c^3$, etc. From this it follows that the influence of the speed of deformation in slow tests on such materials as mild steel tested at normal temperature is comparatively insignificant, provided the speeds do not reach the order of magnitude of impact speeds, thus justifying the establishment of what may be termed the "static theory of plasticity."

However, conditions may change if the temperatures increase, with the result that the same changes in speed of deformation, if the temperatures are sufficiently high, may be followed by a very considerable change in the acting stress. Conversely, a slight change in stress which at low temperatures had a pronounced effect on the speed of deformation, at sufficiently high temperatures will affect but little the rate of flow. If, for example, it is slightly lowered from the value of a yield stress maintained at low temperature, yielding will cease, while, if relations similar to those set forth in Eq. [1] also hold at the higher temperatures, after a certain decrease of stress, yielding may not cease and the bar may creep and deform further.

R. W. Bailey in his paper presented before the World Power Conference, 1929, quoted above, arrived also at a logarithmic expression for the relation between stress and rate of creep at elevated temperatures. He compares creep rates somewhere near the minimum creep rate observed on a time-elongation curve and finds a straight-line relation when plotting stress values against logarithms of creep rates. He uses this information when discussing various cases of stress—for example, the distribution of stress in tubes subjected to internal pressure at elevated temperatures.

III—Load and Stress Surface for Tension. The Infinitely Slow Stress-Strain Diagram for Tension

Another consequence of these facts concerning the slow permanent deformation of metals or other solid materials is that while it was sufficient in a static test to assume the stress as depending on a single variable, namely, the unit elongation ϵ, and to plot a "stress-strain diagram" characteristic for the given material—at least one for each of the simple states of stress, such as pure tension, pure compression, or pure shear—now more independent variables are needed. One of them has already been introduced, namely, the rate of flow $u = d\epsilon/dt$. The question that immediately arises is how many variables or independent quantities are required to describe a series of states of plastic flow.

Let a number of bars of the same material be stressed from the unstressed state at the same temperature but under varying rates of flow u, each bar at a constant rate u. This can be done in some of the testing machines with a certain accuracy when the machine has one head which practically moves but little while the load increases, while the other head is moved by an electric motor with a constant speed (screw-drive type of testing machine); such conditions, however, may only be approached if the bar is stretched permanently to a considerable extent. If a gage length of the bar is thus instantaneously recorded while it stretches uniformly under increasing load (but not by stopping the test and by measuring the length on the unloaded bar)

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[9] Vol. 4, 1st Half. 2nd Part, pp. 359–551. Verlag Ambrosius Barth, Leipzig, 1931.

[10] Particularly referred to are the papers by M. Reiner.

[11] More definitely, in a paper (quoted later) by Deutler it is shown that a large increase of speed, from 0.0011 to 7.68, will not change the yield stress more than about 6 per cent for mild steel.

[12] "Elemente der Technol. Mechanik," Berlin, J. Springer, 1908. See also later papers by P. Ludwik.

[13] Dissertation, Univ. of Göttingen, 1911. See also *Annalen der Physik*, vol. 4 (1911), no. 34, p. 106.

[14] "Experimentelle Untersuchungen über die Abhängigkeit der Zugspannungen von der Verformungs-Geschwindigkeit," *Phys. Zeit.*, vol. 33 (1932), p. 347.

and force values P are taken, a "stress-strain diagram" can be plotted in the shape of a curve $\sigma = f(\epsilon)$ taken under the given rate of flow $u = d\epsilon/dt = \text{const.}$, where $\sigma = P/A$ and A is the actual cross-sectional area (Fig. 1). For each speed of stretching u there will be a diagram. Combining the individual diagrams from the parallel sections into a common surface at a given temperature, a curved surface will be obtained which may be called the stress surface of the material at this temperature for tension, and in which the stress σ will appear as a function of the two independent variables ϵ and u (Fig. 1):

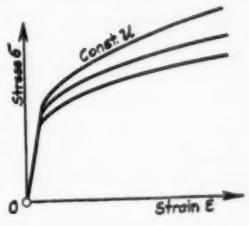


FIG. 1 STRESS-STRAIN CURVES FOR CONSTANT SPEEDS (u = CONST.)

$$\sigma = F(\epsilon, u) \ldots\ldots\ldots [2]$$

It will furthermore be assumed that the total unit strain ϵ observed is an entirely permanent one. A unit strain ϵ may in general consist of an elastic portion ϵ', which may be assumed to be proportional to the acting stress $\epsilon' = \sigma/E$, where E is the modulus of elasticity of the material, and of a permanent portion designated by ϵ''. Considering first only strains in which the plastic portion ϵ'' is predominant, and neglecting the elastic part, the state of plastic flow under these simplified conditions will depend on two quantities, ϵ and u, and the third, namely, the stress σ, will be given by the value of the corresponding ordinate σ of the stress surface of the material, $\sigma = F(\epsilon, u)$. Now if a point P having the rectangular coordinates x, y, z in space is moving on a curved surface, $z = F(x, y)$, the path along which it moves on this surface may be determined, for example, if the coordinates x and y are known functions of the time t. By eliminating the time t from x and y, a second condition will result, or an equation $y = \psi(x)$, which is the horizontal projection of the path of P on the surface, on the xy-plane. Similarly, if for a given material the stress surface is given: $\sigma = F(\epsilon, u)$, and, for example, it is known how the unit elongation ϵ changes with the time t, then $u = d\epsilon/dt$ will also be given, and by eliminating from both expressions the time t, the projection of the path of a point P will be given, which determines a curve on the surface $F(\epsilon, u)$. To describe a sequence of modes of plastic flow—that is, a "special test"—it will be necessary to prescribe besides the known shape of the stress surface $\sigma = F(\epsilon, u)$ a second condition, which may, for example, be a function of the form

$$u = \psi(\epsilon)$$

In practice, the additional condition which may determine a test may appear in various forms. One frequently met is that the load applied to keep the bar under stress is held constant; or the condition may be given more implicitly by the nature of the mechanism by means of which the bar is stretched and the forces are measured, new loads applied, speeds of stretching changed, etc.

An assumption on which the introduction and the use of such a "stress surface" $\sigma = F(\epsilon, u)$ for the tensile test is implicitly based must be particularly mentioned here. It is assumed that the phenomena to be described in the following are of such a nature that the strain ϵ and rate of flow u are the only factors and independent variables that determine the state of stress required to produce them at an instant t, *but that the time itself at which the quantities ϵ, u, σ are observed has no influence on the deformation.* In other words, if a given strain ϵ at a given rate of flow u is produced, then the corresponding yield stress of the material will always be the same, no matter how long the material was left under the influence of the temperature at which the test was made. Now it is well known from observations—for example, about hardness—that previously cold-worked materials may soften with time without being stressed, if subsequently exposed to comparatively low, but slightly elevated temperatures.

The author is particularly indebted to Mr. P. G. McVetty for having made available to him very valuable observations regarding this softening of previously work-hardened material, which are not yet available in published form. Further data concerning observations of an analogous nature on the softening of copper and other metals have been published by G. P. Halliwell and N. B. Pilling.[15] What is essential at the present moment is that these and similar phenomena have to be excluded from the following considerations. Mathematically speaking, however, it may be added that they may perhaps find their adequate expression later, including in the stress function $F(\epsilon, u)$ besides ϵ and u the variable time t, by writing explicitly

$$\sigma = F(\epsilon, u, t)$$

It may be said—and this will appear from what follows more conclusively—that portions of the theory of creep of metals under a simple state of stress may be treated in the first approximation by applying the kinematics of the movements of a material point P having the rectangular coordinates x, y, z on a given surface in space and moving along a prescribed path on this surface. As the basic coordinates x and y may be chosen in the case of the creep problem the strain ϵ and the rate of flow $u = d\epsilon/dt$, and as ordinates z may appear the yield stress σ. To any given path on the curved surface there will correspond a special yielding test, and the projection of this special curve or path on the σ, ϵ-plane is the "stress-strain diagram" corresponding to the special test.

To each constant temperature there will correspond a special surface $\sigma = F(\epsilon, u)$. To make the picture more complete, one may, if in certain cases it is required, also consider such cases of movements of a point P on a surface, in which this latter is either displaced without distortion relatively to a base or in which it will be distorted gradually with time t in a predetermined way. It seems obvious that particularly the second kind of movements will probably become complicated. Using the analogy—the cases where through chemical changes such as the slow precipitation of a component in an alloy, age hardening, etc., or through softening the hardness of the alloy or the yield stress changes with time—this might perhaps be described by gradually allowing a distortion of the stress function or surface $\sigma = F(\epsilon, u)$ with time t. By multiplying, for example, the right-hand side of the equation by an expression containing only the time t:

$$\sigma = F_1(t) \cdot F_2(\epsilon, u)$$

and choosing a particular form for $F_1(t)$, perhaps such cases might be treated in which the yield stresses or the hardness due to internal changes of the structure first increase and later again decrease (softening, age hardening, etc.) while the test goes on or a piece of a construction is subjected to a state of stress of long duration.

To the movement of a point P on a rigid surface which itself may be displaced with respect to some fixed system in space as a rigid body, there corresponds probably in the creep problem the resolution of the total strains ϵ into their elastic and permanent parts ϵ' and ϵ'', as will be shown later.

Finally there is another group of time effects, including the elastic after-effect, elastic hysteresis, and similar phenomena, and particularly those of the integral type studied by Boltzmann, von Kármán, Wiechert, Bennevitz, Gutenberg, and other

[15] Proc. A.S.T.M., vol. 25 (1925), part II, p. 97.

physicists, which must be mentioned briefly here because they are also beyond the scope of the present consideration and cannot be covered by the assumption that stress depends on ϵ and u alone. While elastic hysteresis cannot be treated here, present means will, however, allow us to consider various kinds of hysteresis phenomena due to the yielding of metals and the fact that plastic deformations require a certain time t in which to be produced.

2 If A designates the actual area of the cross-section of the bar and A_0 the original area, then for many purposes (large deformations) it will be sufficient to assume that the volume of the bar does not change during a tensile test:

$$Al = A_0 l_0 \quad \ldots\ldots [3]$$

If σ_0 designates the stress as referred to the original cross-sectional area A_0 and σ the "true stress" found by dividing the load P by A, then as

$$\sigma_0 = \frac{\sigma}{1+\epsilon} \quad \ldots\ldots [4]$$

we may distinguish in the case of larger extensions ϵ between a stress surface

$$\sigma = F(\epsilon, u) \quad \ldots\ldots [5]$$

referring to the "true stress" and a second surface or function of the same variables ϵ, u:

$$P = A_0\sigma_0 = A_0 \frac{F(\epsilon, u)}{1+\epsilon} \quad \ldots\ldots [6]$$

in which the load P or the stress σ_0 is plotted as ordinates.

Besides the first stress surface $\sigma = F(\epsilon, u)$ it may be necessary to consider the second one $\sigma_0 = H(\epsilon, u)$, particularly in the case of large deformations and when the question of stability of a series of subsequent states of stress has to be investigated. Both surfaces may be geometrically represented by plotting their ordinates σ respectively σ_0 above the same abscissas ϵ and u.

As speed of strain or rate of flow u, the derivative of the strain ϵ with respect to the time t was introduced above and will be used in the following. As this strain ϵ was referred to the original length l_0 of the bar and $\epsilon = (l - l_0)/l_0$, the speed of strain

$$u = \frac{d\epsilon}{dt} = \frac{1}{l_0}\frac{dl}{dt} \quad \ldots\ldots [7]$$

indicates the speed with which two points marked on the bar will move relatively to each other. For large deformations this definition for ϵ seems arbitrary and it would be more natural to define ϵ by the instantaneous increase of length divided by the actual length l at which ϵ is sought, or by

$$d\epsilon_n = \frac{dl}{l} \quad \text{or} \quad \epsilon_n = \int_0^l \frac{dl}{l} = \log_e \frac{l}{l_0} = \log_e (1+\epsilon) \quad \ldots [8]$$

The "natural speed of strain" would then be

$$u_n = \frac{d\epsilon_n}{dt} = \frac{1}{1+\epsilon}\frac{d\epsilon}{dt} = \frac{u}{1+\epsilon} \quad \ldots\ldots [9]$$

In the paper by H. Deutler, quoted above, u_n was erroneously defined as

$$u_n = u/(1+\epsilon)^2$$

3 For the following, it will be of interest to consider a few simple cases of stress (in contrast to the static case) encountered in testing methods or in practice.

(*a*) *Uniform Rate of Flow* (u = const.). We have already used one form of these conditions when defining the shape of the first stress surface by means of a set of parallel plane sections or profiles assuming in each u = const. Each curve (Fig. 1) refers thus to a tensile test, which was made under a uniform rate of stretching or of flow, so that the length of the bar or the total unit elongation ϵ was increasing uniformly with time t: $\epsilon = ut$, where u is a constant.

(*b*) *Varying Rate of Flow* (u). Instead of keeping u constant, the relative movement of both heads of the testing machine or the ends of a bar may be given by means of a time function either of the unit elongation $\epsilon = f(t)$ or of the rate of flow $u = d\epsilon/dt = g(t)$.

(*c*) *Constant Load* (P = const.). A very frequently encountered method of testing creep data in practice is to keep the load P constant during a test made at a given constant temperature. This means, then, that a test is conducted along a horizontal section P = const. of the second stress surface $H(\epsilon, u) = \sigma_0$ = const. If the ϵ and u values are determined from a test made under constant load P or a given value of σ_0 = const., which can be done by observing the time-elongation curve (see Fig. 2) under various but constant loads, $\epsilon = f(t)$, and taking derivatives

$$u = \frac{d\epsilon}{dt} = f'(t)$$

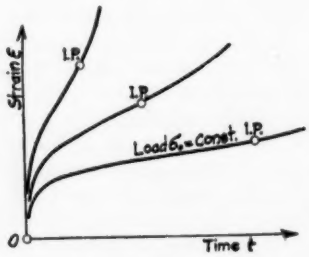


FIG. 2 TIME ELONGATION CURVES FOR CONSTANT LOADS (P OR σ_0 = CONST.)

then corresponding pairs of ϵ, u may be obtained from these last two relations and the curve $H(\epsilon, u) = \sigma_0$ = const. constructed point by point.

If this procedure is repeated subsequently for a number of values of σ_0 = const., a complete picture of the contour lines of the second stress function or the second stress surface will be obtained and thus the surface

$$\sigma_0 = H(\epsilon, u)$$

be experimentally determined, and from this finally the first stress function:

$$\sigma = (1+\epsilon)H(\epsilon, u)$$

(*d*) *Constant Stress* (σ = const.). To maintain a creep test under constant stress when the elongations ϵ are increasing requires a gradual decrease of the load P with the elongations according to

$$P = A\sigma = A_0 \frac{\sigma}{1+\epsilon}$$

or inversely as $1 + \epsilon$, which has been already suggested by some experimenters by having one end of the attachment used in stretching a thin wire plunge a weight into water.

(*e*) *Uniform Rate of Stress* (w = const.). The stress σ may increase directly as the time, $\sigma = wt$, where the rate of stress $w = d\sigma/dt$ = const. This would be realized in a test made with a wire stretched by the weight of a bucket, into which water or a stream of small shot runs uniformly, the quantity per unit of time decreasing inversely as $1 + \epsilon$.

(*f*) *Changing Rate of Stress or Load.* Instead of prescribing the law of the movement of the heads or grips of the testing machine as under (*a*) and (*b*), the law may be given in such a way that either the total load P, the true stress σ, or the rate of the true stress $w = d\sigma/dt$ may vary with the time t.

(*g*) *Relaxation* (ϵ = const.). An important case is that in which after a creep test or after the short elastic stretching or a continuous prolonged action of a load a bar is so constrained that it will not change in length. It will be shown later that for this kind of deformation to occur in a finite time an elastic member is essential and must be included in the system under consideration. Under *pure relaxation* there will be understood a gradual transformation of the elastic part ϵ' of the total unit elongation $\epsilon = \epsilon' + \epsilon''$ into a newly formed permanent elongation ϵ'', while the total unit elongation $\epsilon = \epsilon' + \epsilon'' = $ const. remains unchanged, so that ϵ'' increases at the expense of ϵ'.

Mixed and Compound Relaxation. It is difficult or practically impossible to realize pure relaxation in a test with a solid bar of one of the rigid materials, used in engineering practice and having a high modulus of elasticity. The heads of the test piece have to be attached by grips, screw threads, etc. to other parts, which will themselves also contribute to the elasticity of the system. Relaxation must then occur at the expense of the combined elasticities of the bar and the attachments. In practice mixed relaxation occurs only, for example, in tension members such as cylindrical bolts with screwed ends exposed to high temperature, etc. The piece in question may at the beginning of the process be represented by a scheme such as that shown in the left half of Fig. 3. The spring expresses the action of the elasticity of the adjoining machine parts, which are held under stress by means of the bar shown in the upper part of the figure. The weaker the elastic spring is with which the bar is constrained to stretch, the longer will be the extension in which it will relax, when creeping, until the stress disappears from it. The unloaded end position of the relaxed system is indicated at the right in Fig. 3.

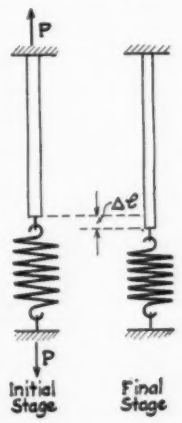


FIG. 3 RELAXATION

If the elastic-spring action is again due to the elasticity of the piece in question, but the stresses therein now vary from point to point (examples: a circular disk shrunk on a cylindrical pin, press fits), we may speak of compound relaxation.

4 *Time and Load Diagrams.* A test conducted in a predetermined manner is represented, as we have seen, by a *condition curve* on the stress surface, indicating a continuous series of modes of stress and strain. The shape of such condition curves may be studied as soon as the stress surface is known or has been determined empirically, and from them the creep-time laws may be derived for the particular case under consideration.

Conversely, time diagrams, such, for example, as those for the

the elongation..... $\epsilon = f(t)$
the true stress..... $\sigma = g(t)$
the rate of flow..... $u = d\epsilon/dt = f'(t)$, or
the rate of stress..... $w = d\sigma/dt = g'(t)$

which might be obtained from experimental observation, will help to determine the various sections u = const., σ = const., σ_0 = const., or other condition curves of the stress surface. It may be seen also that in one or the other of these cases it may be preferable to pursue a test or condition curve on the second of the two surfaces of stress, Eq. [6]. For example, a test made under a load P = const. is characterized on the second surface simply by a contour line, while the corresponding curves σ_0 = const. drawn on the first surface would be of a less simple nature.

This may be of importance if more complex cases in which the stress components vary from point to point in the body have to be studied, such as would be the case when bending, torsion, or other non-homogeneous states of stress have to be investigated. In these latter cases, the quantities determining the external loads, the maximum bending or twisting moment, the pressure in case of a tube subjected to internal pressure, can only be found by integration following the ordinates σ along some cross-section of the first stress surface. To study then the cases of non-homogeneous stress when an externally applied load is kept constant leads automatically to the construction of a particular quantity by an integration expressing a total load P, a moment or torque M, or a pressure p as a function in general of ϵ and of u. The curve P = const., M = const., or p = const. on a surface, whose respective ordinates are P, M, p, etc. plotted above ϵ and u, will then determine the corresponding tests made under constant P, or M, or p, etc.

5 *The Infinitely Slow Stress-Strain Diagram.* The experimental facts concerning the changing plastic behavior of a material with temperature may be expressed by saying that while the surface $\sigma = F(\epsilon, u)$ has a comparatively flat appearance at low temperatures in its sections ϵ = const., this is no longer the case when the temperatures are raised above the limits at which the softening of the material starts. Further, with decreasing rates of flow at normal temperatures, it seems as if the stress would approach a limiting curve, the ordinates of which have a considerable magnitude. To have such a limiting curve for $\sigma = F(\epsilon, 0)$ defined or a stress-strain diagram for a rate of flow equal to $u = 0$ introduced must be explained, because a tensile test cannot be made at an infinitely low speed. What is meant here is that if rates of flow $u = d\epsilon/dt$ are chosen corresponding to the usual duration of a slow tensile test (measured perhaps by hours or days and in every case by small, but finite, values of $u = d\epsilon/dt$), the stresses will then be of an order of several thousand pounds per square inch. But it does not conversely follow, however, that if u were to be further decreased to a much smaller order of magnitude the stress would still remain of the same order of magnitude as before. Rather it seems probable, as has already been stated above, that this may not be the case and that the application of low stresses for a sufficiently long period of time even at normal temperature may cause slow yielding. One reason for believing this is indicated by the behavior of solid matter at sufficiently high temperatures. At these not only are the slopes $\partial\sigma/\partial u$ of the stress function $\sigma = F(\epsilon, u)$ much greater than at normal temperatures, but tests have clearly shown that if such a thing as an infinitely slow stress-strain curve were to exist at these temperatures, its stress ordinates would be considerably smaller than at normal temperatures. Expressed in other words it would thus seem that polycrystalline or amorphous solid materials, at sufficiently elevated temperatures and sufficiently small values of u, more and more approach conditions which, at least within a limited range of ϵ and u, may be expressed by

$$\sigma = Cu = \left(\frac{\partial\sigma}{\partial u}\right)_{u=0} u$$

This is the property which has been called pure viscosity in the case of a slowly moving liquid. Whether a constant has to be added on the right side of this equation or not is perhaps disputable, but in any case it would be of a small magnitude at sufficiently high temperatures when compared with the values of yield stresses at low temperatures and small rates of flow.

As both slopes of the stress surface $\partial\sigma/\partial u$, $\partial\sigma/\partial\epsilon$ will figure more frequently in what follows, referring to the tensile test, it will perhaps be useful to denote both slopes of the stress surface by short names. Corresponding to the meaning attributed to what is called in hydrodynamics the coefficient of viscosity, we may call the slope of the stress function

$$\frac{\partial \sigma}{\partial u} = \phi$$

the "coefficient of viscosity" of the material at the state ϵ, u, and perhaps in an analogous way the second slope

$$\frac{\partial \sigma}{\partial \epsilon} = \psi$$

the "coefficient of work hardening for tension." Both coefficients are thus defined for pure tension only; if this state of stress were to be changed in another, such as compression or pure shear, other values of ϕ and ψ would have to be used. In the hydrodynamics of viscous fluids the coefficient of viscosity η is defined by assuming that a shearing stress τ is produced by a rate of shear v in the planes in which it acts equal to $\tau = \eta v$. It can easily be shown that for non-compressible materials, the coefficient of viscosity for pure tension ϕ is three times that for pure shear, or

$$\phi = 3\eta$$

Now, recalling again what was said about the logarithmic law, Eq. [1], which according to observations would seem to express the relation existing between speed of strain or rate of flow u and stress σ for a limited range of the unit elongation ϵ, we may finally assume that the logarithmic relation may hold excepting in the case of extremely slow rates of flow, at least at lower temperatures. For extremely small values of u, however, the logarithmic curve may gradually change into a straight line, meeting the ϵ-axis at stresses σ which may be close to or be taken as equal to zero (law of true viscous flow). For small values of u, L. Prandtl[16] arrived at a similar conclusion when attempting to express the phenomenon of elastic hysteresis and other related effects by means of a special mechanical model of a solid, which at higher rates of flow also led him to the logarithmic law.[16]

6 *Speed Diagrams.* With regard to the use of curves which express the rate of flow $u = g(\epsilon)$ as a function of the unit elongation ϵ and which are the projections of the condition curves σ = const. or similar curves on a P, M, or p surface on the ϵ, u plane, a few remarks may be in order. If a point P is moving along a straight line, its position being given by $x = f(t)$, one may instead of plotting a second time diagram for its velocity $u = \dot{x} = df/dt$, eliminate the variable time parameter from both equations, thus drawing, for example, a *diagram of the speed* in the form:

$$u = g(x)$$

For a given law of motion $x = f(t)$ there will be a corresponding speed diagram obtained by eliminating the variable parameter t from both equations. In such speed diagrams certain rules must be observed. If $u > 0$, x must increase, and if $u < 0$, x must decrease. The speed diagrams possible to pass through a given point P of the plane with a positive ordinate u must therefore run with their directions as indicated by the arrows in Fig. 4, and if $u < 0$, as indicated at the point Q below. The speed diagram of a continuous motion in which u changes its sign must necessarily cross the x-axis at right angles and in the way indicated in Fig. 5. [An exceptional case may be that indicated by Fig. 5 (right) in which the path becomes tangent to the x-axis; in this case both $u = 0$ and $du/dt = 0$, that is, also the acceleration vanishes.] A path can never cross the axis in the manner indicated in Fig. 6. If a path crosses or intersects the x-axis at an obtuse angle α, Fig. 7, the moving point must come to rest at the intersection point A, but movements are only possible in the directions of the arrows. If the angle α is an acute one, the movements start from rest (B, Fig. 7, following the arrows). Simple harmonic motion has an ellipse for its speed diagram, and uniformly accelerated motion a parabolic speed diagram. A straight path in the speed diagram corresponds to a motion of the type

$$x = C_1 e^{ct}$$

These rules must apply also to the "speed diagrams of plastic tests" $u = d\epsilon/dt = g(\epsilon)$ in the ϵ, u-plane. For example, such "speed diagrams" would be the projections of the important curves P = const. of the second stress surface $\sigma_0 = H(\epsilon, u)$ on the ϵ, u-plane, which would indicate how the speed during a long-time or creep tensile test of a bar will change with its elongation under a constant load P, or when, after some preliminary stretching of a bar according to some rule, the load is held constant for a while or for the remaining time (plastic after-flow). None of these curves is permitted to cross the ϵ-axis at an angle. If such a thing does occur, that is, if, for example, stretching of a bar in one direction is reversed into compression, or vice versa, then the corresponding speed diagram in the ϵ, u-plane must cross the ϵ-axis at right angles.

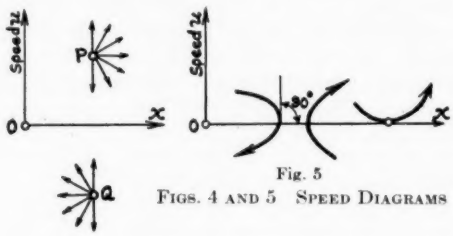


Fig. 4

Fig. 5

FIGS. 4 AND 5 SPEED DIAGRAMS

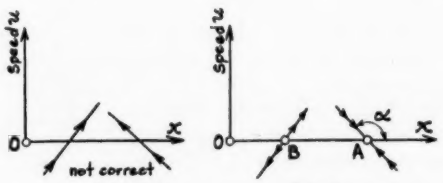


Fig. 6 Fig. 7

FIGS. 6 AND 7 SPEED DIAGRAMS

IV—VISCOUS MATERIAL IN TENSION[17]

(*a*) The simplest case of plastic flow depending on speed of deformation is that in which the stress σ produced by stretching a bar is proportional to the rate of flow

$$\sigma = \phi \frac{\partial \epsilon}{\partial t} = \phi u \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

the constant ϕ is the coefficient of viscosity for tension and $\phi = 3\eta$, where η is the coefficient of viscosity for shear. The equation of the stress surface is the one just given, from which is seen that in this case

$$\sigma = F(\epsilon, u) = \phi u \ldots\ldots\ldots\ldots\ldots\ldots [11]$$

σ does not contain ϵ and is a plane passing through the ϵ-axis. We immediately find the load function according to Eq. [4]

[16] *Zeit. f. ang. Math. u. Mech.*, vol. 8 (1928), p. 85.

[17] In this connection the various attempts by H. Hencky [*Z.V.D.I.*, vol. 69 (1925), p. 695, and in other papers] in which he combined in general terms viscosity, plasticity, and elasticity, should be mentioned here. As Dr. Hencky has kindly brought to the attention of the author, interesting applications of the theory of viscous flow to the movements in solids (glacier ice) are due to Lagally, of Dresden.

$$\sigma_0 = \frac{\sigma}{1+\epsilon} = \frac{\phi u}{1+\epsilon} \quad \ldots\ldots\ldots\ldots\ldots [12]$$

which is in this case a hyperbolic paraboloid (Fig. 8).

This last equation also furnishes the law according to which the unit extension ϵ will change with time t under a constant load $P = A_0\sigma_0 = \text{const}$. Writing for $u = d\epsilon/\partial t$ and integrating (assuming $\sigma_0 = \text{const.}$),

$$\frac{d\epsilon}{1+\epsilon} = \frac{\sigma_0}{\phi}\, dt \quad \ldots\ldots\ldots\ldots\ldots [13]$$

ϵ is found from

$$1 + \epsilon = \frac{\sigma}{\sigma_0} = e^{\sigma_0 t/\phi} \quad \ldots\ldots\ldots\ldots\ldots [14]$$

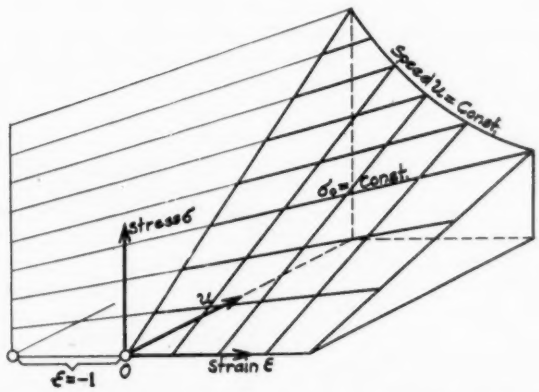


FIG. 8 SHOWING THE LOAD SURFACE $\sigma_0 = H(\epsilon, u)$ FOR VISCOUS FLOW TO BE HYPERBOLIC PARABOLOID

which is nothing but another expression of the simple rule mentioned above, namely, that if the "speed diagram" $u = f(\epsilon)$ is a straight line, the motion follows the exponential time law. Under a constant load the length of a bar of a viscous material will increase with the time according to an exponential function, and such a bar will always be in an unstable condition under any constant load, no matter how small. The rate of flow u will be proportional to the weight attached to the bar. This behavior is obvious on account of the continuously decreasing cross-section A of the bar, which will also change exponentially with the time:

$$A = A_0 e^{-\sigma_0 t/\phi} \quad \ldots\ldots\ldots\ldots\ldots [15]$$

Some of the characteristic curves for this case are as follows: the elongation-time curves are exponential functions; the stress-strain curves for uniform rate of flow are straight horizontal lines; the speed diagrams for constant load are a set of straight lines radiating from a common point (A, Fig. 10) with the abscissa $\epsilon = -1$; the load-elongation diagrams for constant rates of flow are equilateral hyperbolas having for their vertical asymptote the line $\epsilon = -1$. There are two sets of straight lines on the hyperbolic paraboloid, namely, the lines $\epsilon = \text{const.}$, having the equation

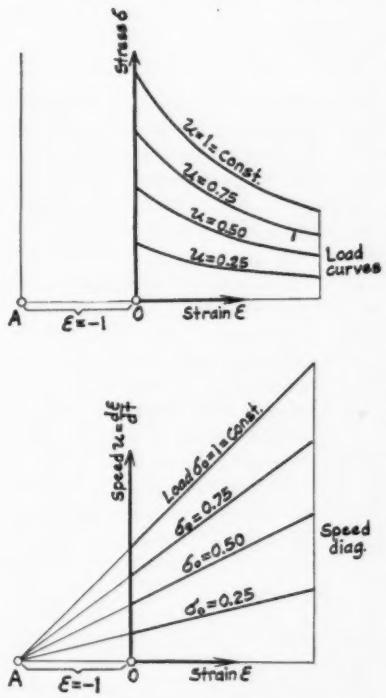
$$\sigma_0 = \left[\frac{\phi}{1+\epsilon}\right] u \quad \ldots\ldots\ldots\ldots\ldots [16]$$

where the term in the bracket is a constant, and the contour lines of the surface corresponding to constant loads or $\sigma_0 = \text{const.}$, namely,

$$u = \frac{\sigma_0}{\phi}(1+\epsilon) \quad \ldots\ldots\ldots\ldots\ldots [17]$$

Such a load surface, Eq. [3], is indicated in Figs. 8, 9, and 10.

(*b*) *Process of Necking of Bar.* Another consequence of the assumed laws for plastic flow of this kind would be that the slightest irregularity in the magnitudes of the cross-sections of a bar stressed under a constant load would finally cause the necking always to start at the smallest cross-section of the bar. It can readily be seen that if one of the cross-sections of the bar is slightly smaller than the others, the stresses will be correspondingly larger there. But as larger stresses must produce more rapid deformations, the smaller cross-sections will start to contract more rapidly than their larger neighbors, and thus necking will be localized more and more and accelerated at the place where it started. The necking process can easily be followed if the law of the variation of the original areas of cross-section is known and if the cross-sections do not change too rapidly along the bar.



FIGS. 9 AND 10 THE LOAD SURFACE $\sigma_0 = H(\epsilon, u)$ FOR VISCOUS FLOW REPRESENTED BY TWO PROJECTIONS

Let A be the variable cross-sectional area of the bar at the distance x and at the time t, and A^* be the original cross-sectional area at the same place and at the time $t = 0$ (Fig. 11). If the load P remains constant, then at any given distance x

$$P = A^*\sigma^* = A\sigma = \text{const.} \quad \ldots [18]$$

As each element dx of the bar will behave as one having a constant cross-sectional area, the areas A at a given distance x will change with time t according to Eq. [15], or

$$A = A^* e^{-\sigma^* t/\phi} \quad \ldots [19]$$

where σ^* designates the stress acting at the time $t = 0$. Replacing σ^* by P/A^*,

$$A = A^* e^{-Pt/\phi A^*} \quad \ldots\ldots\ldots\ldots [20]$$

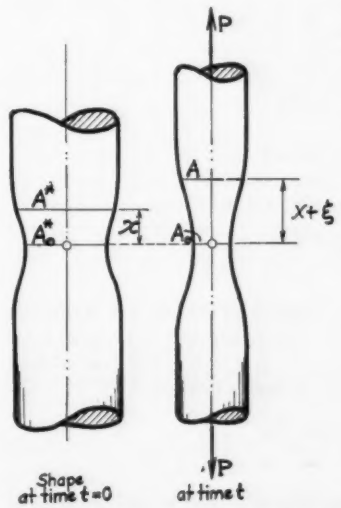


FIG. 11 NECKING OF BAR OF VISCOUS MATERIAL

which equation will determine how the cross-sectional areas A must change both with the distance x and the time t. In this last equation, the original cross-sectional areas A^* of the bar

must be given; A^* is a known function of x. In Appendix No. 1 it is briefly indicated how, for a given profile $A^* = f(x)$, the subsequent profiles of the bar at later times may be easily computed and thus the process of necking follows. Fig. 12 shows two stages of the necking of a bar in true scale after two equal time intervals t_0, assuming that the original profile of the bar was given in an interval $-a < x < a$ by an expression of the form:

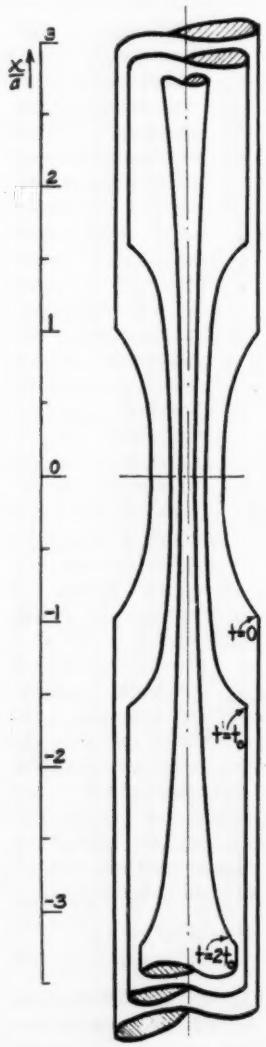


FIG. 12 THREE SHAPES TAKEN BY A BAR OF PURELY VISCOUS MATERIAL WHEN STRETCHED UNDER A CONSTANT LOAD

$$A = \frac{A_0^*}{1 - \beta^2 x^2/a^2} \quad \ldots [21]$$

where β, a, and A_0^* are constants and A_0^* is the minimum original cross-section of the neck. The widths shown refer to the *areas* (not to the diameters). After the minimum section is reduced to half of its former value, the bar stretches much more rapidly during the second time interval, producing a similar percentage decrease in section due to the increasing stresses in the necked portion.

(*c*) *Tensile Test With Constant or Varying Rate of Stress.* If a tensile test is conducted so that the stress σ uniformly increases with time t,

$$\sigma = wt \quad \ldots [22]$$

$$\frac{\partial \sigma}{\partial t} = w = \text{const} \quad \ldots [23]$$

The elongation-time curve is found from the tensile-stress surface

$$\sigma = F(\epsilon, u) = \phi u = \phi \frac{\partial \epsilon}{\partial t} \quad \ldots [24]$$

By equating Eq. [11] and Eq. [14]

$$\phi \frac{\partial \epsilon}{\partial t} = wt \quad \ldots [25]$$

from which

$$\epsilon = \frac{w}{2\phi} t^2 \quad \ldots [26]$$

assuming that for $t = 0$ the unit elongation ϵ is equal to zero: $t = 0$, $\epsilon = 0$. The rate of flow in such a test, or

$$u = \frac{\partial \epsilon}{\partial t} = \frac{w}{\phi} t \quad \ldots [27]$$

increases uniformly with time, and it is seen at once that for a viscous material a test in which the rate of stress w is constant produces uniformly increasing rates of flow u. Elimination of t from the expressions for σ and ϵ furnishes the equation of the stress-strain curves for constant rate of stress:

$$\sigma = \sqrt{2w\phi}\ \sqrt{\epsilon} \quad \ldots [28]$$

while elimination of t from the expressions for u and ϵ leads to the equation of the speed diagrams:

$$u = \sqrt{\frac{2w}{\phi}}\ \sqrt{\epsilon} \quad \ldots [29]$$

both of which curves are obviously ordinary parabolas (Fig. 13). If the rate of stress $\partial\sigma/\partial t$ is not constant, but σ is given as a function of time t,

$$\sigma = f(t) \quad \ldots [30]$$

then from

$$\sigma = \phi \frac{\partial \epsilon}{\partial t} = f(t), \qquad \epsilon = \frac{1}{\phi} \int_0^t \sigma dt \quad \ldots [31]$$

We may use these expressions also in the case of unloading. If, for example, a tensile test has been maintained under uniform rate of flow $u = u_1 = \text{const.}$ which produced the stress $\sigma_1 = \phi u_1$, and then the bar is unloaded under uniformly decreasing stress

$$\sigma = \sigma_1 \left(1 - \frac{t}{t_1}\right) \quad \ldots [32]$$

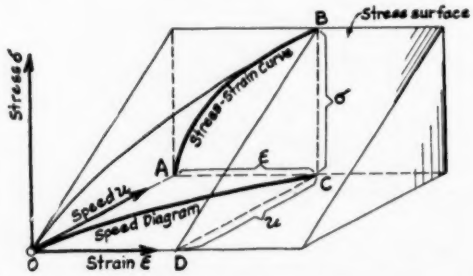


FIG. 13 STRESS-STRAIN CURVE AND SPEED DIAGRAM FOR A TENSILE TEST MADE ON VISCOUS MATERIAL AT A CONSTANT RATE OF STRESS

with the negative rate of stress $w = -\sigma_1/t_1$ then the unit elongations produced must be equal to

$$\epsilon = \epsilon_1 + \frac{\sigma_1(2t_1 - t)t}{2\phi t_1} \quad \ldots [33]$$

where ϵ_1 designates the value of ϵ for $t = 0$, at the instant from which unloading started.

From the last equation it follows that for $t = t_1$, when $\sigma = \sigma_1$, the *amount of after-flow*

$$\epsilon - \epsilon_1 = \frac{\sigma_1 t_1}{2\phi} = \frac{u_1 t_1}{2} \quad \ldots [34]$$

this is only *half* as much as the amount by which the bar would have yielded or crept during a time equal to the unloading time t_1 had the stress σ been kept constant and equal to σ_1.

If the bar is loaded at a slow but constant rate of stress and subsequently unloaded at the same rate, the resulting two branches of the stress-strain curve are congruent parabolas, thus enclosing half or—if the stresses are subsequently also reversed—a complete cycle of a hysteresis loop in the σ,ϵ-plane. The higher the speed w, the narrower become these loops. If the stress $\sigma = \sigma_0 \sin \omega t$ changes according to the harmonic motion, the corresponding hysteresis loops will be ellipses. Hitherto the elasticity in these loops of the material was not considered, but it can also be easily included, as will be indicated later.

V—CREEP IN PURE TORSION OR BENDING

In Appendix No. 2 a more detailed analysis of one of these cases, namely, the flow of a cylindrical bar subjected to torsion,

is given, assuming certain simplifications concerning the nature of the phenomena but an arbitrary law of plastic deformation, and including also the effect of speed upon these deformations. The case of torsion of a bar with a circular cross-section has been chosen as an example, but the basic considerations are essentially the same also for the case of pure bending of bars having simple cross-sectional shapes and a non-variable moment of inertia. The division of the total strain into an elastic and a permanent portion can also be considered, if necessary, but for the present this has been neglected. Other time effects, as the elastic after-effect or hysteresis, have, however to be treated by other assumptions.

Under the conditions stated in Appendix No. 2, a moment-torsion diagram may be constructed, if the stress surface

$$\tau = F(\gamma, v) \quad \ldots\ldots [35]$$

for pure shear is known and a condition curve is given which indicates how one of the variables τ, γ, or v shall change with time t. τ is the shearing stress in the bar, γ the unit shear, and $v = d\gamma/dt$ the rate of shear at a point distant r from the axis of the bar. A test conducted under certain conditions is represented by a certain section of the stress surface or a curve thereon. If, for example, a torsion test with a round bar is made by twisting it permanently at a uniform angular speed, then the section is a plane passing through the origin O (see Fig. 27) and perpendicular to the γ, v-plane, and this plane has to be rotated in a given manner. The values of τ will be given by the ordinates τ along this plane section, which will act along a radius of the circular cross-section of the bar.

VI—Resolving the Total Strain and Rate of Strain Into Their Elastic and Permanent Parts

At larger permanent deformations produced by plastic flow the part of the total strain ϵ' which is elastic and will disappear as soon as the bodies are unloaded, is small compared to the total strain. In many cases it will therefore not be necessary to consider this part at all. If the permanent strains ϵ'', however, are comparable with the elastic strains ϵ', there is no justification in neglecting the latter. Also when a body has been deformed to a greater extent, it will in certain cases be necessary to take into consideration the elastic parts of the total strains. This is necessary, for example, if the system of residual stresses, which in general remains in a body unloaded from a non-homogeneous plastic state of stress, is to be determined. Another important case is that of the initial creep of a metallic piece at a comparatively low temperature, where creep has just slowly started, but has not yet had time enough to develop farther. This is perhaps the case in most of the design problems which depend on creep rates. On closer examination, it may become difficult to state more exactly which part of the strain at a given load is the elastic and which the plastic part, because of the well-known fact that for many materials there does not exist a distinct purely elastic state of stress and for them loading and unloading curves—if plastic yielding is excluded—may differ to a considerable extent (elastic hysteresis).

Excluding again similar cases of elastic after-effects, one may assume as a first approximation that the total strain ϵ is the sum of an elastic portion ϵ' and a permanent portion ϵ''. With regard to the elastic portion ϵ' one may assume that this depends only on the *actual* values of the stress σ (and not, for example, on the previous history of stressing, etc.), using the linear relation $\epsilon' = \sigma/E$, where E is the modulus of elasticity

$$\epsilon = \epsilon' + \epsilon'' = \frac{\sigma}{E} + \epsilon'' \quad \ldots\ldots [36]$$

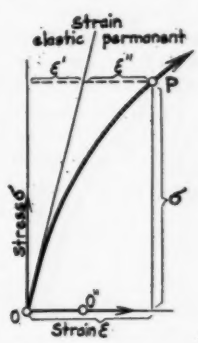
Differentiating,

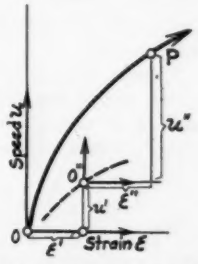
$$\frac{\partial \epsilon}{\partial t} = \frac{\partial \epsilon'}{\partial t} + \frac{\partial \epsilon''}{\partial t} = \frac{1}{E}\frac{\partial \sigma}{\partial t} + \frac{\partial \epsilon''}{\partial t} \quad \ldots\ldots [37]$$

Using again the notations $u = \partial\epsilon/\partial t$ and $w = \partial\sigma/\partial t$ for rates of strain and of stress, this last equation may also be written in the form:

$$u = u' + u'' = \frac{w}{E} + u'' \quad \ldots\ldots [38]$$

Assuming now certain laws, regarding the dependence of stress on the rate of flow u'' or on previous strain ϵ'' (work hardening), and that a condition curve in some form is known, we see that while such a condition curve in the ϵ,u,σ-space is described, the origin O from which the total strains ϵ and total rates of flow $u = \partial\epsilon/\partial t$ were measured remains at rest, and that a second origin O'' (Figs. 14 and 15) in the ϵ,u-plane must exist, the rectangular coordinates of which are the elastic strain ϵ' and the elastic rate of strain u' (Fig. 15). This second origin O'' will start in general to move in the ϵ,u-plane as soon as the point P moves along a given condition curve OP. To find the permanent strains ϵ'' or the permanent rates of flow $u'' = \partial\epsilon''/\partial t$, we shall have to define them as the *relative coordinates* of the moving point P with respect to the moving axes ϵ'' and u'' in this system ϵ'',u'' with the origin O''.





FIGS. 14 AND 15 ELASTIC AND PERMANENT PARTS OF STRAIN ϵ AND SPEED OF STRAIN u

This being established, we may now essay to follow in a few simple cases these relative movements in detail. Before doing so, however, it should be recalled that all this refers only to the parallel translation of axes, that is, assuming *no rotation of the stress vector σ relative to the stressed body.* If the stress vector σ rotated relatively to the body under plastic flow, conditions may become different. This interesting and more difficult case may be treated on a later occasion.

The shape of the curve described by the second origin O'' (from which the permanent strains and rates of strain have to be measured) will depend on how the stress σ varies or how the rate of stress $w = \partial\sigma/\partial t$ changes with the time. If this is known,

$$\sigma = f(t), \qquad w = \frac{\partial \sigma}{\partial t} = \frac{df}{dt} \quad \ldots\ldots [39]$$

the coordinates ϵ' and u' of the origin O'' are given by

$$\epsilon' = \frac{\sigma}{E}, \qquad u' = \frac{1}{E}\frac{\partial \sigma}{\partial t} = \frac{w}{E} \quad \ldots\ldots [40]$$

In general, however, σ will not be directly known as a function of the time t. The coordinate transformation:

$$\left.\begin{aligned} \epsilon &= \epsilon' + \epsilon'' = \frac{\sigma}{E} + \epsilon'' \\ u &= u' + u'' = \frac{w}{E} + u'' \end{aligned}\right\} \quad \ldots\ldots [41]$$

by which the total strains are connected to the permanent strains, indicates that the coordinates u may at certain points undergo discontinuous changes. This is important to note, because in the ordinary testing of materials it is a very frequent thing to change suddenly the speeds of deformation. If one associates the permanent part of the rate of strain with the stress, it will always require a certain time before a stress will adjust itself to a new rate of strain. What continuously changes is, in most cases, the permanent rate of strain u''. If the rate of stress w suddenly changes from w_1 to w_2, that is, if the time curve for the stress $\sigma = f(t)$ has a break of the tangent, the total rate of strain u will jump by an amount

$$u_2 - u_1 = \frac{w_2 - w_1}{E} \quad [42]$$

while the permanent rates u'' which are associated with the stress will still change continuously. We see, therefore, that in the speed diagrams where the total rates of flow u are shown depending on the total strain ϵ, discontinuities are permissible, and *it will be even necessary to introduce discontinuities* of u each time when the stress rate jumps or changes abruptly.

From the foregoing is seen that, for example, when the stress σ = const. ("after-flow or creep" under constant stress), the point O'' will move along the ϵ-axis, as u' remains zero. If, however, the rate of flow is kept uniform, u = const., and the bar is stretched with a uniform speed, then along the path, $\partial u/\partial t = 0$ and $w = \psi u$, hence

$$\epsilon' = \frac{\sigma}{E}, \qquad u' = \psi \frac{u}{E} \quad [43]$$

the rate of stress w is proportional to the rate of flow u, the proportionality factor being the coefficient of strain hardening $\psi = \partial\sigma/\partial\epsilon$.

Viscous Body Having Elasticity. This is characterized by an elastic part of the strain $\epsilon' = \sigma/E$ and by assuming that the rate of flow $u'' = \partial\epsilon''/\partial t$ of the plastic part is proportional to the stress $\sigma = \phi u''$. We have thus first

$$\epsilon = \frac{\sigma}{E} + \epsilon'' \quad [44]$$

and by differentiating this with respect to the time, we obtain

$$\frac{\partial\epsilon}{\partial t} = \frac{1}{E}\frac{\partial\sigma}{\partial t} + \frac{\partial\epsilon''}{\partial t} \quad [45]$$

where $\partial\epsilon''/\partial t = \sigma/\phi$. Hence

$$\frac{\partial\epsilon}{\partial t} = \frac{1}{E}\frac{\partial\sigma}{\partial t} + \frac{\sigma}{\phi} \quad [46]$$

Exactly this equation has been suggested by J. C. Maxwell describing the phenomena of relaxation.[18] Maxwell's law of deformation refers thus to a viscous body with perfect elasticity.[19] It is perhaps of interest to interpret this law of deformation, using the stress surface. Here this may be considered defined not for the *total* strains ϵ but for the permanent strains ϵ'' by the fundamental equation expressing the law of viscous flow:

$$\sigma = \phi u'' = \phi \frac{\partial\epsilon''}{\partial t} = F_1(\epsilon'', u'') \quad [47]$$

The stress surface for the permanent strains is thus given as an inclined plane containing the ϵ-axis. This equation of the stress surface has to be combined with the other equation

$$\epsilon = \frac{\sigma}{E} + \epsilon'' \quad [48]$$

A few examples may give simple applications.

1 *Constant Rate of Flow.* A test starting from rest with a constant rate of flow $u = u_1$ = const. is given by

$$\epsilon = u_1 t = \frac{\sigma}{E} + \epsilon'' \quad [49]$$

Replacing σ by

$$\sigma = \phi \frac{d\epsilon''}{dt} \quad [50]$$

we obtain the following differential equation for the permanent strain ϵ'':

$$\frac{\phi}{E}\frac{d\epsilon''}{dt} + \epsilon'' = u_1 t \quad [51]$$

from which

$$\epsilon'' = u_1 t - \frac{\phi}{E} u_1 + ce^{-Et/\phi} \quad [52]$$

The sum of the second and third terms of [52] is just equal to $-\epsilon'$. Assuming that for $t = \infty$ the elastic part of the strain $\epsilon' = \epsilon_1 = \phi u_1/E$ when $\sigma = \sigma_1$, we obtain finally

for the elastic strain $\epsilon' = \epsilon_1(1 - e^{-Et/\phi})$ [53]

for the stress $\sigma = \sigma_1(1 - e^{-Et/\phi})$ [54]

for the permanent strain $\epsilon'' = \epsilon - \epsilon'$ [55]

Replacing t by ϵ_1/u_1 and $\phi u_1/E$ by ϵ_1 from [55],

$$\sigma = \sigma_1(1 - e^{-\epsilon/\epsilon_1}) \quad [56]$$

Obviously this is *the equation of a "stress-strain diagram taken under a uniform rate of stress"* $u = u_1 = const.$ The stress curves for various speeds $u_1 = E\epsilon_1/\phi$ are exponential curves, tangent to an inclined line OA (Fig. 16) through the origin O, having the slope $(d\sigma/d\epsilon)_0 = E$, and approach rapidly the horizontal asymptotes $\sigma = \sigma_1$. The orbit of the second origin O'' having the coordinates ϵ' and u' in the ϵ,u-plane (Fig. 17), where

$$\left.\begin{aligned} \epsilon' &= \epsilon_1(1 - e^{-Et/\phi}) \\ u' &= \frac{\epsilon_1 E}{\phi} e^{-Et/\phi} \end{aligned}\right\} \quad [57]$$

is here the straight line $\overline{BC}$ with the equation

$$\epsilon' = \epsilon_1 - \frac{\phi}{E} u' \quad [58]$$

[18] H. Jeffreys distinguishes between two kinds of imperfections in the elastic behavior of matter. One is expressed by $\tau = G(\gamma + t_0\dot{\gamma})$, and the other by $\dot{\gamma} = \dot{\tau}/G + \tau/\eta$. The first is the classical expression of an elastic restoring force and a damping force as introduced in the ordinary theory of damped vibrations; the second expresses Maxwell's law of deformation. Jeffreys calls the first modes of imperfections "firmoviscosity," and the second ones "elasticoviscosity." (See "Earth . . .," p. 263.) He also calls attention to the fact that the theory of the first kind of imperfections for small deformations consists in simply replacing in the equation of "perfect elasticity," $\tau = G\gamma$, the elastic constant (the modulus of rigidity) G by a functional operator, namely, $G(1 + t_0\, d/dt)$, and that similarly in the case of elasticoviscous imperfections the constant G has to be replaced by the operator

$$G\left[1 + \frac{1}{t_1 d/dt}\right]$$

[19] Compare also the instructive detailed review of the phenomena of the elastic after-effect in the excellent articles by H. Fromm in "Handbuch der physikalischen und technischen Mechanik," vol. iv, 1, part 2, p. 533, Leipzig, 1931.

2 *Sudden Changes of Rate of Flow* must be accompanied by corresponding changes of the stress, which latter, however, will require a certain time for their development. The curve in the stress-strain diagram will be shifted up or down if the rate of flow is increased or decreased suddenly as indicated in Fig. 18. The jumps are exponential curves with a slope given by the elastic modulus of the material $d\sigma/d\epsilon = E$ at the point where the speed of stretching changes discontinuously.

3 *Rate of Stress Given. Hysteresis Due to Yielding.* Assuming any given time law, according to which the stress is changed, namely,

$$\sigma = f(t) \quad \ldots\ldots [59]$$

we may easily determine the corresponding curves

$$\sigma = F(\epsilon) \quad \ldots\ldots [60]$$

For example, loading under constant rates of stress w = const., including now also the elastic deformation, will again yield parabolas as stress-strain curves (Fig. 19). The difference between this and the similar case first treated (see Sec. IV-*c*) is now that all loading curves and parabolas will be tangent to a straight line having the slope $d\sigma/d\epsilon = E$ given by the elastic modulus E of the material in agreement with experiments (Fig. 19). An arc of a loading curve, assuming $\sigma = wt$ uniformly increasing with time or for a constant rate of stress w, will be given by the equation

$$\epsilon = \frac{\sigma}{E} + \frac{\sigma^2}{2\phi w} \quad \ldots\ldots [61]$$

the speed of stretching by

$$u = \frac{w}{E} + \frac{\sigma}{\phi} \quad \ldots\ldots [62]$$

and the speed-strain relation by

$$\epsilon = \frac{\phi}{E}\left(u - \frac{w}{E}\right) + \frac{\phi}{2w}\left(u - \frac{w}{E}\right)^2 \quad \ldots\ldots [63]$$

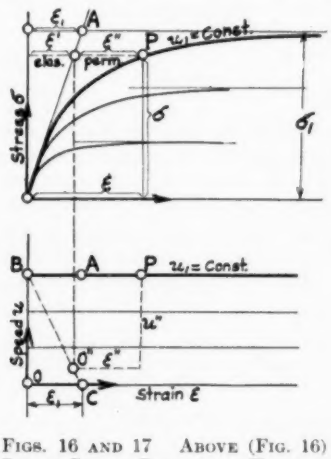
An arc of an unloading curve under the same circumstances is given by:

$$\epsilon = \epsilon_1 - \frac{\sigma_1 - \sigma}{E} + \frac{\sigma_1^2 - \sigma^2}{2\phi w} \quad \ldots\ldots [64]$$

$$u = -\frac{w}{E} + \frac{\sigma}{\phi} = u_1 - \frac{\sigma_1 - \sigma}{\phi} \quad \ldots\ldots [65]$$

where ϵ_1, u_1, σ_1 refer to the initial state and $\sigma = \sigma_1 - wt$. That yielding or the production of a permanent strain or deformation requires a finite time for its development while an elastic strain may instantaneously be changed, will now find its adequate expression: All loading or unloading curves $\sigma = F(\epsilon)$ will have their tangents at the points where they pass through the ϵ-axis inclined under the slope given by $d\sigma/d\epsilon = E$. For constant rates of stress (w = const.) hysteresis loops will be obtained, such as those indicated in the Figs. 20–23. Their arcs are all parabolas, the axes of the loops in the σ, ϵ-plane being now inclined as required under a slope = E. The quicker a test is made, the slenderer the loop will become. The figures show three loops corresponding to speeds changing in the ratio 1:2:4. From the point of view of a clear analysis of these loading and unloading curves, it must be noted that the "corresponding speed diagrams" for these hysteresis loops will have discontinuities. After the loading under constant rate of stress w is reversed into unloading with the rate $-w$, the rate of stress jumps at the point of inversion. Consequently the total rate of flow must also jump discontinuously and by a corresponding amount, equal to $2u_0 = 2w/E$. This is indicated by the shifting of one-half of the loop with respect to the other half in the lower part of Figs. 20, 21, which indicates the shape of the speed diagram for the first example, or a projection of the hysteresis loop on the ϵ, u-plane. In this case the origins O'' move first on a parallel line $u = u_0$ = const. in Fig. 21 for one sense of the movement, and return on a second parallel line $u = -u_0$ for the reversed sense. A check for the preceding statements is that these two arcs in the plane of Fig. 21 become exactly perpendicular to the ϵ-axis at the two points where they cross this axis, in correspondence with the rules, which were established for speed diagrams. These arcs are also parabolas. The loops can be drawn by means of their tangents, as is indicated by the light lines in the Figs. 20–23.

In Fig. 24 another case is shown in which the stress changes according to a harmonic function of the time $\sigma = \sigma_0 \sin \omega t$; the corresponding "stress-strain curve" is now an ellipse, tangent to an oblique direction having an angle of inclination given by E. These tangents together with the horizontal ones are parallel to a pair of conjugate diameters of the ellipse. The higher the speed, the narrower becomes the ellipse. In the ϵ, u-plane a continuous diagram now appears as an ellipse. If a cycle is described the point P travels along the ellipse while the point



Figs. 16 and 17 Above (Fig. 16) Stress-Strain Curves for Uniform Rates of Stretching. Below (Fig. 17): Projection of Stress-Strain Curves on ϵ, u-Plane

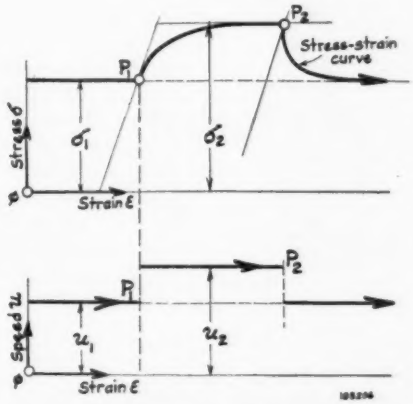


Fig. 18 Jumps in Stress-Strain Curves Caused by Sudden Changes of Rate of Stretching

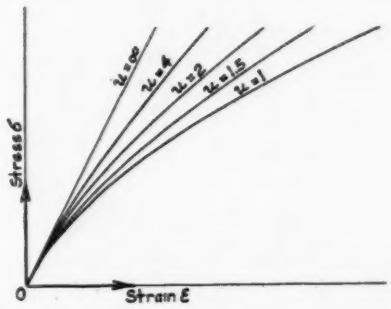
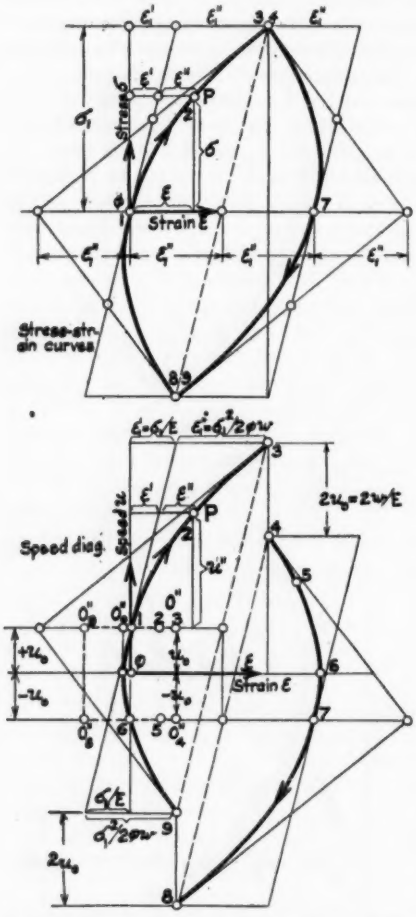
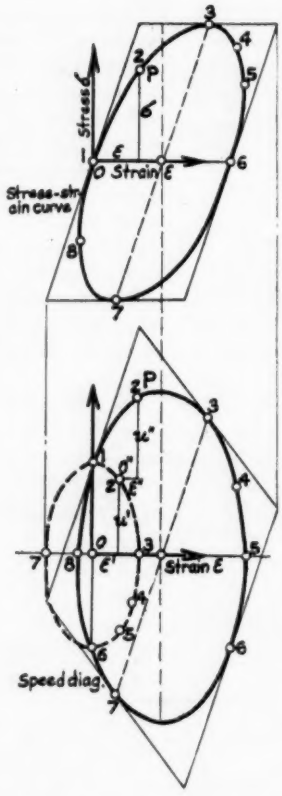


Fig. 19 Elastico-Viscous Material: Stress-Strain Curves for Constant Rates of Stress (w = Const.)

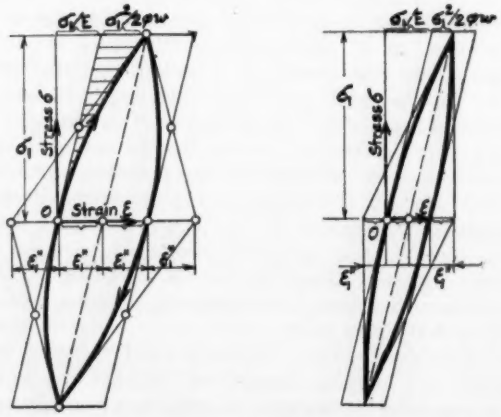
O'' describes another and smaller ellipse, which in Fig. 25 is indicated by a broken line; the figures 1, 2, 3, . . . refer to corresponding positions of the points P and O'', thus showing respectively the elastic and plastic portions of the strains and the rate of strain.



FIGS. 20 AND 21 ELASTICO-VISCOUS MATERIAL: HYSTERESIS LOOP FOR CONSTANT RATE OF STRESS (w = CONST.) DUE TO YIELDING



FIGS. 24 AND 25 ELASTICO VISCOUS MATERIAL: HYSTERESIS LOOP DUE TO YIELDING IF STRESS CHANGES ACCORDING TO A HARMONIC FUNCTION OF TIME



FIGS. 22 AND 23 ELASTICO-VISCOUS MATERIAL: HYSTERESIS LOOPS DUE TO YIELDING

4 *Relaxation.* As already mentioned, the phenomenon of relaxation of stress is a particular consequence of the plastic properties of solids and it manifests itself in numerous ways, due to various reasons to which it may be attributed. In the following, only the classical case of Maxwell's relaxation theory will be briefly reviewed. We may approach it by first stretching a bar at a given temperature according to one of the various condition curves described in the preceding sections. If the stretching is then interrupted and the length of the bar thereafter kept constant, we have for the following period of relaxation the condition that the *total strain* ϵ remains constant, or that

$$\epsilon = \epsilon' + \epsilon'' = \frac{\sigma}{E} + \epsilon'' = \epsilon_1 \dots\dots [66]$$

where ϵ_1 is the strain at the time $t = 0$ when the relaxation of stress begins. Let σ_1 be the stress at the same instant. Then, differentiating with respect to time,

$$\frac{d\epsilon}{dt} = \frac{d\epsilon'}{dt} + \frac{d\epsilon''}{dt} = 0 \dots\dots [67]$$

or

$$u = u' + u'' = 0 \dots\dots [68]$$

We see, therefore, that while there is no further rate of flow and the total speed of strain u vanishes, a process of relaxation of stress is still characterized by existing finite rates of elastic or plastic strain u' and u'' as the elastic part of the strain ϵ' will gradually be transformed into a permanent strain ϵ''.

Using now the law of viscous flow for the permanent part of the strain:

$$\sigma = \phi u'' = \phi \frac{d\epsilon''}{dt} \dots\dots [69]$$

we obtain from Equations [66] and [67]

$$\frac{1}{E}\frac{d\sigma}{dt} + \frac{\sigma}{\phi} = 0 \dots\dots [70]$$

or

$$\sigma = \sigma_1 e^{-Et/\phi} \dots\dots [71]$$

Stress, according to Maxwell's law of deformation, relaxes according to an exponential function of time, and the time t required for a given decrease of stress from a value σ_1 to a value σ is equal to

$$t = \frac{\phi}{E} \log_e \frac{\sigma_1}{\sigma} \dots\dots [72]$$

The more elastically rigid (E large) or the less viscous (ϕ or η small) a material is at a given temperature, the shorter will be the time of relaxation required for a given decrease of stress, and, on the contrary, the more elastic or the more viscous it is, the longer it will last, until stress vanishes. We see also that if such a relation would hold for the metallic materials, and there are certain reasons to believe that for extremely small rates of flow at least perhaps something similar may be not far from being true, the coefficient of viscosity η (or ϕ for tension) must decrease much more rapidly with increasing temperatures than the modulus of rigidity (or of elasticity). This is indeed well known for liquids or glasses (amorphous solids) because temperature seems to change the viscosities of liquids much more rapidly (according to exponential functions) than their volume elasticity (compressibility).

The idea of increasing the time of relaxation of a metallic bar exposed to an elevated temperature by allowing it to expand permanently against the contraction of additional elastic-spring members has been recently suggested by W. Barr and W. E. Bardgett[20] as a method for testing creep.

If a bar in tension be forced to creep while it is connected to a helical spring, the same formula would hold for an elastico-viscous material as that just derived, but with the difference that instead of the modulus of elasticity E of the bar, an "apparent modulus" of elasticity E would have to be applied, which would be given by:

$$E^* = \frac{E}{1 + cAE/l} \quad [73]$$

where E is the modulus of elasticity of the bar, A its area of cross-section, l the length, and c the spring constant of the helical spring. The time of relaxation would thus be increased in the proportion $E:E^*$ to

$$t = \left(1 + \frac{cAE}{l}\right)\frac{\phi}{E}\log_e\frac{\sigma_1}{\sigma} \quad [74]$$

where c will many times be larger than l/AE.[21]

It may be of interest here to note that it has been recently also suggested by W. Rohn[22] that creep of metals be tested by keeping the lengths of the bar as well as the stress constant, the creep being balanced against the thermal contraction due to decreasing temperature while the test is made. This is a somewhat different case of relaxation in which, however, primarily temperature is changed. It is well known that the elastic modulus as well as the thermal-expansion coefficient do change with temperature. From this it is seen that simple conditions in such a test will probably not be realized because at the same time at least three variables, namely, temperature, coefficient of thermal expansion, and the elastic modulus, are changing while creep occurs.

Summary

In the preceding, a number of idealized cases of plastic flow of a more or less simple nature have been reviewed with the primary intention of eliminating the effect of the speed of yielding on the other quantities to be observed when metallic bars are permanently stretched, twisted, or bent at normal or at elevated temperatures. Some of the main attempts to obtain or to derive quantitative results must be considered as of a preliminary nature until it has been sufficiently investigated, both theoretically as well as experimentally, whether such results can be established with a sufficient degree of exactness by using expressions in a finite form, or whether these results will have to be derived from differential relations connecting only the differentials of a path or a condition curve, to be integrated subsequently.

Acknowledgments

Acknowledgments are particularly due to Mr. P. G. McVetty, with whom during the last few years the questions referring to creep have been frequently discussed from various points of view. The author expresses further his indebtedness to Mr. L. W. Chubb, Director of the Research Laboratories of the Westinghouse Electric & Manufacturing Company in East Pittsburgh, and to Mr. R. E. Peterson, Manager of the Mechanics Division of that same company, for facilities afforded to him.

Appendix No. 1

Necking of a Bar of Viscous Material Under a Constant Load

Let P be the load. Let A be the variable cross-sectional area of the bar at the distance x and at the time t; A^* the original cross-sectional area at x for the time $t = 0$; A_0 the area at distance $x = 0$ and $t = 0$; and σ, σ^*, σ_0, σ_0^* the corresponding stresses [the asterisk refers to the original stage at time $t = 0$, the subscript zero to a given point ($x = 0$) of the bar—for example, the minimum section in the neck]. We then have

$$P = A^*\sigma^* = A\sigma \quad [75]$$

and as

$$A = A^*e^{-\sigma^* t/\phi} \quad [76]$$

$$A = A^*e^{-Pt/A^*\phi} \quad [77]$$

which shows how the cross-section A at a distance x will be changed with time t. For a numerical computation, it is convenient to introduce the ratios

$$k = A/A_0, \quad k^* = A^*/A^*_0 \quad [78]$$

The original profile of a bar will be thus given by the function k^* and this will depend only on x, while k will contain both x and t. Writing down now Eq. [77] for the minimum cross-section which shall be chosen at point $x = 0$, we must also have

$$A_0 = A_0^*e^{-Pt/A_0^*\phi} \quad [79]$$

It is convenient to introduce here the "time of half value" t_0. By this is to be understood the time at which the minimum area A_0 of the bar would neck down to half of its magnitude, so that for

$$\frac{A_0}{A_0^*} = \frac{1}{2} = e^{-\frac{Pt_0}{A_0^*\phi}} \quad [80]$$

we see that the half-value time t_0 is given by

$$t_0 = \phi\frac{A_0^*}{P}\log_e 2 = \frac{\phi}{\sigma_0^*}\log_e 2 = 0.693\frac{\phi}{\sigma_0^*} \quad [81]$$

We then have

$$A_0 = A_0^*e^{-\frac{t}{t_0}\log_e 2} \quad [82]$$

and

$$k = k^*e^{-\frac{(1-k^*)}{k^*}\frac{t}{t_0}} \quad [83]$$

---

[20] "An Accelerated Test for a Determination of the Limiting Creep Stress of Metals," *The Engineer*, vol. 153 (1932), p. 241.

[21] At the time $t = 0$ the sum of the elastic extensions for the bar and spring is

$$\Delta l_0 + \Delta L_0 = \frac{Pl}{AE} + cP_0$$

and similarly, at the time t

$$\Delta l + \Delta L = \Delta l_0 + \Delta L_0,$$

from which, by introducing the unit elongations $\epsilon = \Delta l/l = \epsilon' + \epsilon''$ and stresses $\sigma = P/A$ for the permanent strain,

$$\epsilon'' = \frac{\Delta l}{l} - \frac{\sigma}{E} = \frac{(\sigma_0 - \sigma)}{E}\left(1 + \frac{cAE}{l}\right)$$

is found. The derivative of ϵ'' with respect to time is thus found to be

$$u'' = -\left(1 + \frac{cAE}{l}\right)\frac{1}{E}\frac{d\sigma}{dt}$$

instead of the former expression $u'' = \frac{1}{E}\frac{d\sigma}{dt}$.

[22] Festschrift zum 70. Geburtstage von Dr.-Phil. Dr.-Ing. E. H. Wilhelm Heraeus. Hanau. Verlag von G. M. Albertis, Hofbuchhandlung Bruno Clauss, 1930, p. 80.

which last equation indicates how the original profile given by the function k^* of the variable x has changed into the new bar profiles given by k at the time t.

To find the location of the area A after the stretching, we have to compute the displacement ξ of this cross-section:

$$\xi = \int_0^x \epsilon dx \quad \ldots\ldots [84]$$

Between the unit elongations ϵ at the variable distance x and the cross-sectional areas, A we have the relation

$$1 + \epsilon = \frac{A^*}{A} = e^{Pt/A^*\phi} \quad \ldots\ldots [85]$$

which furnishes the new coordinate $x' = x + \xi$ at which the new cross-section A will be found at the time t:

$$x' = x + \xi = \int_0^x e^{Pt/A^*\phi}\, dx \quad \ldots\ldots [86]$$

The two expressions, Equations [83] and [86], for k and x' determine the new profile of the bar. As an example, the case was worked out of a bar having originally a symmetrically constrained portion with the profile

$$A^* = \frac{A_0^*}{1 - \beta^2 x^2/a^2} \quad \ldots\ldots [87]$$

Here A_0^* designates obviously the minimum original section occurring at $x = 0$, β is a constant and a is half of the length of the constrained portion. Making the substitution

$$c^2 = \frac{Pt}{A_0^*\phi} \quad \ldots\ldots [88]$$

we find that

$$x' = \int_0^x e^{\frac{Pt}{A_0^*\phi}\left(1 - \frac{\beta^2 x^2}{a^2}\right)} dx = e^{c^2} \int_0^x e^{-\frac{\beta^2 c^2 x^2}{a^2}} dx. \quad [89]$$

or by using z as a new variable for the integration

$$z = \beta c x/a \quad \ldots\ldots [90]$$

$$x' = e^{c^2} \frac{a}{\beta c} \int_0^z e^{-z^2} dz \quad \ldots\ldots [91]$$

the probability function

$$\phi(z) = \frac{2}{\pi} \int_0^z e^{-z^2} dz \quad \ldots\ldots [92]$$

$$x' = \frac{a e^{c^2} \sqrt{\pi}}{2c\beta} \phi(z) \quad \ldots\ldots [93]$$

The bar profiles for this case were computed with $\beta^2 = 1/2$ for the two times $t = t_0$ and $t = 2t_0$. The original profile of the bar was

$$A^* = \frac{A_0^*}{1 - x^2/2a^2} \quad \ldots\ldots [94]$$

It may be seen from Fig. 12, which contains the results of the computation, that while in the first time period equal to the half-value time t_0, the minimum section of the bar was decreased to half of its original magnitude and the constrained portion of length $2a$ was stretched by 60 per cent, during the following equal time period, which produced an equal proportional reduction of area at the neck, the constrained portion was stretched to 3.24 times its original length.

The following table shows how the shape of the bar has changed:

Original distance from neck:	$x = 0$	$0.25a$	$0.50a$	$0.75a$	1
Ratio of areas originally, at time $t = 0$	$A/A_0^* = 1$	1.03	1.14	1.39	2
Distance x' from neck at time $t = t_0$	$x' = 0$	..	0.98	1.41	1.60
Ratio of areas at time $t = t_0$	$A/A_0^* = 0.05$	..	0.65	0.92	1.65
Distance x' from neck at time $t = 2t_0$	$x' = 0$	0.94	1.89	2.65	3.24
Ratio of areas at time $t = 2t_0$	$A/A_0^* = 0.25$	0.28	0.37	0.61	1.36

Appendix No. 2

CREEP OF CYLINDRICAL BAR IN TORSION[23]

As a last application may be considered the case of a cylindrical bar which is twisted by a couple, assuming an arbitrary law of deformation.

Let r = distance of a point P of the bar from its axis (Fig. 26)
a = radius of the bar
t = time
ϕ = angle of twist per unit of length of the bar
$\omega = d\phi/dt$ = angular speed with which it is twisted
γ = unit shear at the distance r from the axis of the bar
$v = d\gamma/dt$ = rate of shear at distance r
τ = shearing stress at distance r
M = torque, and
$\tau_a\ \gamma_a\ v_a$ = quantities referring to the outer surface ($r = a$) of the bar.

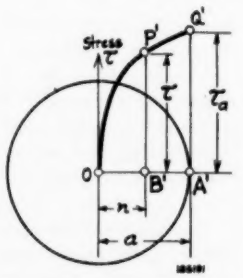


FIG. 26 TORSION SHEARING STRESSES [$\tau = f(r)$] IN ROUND BAR

As the case of plastic static torsion has already been treated by the author,[24] the following will refer to the influence of the speed of deformation only. All points in the interior of a round bar in torsion are subjected to similar states of stress, namely, pure shear. Due to the relative twist of two cross-sections of the bar a unit of length apart, unit shear γ is produced at the distance r from the axis which is equal to

$$\gamma = r\phi \quad \ldots\ldots [95]$$

This equation differentiated with respect to the time t gives

$$\dot{\gamma} = r\dot{\phi} \quad \ldots\ldots [96]$$

whence the rate of shear is found equal to

$$v = r\omega \quad \ldots\ldots [97]$$

A small element on the circumference of the circular cross-section ($r = a$) will deform with a rate of shear equal to

$$v_a = a\omega \quad \ldots\ldots [98]$$

while it will be distorted to a unit shear

$$\gamma_a = a\phi \quad \ldots\ldots [99]$$

[23] Creep tests in torsion with metals are described by R. W. Bailey (paper presented at Tokio World Power Conference, 1929), to whose work special reference is made. Cf. also Everett's paper already quoted.

[24] Trans. A.S.M.E., 1929.

From these equations, we see that at any time t

$$\frac{\gamma}{v} = \frac{\gamma_a}{v_a} = \frac{\phi}{\omega} \quad \ldots\ldots [100]$$

the rates of shear v in the bar are proportional to the unit shears γ. It will be assumed that for the state of pure shear a stress surface

$$\tau = F(\gamma,v) \quad \ldots\ldots [101]$$

is known. This function could perhaps be obtained from experiments made under constant angular speeds in a way similar to that by which the stress-strain curve for pure shear was derived from a static torsion test with a round bar. If a bar is twisted through an angle ϕ at an angular speed ω, the distorsion γ_a of a small element at a point A on the circumference $r = a$ of the bar will be known and given by [95] and the rate of shear v_a at the same points will be given by [97]. To these values of γ_a and v_a (Fig. 27) as abscissas will correspond a given shearing

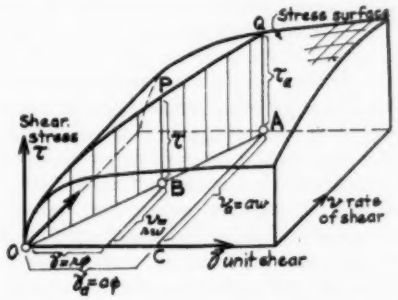


FIG. 27 STRESS SURFACE FOR PURE SHEAR

stress τ_a as the ordinate AQ in the stress surface [101]. Thus the maximum shearing stress τ_a acting in the point A' of the cross-section (Fig. 26) at the time t will be known. With this also the shearing stresses τ at any other point B' (Fig. 26) will be given, because at any point B' distant r from the axis of the bar, according to [95], [97], and [100] γ and v are known. Interpreting the variables γ and v as rectangular coordinates, Equation [100] or

$$\frac{\gamma}{v} = \frac{\gamma_a}{v_a} \quad \ldots\ldots [102]$$

is the equation of a plane passing through the origin O and the point A in the basic plane of the stress surface, Fig. 27. This plane evidently intersects this surface in a plane section, the ordinates of which represent the shearing stresses τ acting at the time t in the bar. The section $OBAQP$ of Fig. 27 has to be drawn over the radius $O'A' = a$ of Fig. 26, using the same ordinates τ but changing the scale in the horizontal direction in the proportion of the length $O'A':OA$. The equation of the section through the stress surface [101] is given by

$$\tau = F\left(\gamma, \frac{\gamma\omega}{\phi}\right) \quad \ldots\ldots [103]$$

and the torque M is thus given by the integral

$$M = 2\pi \int_0^a \tau r^2 dr = \frac{2\pi a^3}{\gamma_a^3} \int_0^{\gamma_a} F\left(\gamma, \frac{\gamma\omega}{\phi}\right) \gamma^2 d\gamma \ldots [104]$$

where the upper limit is

$$\gamma_a = a\phi \quad \ldots\ldots [105]$$

This is the moment required to twist a round bar by an angle ϕ and at an angular speed ω.

TORSION TESTS MADE AT A UNIFORM ANGULAR SPEED ω = CONST.

In this case the angle of twist of the round bar is

$$\phi = \omega t \quad \ldots\ldots [106]$$

At a time t the unit shear γ_a at the circumference $r = a$ of a section will be

$$\gamma_a = a\omega t \quad \ldots\ldots [107]$$

and the rate of shear at this point (represented by AC in Fig. 27) will be given by

$$v_a = a\omega = \gamma_a/t \quad \ldots\ldots [108]$$

If now ω = const., the rate of shear v_a will also be v_a = const. and the unit shear $\gamma_a = v_a t$ will increase uniformly with the time. This means that the vertical section $OAQO$ through the stress surface will have to be turned around point O so that point A describes a line AE parallel to the γ-axis. To obtain subsequent distributions of stresses $\tau = f(r)$ in the bar for various angles of twist $\phi = \omega t$, the vertical section $OAQP$ (Fig. 27) which cuts this distribution out from the stress surface $\tau = F(\gamma v)$ has to be turned around the origin in such a way that its end point A moves along the straight line v_a = const. or along AE with uniform speed. Reducing these sections (in the horizontal direction always) to the length $O'A' = a$ of Fig. 26 furnishes the various stress curves $\tau = f(r)$ above the radii r, and the corresponding torques M according to Eq. [104]. A torsion test made under a uniform angular speed thus produces a maximum uniform rate of shear v_a = const. at the circumference of the cross-section of the bar.

These relations may be applied to forms of the stress surface which are expressible by analytical means, such as power functions or others, or are known by graghs. If the influence of the speed of deformation could perhaps be expressed by a stress function of the form

$$\tau = \tau_0(1 + \eta v) \quad \ldots\ldots [109]$$

where τ_0 would depend only on the unit shear γ, but not on the rate of shear v:

$$\tau_0 = f(\gamma) \quad \ldots\ldots [110]$$

then a torsion test made by twisting a round bar at a uniform angular speed ω = const. could produce a rate of shear v at any point distant r from the axis, which would be equal to

$$v = \frac{v_a}{\gamma_a}\gamma = \frac{\omega}{\phi}\gamma \quad \ldots\ldots [111]$$

Hence along the section of the stress surface [109] in which v would depend on γ as expressed by [111], the shearing stress τ would be

$$\tau = f(\gamma)\left[1 + \frac{\eta\omega}{\phi}\gamma\right] \quad \ldots\ldots [112]$$

and the corresponding torques would be equal to

$$M = 2\pi \int_0^a \tau r^2 dr = \frac{2\pi a^3}{\gamma_a^3} \int_0^{\gamma_a} \left[1 + \frac{\eta\omega}{\phi}\gamma\right] f(\gamma)\gamma^2 d\gamma. \quad [113]$$

or

$$M = \frac{2\pi a^3}{\gamma_a^3}\left[\int_0^{\gamma_a} f(\gamma)\gamma^2 d\gamma + \frac{\eta\omega}{\phi}\int_0^{\gamma_a} f(\gamma)\gamma^3 d\gamma\right]. \quad [114]$$

This can be written also in the form

$$M = M_0 + M_1 \ldots\ldots\ldots\ldots\ldots\ldots [115]$$

where M_0 designates the part obtained using the first term in the expression [114] of the torque M, and M_1 the remaining term. To both terms there could be attributed a simple physical meaning. The stress function $\tau_0 = f(\gamma)$ would be what could be called *the infinitely slow stress-strain diagram for pure shear.* The function $\tau_0 = f(\gamma)$ would express the fact that for the rate of shear here a finite diagram exists and the first portion M_0 in the expression [114] for the torque M,

$$M_0 = \frac{2\pi a^3}{\gamma_a{}^3} \int_0^{\gamma_a} f(\gamma)\gamma^2 d\gamma \ldots\ldots\ldots\ldots [116]$$

would be the "static torque," as derived from the static theory of torsion, corresponding to a maximum unit shear $\gamma_a = a\omega t$ or an angle of twist $\phi = \gamma_a/a = \omega t$. The second term in [114],

$$M_1 = \frac{2\pi a^3 \eta\omega}{\gamma_a{}^3\phi} \int_0^{\gamma_a} f(\gamma)\gamma^3 d\gamma \ldots\ldots\ldots\ldots [117]$$

would be the "correction due to the effect of speed of deformation" which would have to be added to M_0.

If, for example, the static stress-strain curve $\tau_0 = f(\gamma)$ for pure shear could be approximated by a power function[25] of γ:

$$\tau_0 = \tau_1 \left(\frac{\gamma}{\gamma_1}\right)^n, \quad (0 < n < 1) \ldots\ldots\ldots\ldots [118]$$

then the static torque required to twist a round bar to a maximum shearing stress τ_a would be equal to

$$M_0 = \frac{2\pi a^3 t_a}{n+3} \ldots\ldots\ldots\ldots\ldots\ldots [119]$$

and the actual torque M would be given by:

$$M = M_0 \left[1 + \frac{n+3}{n+4}\eta a\omega\right] \ldots\ldots\ldots\ldots [120]$$

In the well-known case of true viscous flow, which has been frequently considered in the literature,[26] neglecting the elasticity of the material, the stress surface $\tau = F(\epsilon, v)$ is a plane:

$$\tau = \eta v \ldots\ldots\ldots\ldots\ldots\ldots [121]$$

and for a torsion test made at a uniform angular speed ω = const. or with $v_a = a\omega$ = const.,

$$\tau_a = \eta v_a = a\eta\omega$$

the shearing stress τ is

$$\tau = \tau_a \frac{r}{a} = \eta\omega r \ldots\ldots\ldots\ldots\ldots\ldots [122]$$

and the torque

$$M = \frac{\pi\eta\omega a^4}{2} \ldots\ldots\ldots\ldots\ldots\ldots [123]$$

This would remain constant as the bar is twisted further with the uniform angular velocity ω = const., but would change proportionally to ω if ω assumed another value.

It is comparatively easy to derive analogous expressions using other laws of deformation, for example, for the case of a logarithmic law of speed, which will not be further considered here. We may also use the stress surface to solve or answer questions which refer to torsion tests made under a given torque M = const. This would perhaps be of some interest also, for various suggestions have been made recently to use the torsion test of a round bar in creep tests, keeping the torque constant. Analogous to what was said previously with reference to tensile tests made under constant loads, it is required now to plot the torques M which were obtained, for example, from torsion tests made under uniform rates of shear or uniform angular velocities ω = const., above the same coordinates γ and v used in the expression of the stress surface $\tau = F(\gamma, v)$. Thus a second surface and an M or torque surface would be obtained, and it would be necessary to study finally the shapes of the curves M = const. on this torque surface:

$$M = H(\gamma, v) \ldots\ldots\ldots\ldots\ldots\ldots [124]$$

The family of condition curves M = const. on this surface, that is, the contour lines thereof, as they appear in a projection on the base plane γ, v would describe the torsion tests made under a constant torque. It is intended to treat this subject on a later occasion in more detail, when more experimental evidence regarding the shape of actual stress surfaces may be available. It is obvious that constructing the M-surface for the torsion tests will be a somewhat more tedious process than introducing the simpler P-surface in the tension test, because the values of the torques can be obtained only by an integration process from the shearing stresses τ and conversely, deriving τ-values of observed torques or M-values may require a somewhat tedious process of differentiation. It has already been mentioned that the problem of creep in pure bending for a bar having a simple and uniform cross-section leads to quite an analogous or, in special cases, nearly identical theory to that which has been briefly indicated for the case of torsion of a round bar.

Discussion

MAYO D. HERSEY.[27] The author's method of representing creep phenomena by a series of ideally simplified limiting cases should be of considerable scientific and educational value. In a similar attempt of a less comprehensive scope it was shown how certain types of material might give an ellipse on the load vs. deformation diagram when the load is applied sinusoidally;[28] also a diagram was constructed indicating how certain phenomena would appear when the deformation is represented as a function of two variables, the applied load and the time elapsed.[29] It is interesting that the results of these various studies have so much in common when the underlying assumptions are so different.

R. G. STURM.[30] This paper, which presents a number of idealized cases of plastic flow, certainly aids in visualizing what might be taking place as a metal continues to deform under a sustained load.

Nearly all of the references mentioned by the author deal with creep at high temperatures. The fact that the temperatures were not the same complicates the processes of arriving

[25] This formula cannot be a correct expression of the stress for very small unit shears γ, because in this region it would correspond to an infinite modulus of shear. For a larger γ however, it may perhaps be used to obtain numerical data in the first approximation.

[26] See, for example, Bingham, "Fluidity and Plasticity," McGraw-Hill Book Co., New York.

[27] Research and Development Dept., Vacuum Oil Company, Inc., Paulsboro, N. J. Mem. A.S.M.E.

[28] M. D. Hersey, "On the Theory of Irreversible Time Effects," *Jl. Wash. Acad. Sci.*, vol. 11 (1921), pp. 149–155.

[29] M. D. Hersey, "Diaphragms for Aeronautic Instruments." Rep. No. 165, Nat. Adv. Com. Aero., Washington, D. C., 1923, pp. 155–186.

[30] Research Engineer-Physicist, Aluminum Research Laboratories, Aluminum Company of America, New Kensington, Pa.

at some fundamental relations between creep, stress, time, temperature, etc. Creep tests on a number of metals, including copper, aluminum, steel, and monel metal, have been made at room temperature at the Aluminum Research Laboratories at New Kensington. In these tests creep in direct stress was measured on sheet and wire specimens, and creep in bending was measured on simple beams and constant-stress cantilevers. The results thus far obtained indicate that many metals will creep a very small but definite amount at very low stresses at room temperatures; also that creep continues for an indefinitely long period of time. In view of these indications it would seem that the determination of a limiting creep stress below which no creep occurs would depend upon the sensitivity of the measuring apparatus much the same as does the determination of the proportional limit.

The determination of the maximum limit of creep would also be very indefinite because, after a period of six years, specimens have been found to continue to deform. In view of the results of the tests made at room temperature and the great number of results of tests at high temperatures recently published, it would seem that the designation of creep should include not only the amount of creep in a definite period of time for various stresses, but also a factor representing the rate of change of creep at the end of this period of time. Such a definition of the creep characteristics of a metal would necessarily be of an arbitrary nature, but the values could be readily used by the designer and at the same time could be definitely determined in tests, and the results obtained by various investigators correlated. Just what values of time and creep should be used for defining the creep characteristics of materials cannot be definitely established until more information in regard to creep at a given temperature is available.

C. Richard Soderberg.[31] In the earlier investigation of the physical properties of materials under elevated temperatures, the existence of a stress-strain curve was tacitly assumed. A considerable amount of the information collected during the past ten years has been based on this fallacious assumption, with the result that much of it is worthless, particularly the results obtained by vaguely defined short-time tests. The long-time creep tests have yielded much valuable information, but when the results are judged in the light of the expense and effort applied, the overall efficiency of the work is disappointingly low.

It is not given to human beings to be able to carry out well-controlled laboratory tests extending over years; something invariably happens to spoil the continuity, and the point of view is continually changing. This does not mean that the efforts applied so far have been entirely wasted, but it is beginning to be understood that real progress on this subject will require fundamentally different methods.

[31] Turbine Apparatus Division Engineer, Westinghouse Elec. & Mfg. Co., Philadelphia, Pa. Mem. A.S.M.E.

The paper represents a valuable contribution in this respect. It will probably be criticized by many as academic and somewhat remote from the sphere of practical applications, but the writer is convinced that it represents the beginning of a rational treatment of the subject. It is only through efforts along the lines of systematic research suggested that we shall arrive at an understanding of the twilight region between the elastic and viscous behavior of our engineering materials.

At present there appears to be more uncertainty with regards to the most profitable methods of testing in high-temperature work than formerly. The short-time tensile test is admittedly of no value without control of the creep rate; the long-time creep test is entirely too cumbersome. The most promising method of approach appears to be tensile tests under high values of creep rates. The extrapolation to lower creep rates of immediate practical importance must be accomplished through empirical formulas of the type given by Equation [1].

The definite establishing of the stress-creep rate equation for our engineering material is thus one of the most important problems at the present time.

Author's Closure

As stated in the paper, details with reference to elastic after-effects have been intentionally omitted in what precedes, the investigation being confined to pure creep phenomena or to a few of the simplest imaginable relations existing between rates of flow and stresses, including also the action of elasticity. The author is aware of not having quoted work as far as this referred also to the elastic after-effects, and he acknowledges the remarks made by Dr. M. D. Hersey, indicating that similar relations were described by him under other assumptions as to the nature of these effects.

The observations to which Mr. R. G. Sturm refers, pointing to the extensive experimental work done at the Research Laboratories of the Aluminum Company in New Kensington on the creep of aluminum at normal temperature, are valuable because of the additional experimental evidence to which they point of the non-existence of the so-called creep limit for a metal such as aluminum when tested at temperatures at which recrystallization does not yet become appreciable.

The author appreciates the remarks by Mr. C. R. Soderberg. They seem to point to the demands of the designers of heavy machinery to have more reliable or simpler rules at their disposal when machine parts are to be built which will have to withstand prolonged stresses at elevated temperatures. Here, as in other similar cases, the gathering of information of an empirical nature has preceded other attempts. The value of the long-time creep test was, and still is, to furnish the first reliable data on which such considerations can be based. It can be hoped that the results of these tests in conjunction with a rational description of rules may help also sooner or later to pave the way for a simplification also of the technique of these tests.

APM-55-11

Model Testing as Applied to Strength of Materials

By R. E. PETERSON,[1] EAST PITTSBURGH, PA.

This paper is primarily concerned with the effect of size on the ordinary strength properties of materials. The use of models in stress analysis is described briefly. The effect of size in tension, fatigue, and impact tests is considered in detail. Possible explanations for departure from the law of similarity are stated.



THIS paper is concerned primarily with the effect of size on the strength properties as determined from specimens and models tested to destruction. Models are, however, often used in the elastic range, particularly in the field of stress analysis. A few such applications will now be mentioned, but very briefly, since no problem of size effect exists so long as elastic conditions are maintained. Problems of plane stress can be solved photoelastically[2] by the use of models of transparent material (such as celluloid or bakelite) stressed in a field of polarized light. Statically indeterminate structures can be studied by means of the deformator method,[3] making use of celluloid or fine-quality cardboard. Based on the torsion analogy of Prandtl, a soap-film method[4] has been used to determine stress distributions of irregular sections in torsion. Rubber models also have been used with some success in studying stress concentration; a recent promising method[5] involves measurements at a series of deformations, the plotted results being extrapolated to zero deformation. The foregoing methods all apply to the elastic range of materials, and as previously mentioned, the size of the model has no effect on the results.

However, in the field of strength of materials where models are tested to destruction, it is necessary to consider carefully the effect of size on the ordinary strength properties of materials, such as ultimate strength, endurance limit, and impact value. Such information must necessarily form a basis for model testing, and is also of prime importance to designers and engineers concerned with specification of materials and standardization of tests.

The Tension Test

Under this category are considered all tests made at slow speed in the manner of the ordinary tensile test as performed in material-testing laboratories. The tensile-test specimens that have been standardized in various countries are of constant diameter over a gage length, free from stress-concentration effects. For such specimens the law of similarity was found to apply by Barba[6] and Kick.[7] A plot of some of the data of Barba is shown in Fig. 1. Confirmatory results have been obtained by Bauschinger,[8] Martens,[9] Rudeloff,[10] Moser,[11] and others. Compressive tests made under similar conditions by Kick, Bauschinger,[12] and others show agreement with the law of similarity. It would seem, therefore, that for slow loading of specimens free from stress concentration, the law of similarity holds for ordinary engineering materials.

An exception is apparently found in specimens of very small diameter. Wires are known to have a higher tensile strength than standard test specimens of the same material, but the effect of working is generally present. In considering the size effect, obviously variables due to cold working, quenching, surface finish, etc. must be avoided. However, very small-diameter glass fibers[13] have exhibited unusually high tensile strengths. High values also have been obtained for annealed silver and gold threads.[14] If, however, exceptionally small diameters are excluded and if in metallic specimens a large enough number of grains are present to produce a statistical effect (which conditions are generally fulfilled in engineering design), then the law of similarity may be considered to apply to the slow loading of tensile specimens free from stress-concentration effects.

The effect of size in tensile specimens having stress concentration effects (due to notches or other abrupt changes of contour) has not been thoroughly investigated. Results recently reported by Kuntze[15] indicate that for geometrically similar notched

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Contributed by the Applied Mechanics Division and presented at the Semi-Annual Meeting of The American Society of Mechanical Engineers, Bigwin Inn, Lake of Bays, Ontario, Canada, June 27 to July 1, 1932.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

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[13] Griffith, A. A., "Phenomena of Rupture and Flow in Solids," Phil. Trans. Roy. Soc., vol. 221-A (1920), p. 163.

[14] Karmarsch, Mitt. des gew. Ver. für Hannover (1858), p. 138.

[15] Kuntze, W., "Plastizität und Festigkeit bei Einkerbungen," *Zeit. für Physik*, vol. 74 (1932), p. 45.

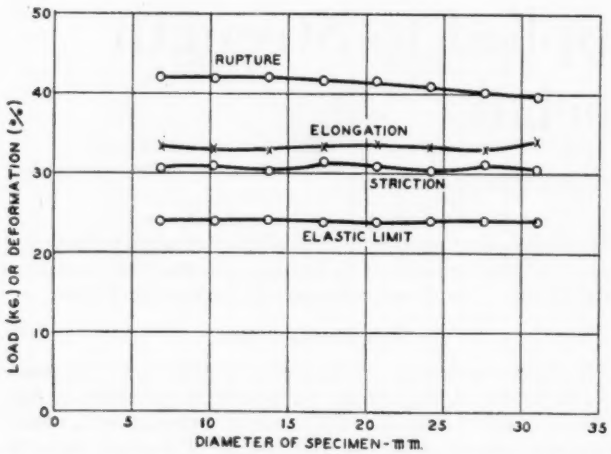


FIG. 1 PLOT OF DATA OF BARBA ON TENSILE TESTS OF MILD STEEL

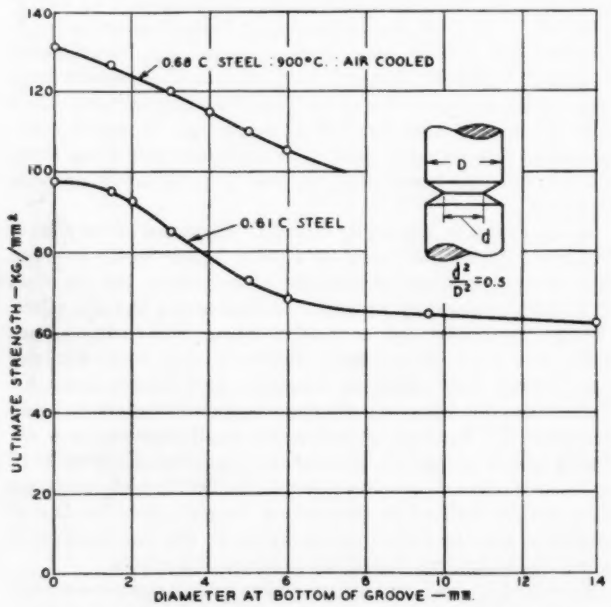


FIG. 2 TENSILE TESTS OF GROOVED SPECIMENS (KUNTZE)

tensile specimens, small sizes give higher ultimate strengths, as shown in Fig. 2. Kuntze makes use of an idea of Föppl[16] in attributing the phenomenon to an outer layer that acts in a manner different from that of the material inside, somewhat analogous to the phenomenon of capillarity. Kuntze also mentions the possibility of a critical amount of slipping, independent of size, which causes departure from the law of similarity if the number of slip planes is artificially limited, as in a notched specimen. The use of a sharp notch by Kuntze was somewhat unfortunate from the standpoint of geometrical similarity, since actually a microscopic rounding exists at the notch bottom. Recent tests by Dr. Nadai and Mr. MacGregor, of Westinghouse Research Laboratories, indicate no appreciable size effect for steel and aluminum specimens having notches with definite small radii at the notch bottom. It is expected that these data will be published during the forthcoming year.

[16] Föppl, A., "Drang und Zwang," vol. 11, p. 236.

THE FATIGUE TEST

Data on size effect in fatigue are very meager. Equipment has been developed for making fatigue tests of large specimens[17] up to $3^3/_{16}$ in. in diameter and of small specimens[18] down to 0.050 in. in diameter.

In testing size effect, it is essential that other variables are excluded, and hence the material must be uniform—a require-

[17] Peterson, R. E., "Fatigue Tests of Large Specimens," Proc. Am. Soc. Test. Matls., vol. 29, Part II (1929), p. 371.

[18] Peterson, R. E., "Fatigue Tests of Small Specimens With Particular Reference to Size Effect," Proc. Am. Soc. Steel Treat., vol. 18 (1930), p. 1041.

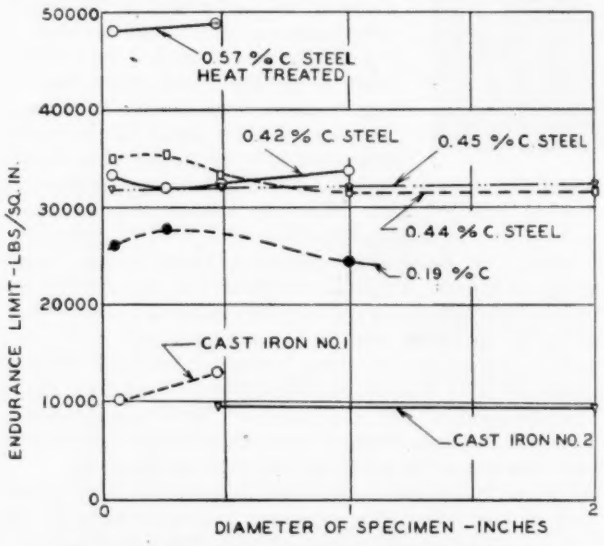


FIG. 3 EFFECT OF SIZE ON THE ENDURANCE LIMIT
(Without stress concentration.)

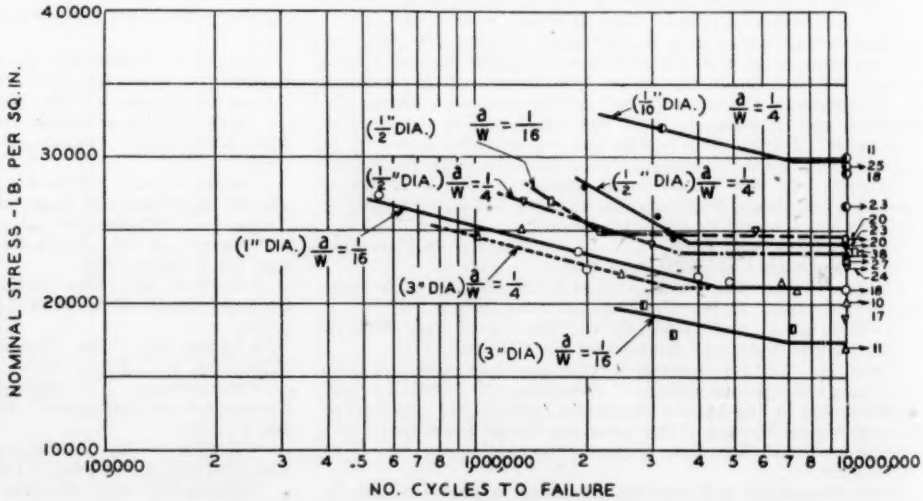


FIG. 4 FATIGUE DATA—SPECIMENS WITH HOLES
(0.45% C steel.)

ment difficult to meet and one that is perhaps of greater importance in dynamic tests. In the investigation of size effect in fatigue it was found necessary that specimens be made according to the following requirements: (1) Material near the outside of a bar or at the center of a bar should not be used as the surface of a test specimen. (2) Small specimens should be taken from the same portion of the bar as that used for the surface of the largest specimen. Thus if a specimen 2 in. in diameter is the largest of a series, the small specimens (say, 0.05 in. in diameter) should be taken from the bar at a radial distance of 1 in. from the center of the bar. (3) All specimens should be taken from the same bar, the ends of the bar not being used. (4) Hardness readings should be made on all specimens.

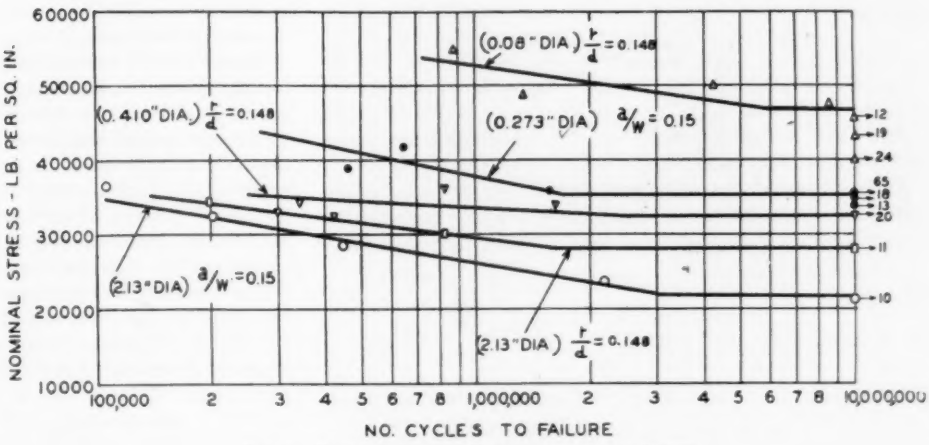


FIG. 6 FATIGUE DATA—FILLET AND HOLE TESTS
(0.57% C steel, heat treated.)

In testing fatigue specimens varying in diameter from 0.05 in. to 2 in., no important size effect was found[18] for the steels tested (carbon contents, 0.20, 0.42, 0.44, 0.57 per cent), as shown in Fig. 3. For cast iron widely scattered results were obtained with very small specimens,[18] which was not surprising, since the cast iron was unusually porous and flaky. Subsequent tests showed no size effect in larger sizes (Fig. 3). Mr. R. L. Templin states[19] that he has found no size effect in fatigue tests of cast aluminum alloys. Prof. H. F. Moore, assisted by Mr. Schorah, made fatigue tests of specimens of two different diameters and found no size effect[19] for medium carbon steel. For steels of low and high carbon contents a size effect was found. Further tests of this kind are necessary.

In the fatigue tests discussed up to this point, smooth contours without stress concentration were tested in each case. It is of practical importance to study the effect of size in fatigue tests where stress-concentration effects are present, since this is practically always the case in machines, due to the presence of fillets, holes, grooves, thread, keyways, notches, etc. The results of the following fatigue tests will be given in terms of stress-concentration factor[20] $k = s_e/s_e' =$ the ordinary endurance limit divided by the endurance limit of specimens having stress concentration.

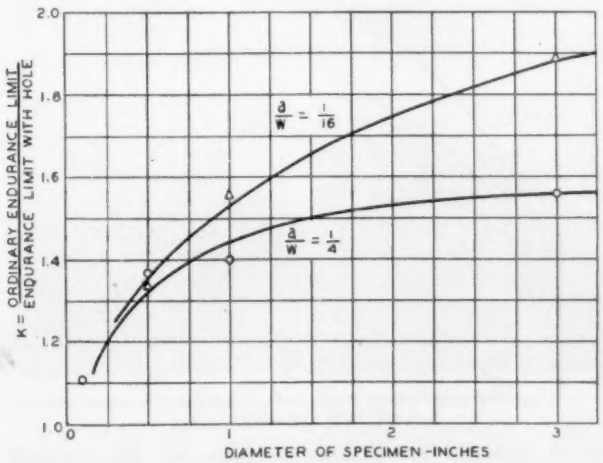


FIG. 5 FATIGUE STRESS CONCENTRATION FOR SPECIMENS WITH HOLES
(0.45% C steel.)

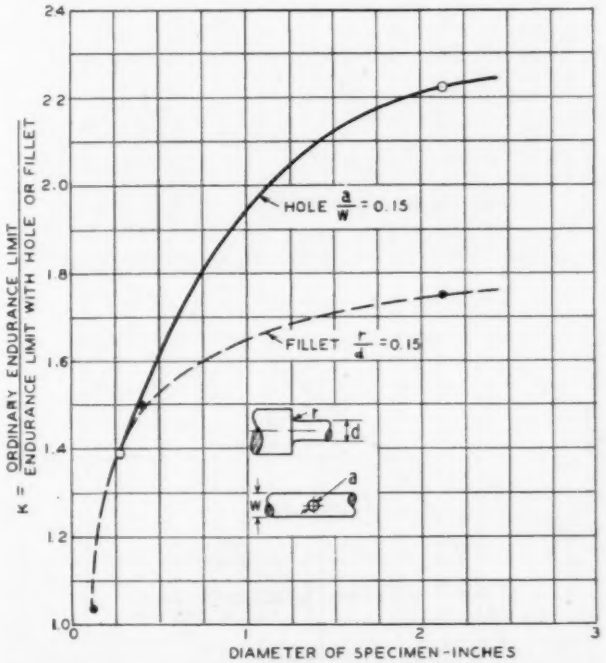


FIG. 7 FATIGUE STRESS CONCENTRATION FACTORS
[Fillet and hole tests (0.57% C steel, heat treated).]

Tests were made of the effect of holes in fatigue specimens of various diameters. The material used was an unusually clean medium carbon steel made under the supervision of Dr. C. H. Herty, Jr. The analysis in per cent: C, 0.45; Mn, 0.79; Si, 0.18; S, 0.032; P, 0.013. Mechanical properties: Yield point (0.2 per cent plastic deformation), 32,500 lb per sq in.; ultimate strength, 76,000 lb per sq in.; elongation (2 in.), 32 per cent; reduction of area, 50 per cent. The endurance limit was found to be 33,000 lb per sq in., the same for large and small specimens (Fig. 3). In investigating the size effect of specimens having holes, it is necessary to keep constant a/w = diameter of hole

[19] Private communication.

[20] Report of Research Committee on Fatigue of Metals (Appendix), "Present-Day Knowledge of Fatigue," Proc. Am. Soc. Test. Matls., vol. 30, Part I (1930), p. 298.

divided by diameter of specimen. This was done for two values $a/w = 1/4$ and $1/16$. In computing nominal stresses for the specimens having holes, the net section was used in the determination of the moment of inertia. From Fig. 4 it is apparent that the endurance limits for small specimens having holes are higher than for large specimens. The corresponding stress-concentrations factors are shown in Fig. 5.

Further test data for specimens having holes are summarized in Figs. 6 and 7, the material being 0.57 per cent carbon steel, heat treated.[18] For the same material, fatigue tests were made

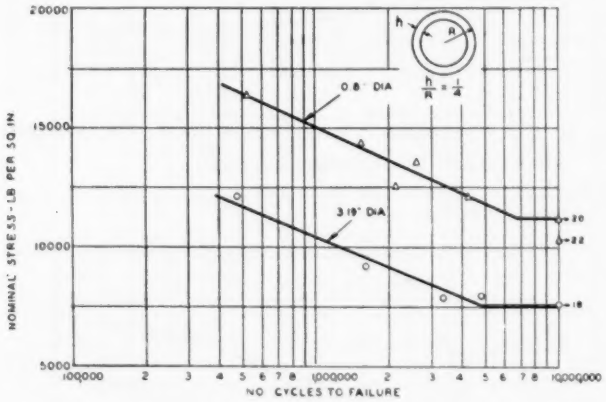


FIG. 8 FATIGUE DATA
[Specimens with circumferential cracks (0.44% C steel).]

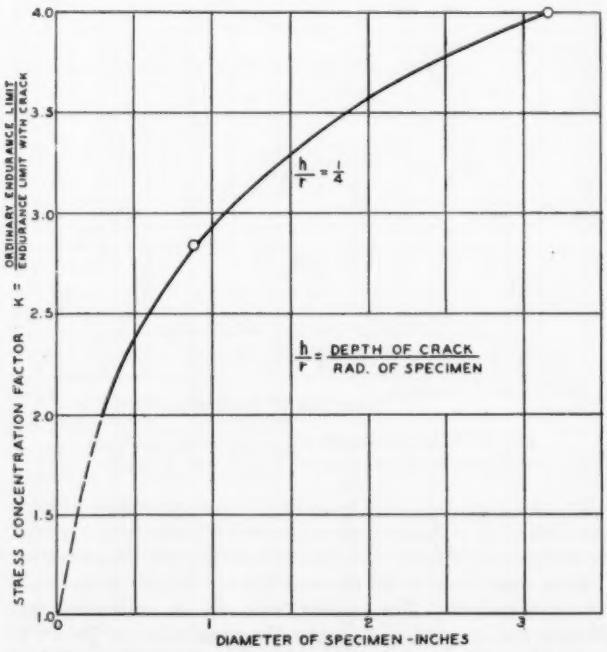


FIG 9 FATIGUE STRESS CONCENTRATION FACTORS FOR SPECIMENS HAVING ARTIFICIAL CRACKS
(0.44% C steel.)

of specimens having fillets with r/d = radius of fillet divided by diameter of specimen = 0.148. The results of the fillet tests are also given in Figs. 6 and 7.

The reason for the apparent size effect is not altogether clear. Different degrees of cold work due to the machining of different sized discontinuities may be an important variable affecting the results, particularly in the case of small discontinuities. Further possible reasons for departure from the law of similarity will be discussed in the closing section of the paper.

In order to test a severe case of stress concentration, specimens having artificial cracks were produced, the method being suggested by Mr. P. G. McVetty. A very narrow V-bottomed groove was turned in a round bar, which was heated and compressed axially to close the groove. The bar was subsequently annealed and machined into a test piece having a circumferential crack. Two series of such specimens were tested, 0.8 in. and $3^3/_{16}$ in. in diameter, with h/r = depth of crack divided by radius = $1/4$. The material was 0.44 per cent carbon steel.[18] The nominal stresses were computed using a moment of inertia based on the following assumptions: (1) Linear stress distribution, (2) crack carries compression, but not tension. The final results are summarized in Figs. 8 and 9. It is intended to present more details of these tests in a future publication, since in this paper only data bearing on size effect are given.

The size-effect tests of specimens having cracks may throw some light on corrosion-fatigue results with different diameter specimens reported by McAdam.[21] The corrosion-fatigue graph for large specimens 1.7 to 2.3 in. in diameter (250 to 400 rpm) was lower than the graph for small specimens $1/2$ in. in diameter (250 to 400 rpm), which in turn was lower than the graph for $1/2$-in. specimens (1450 rpm). Corrosion fatigue involves "notching," and consequently some similarity with crack tests might be expected. Evidence on hand indicates that deeper cracks have higher stress-concentration factors, in some such manner as shown in Fig. 10. If corrosion penetration does not depend on size, h/r would be different for the large and small specimens at slow speed (Fig. 10). The small specimen at high speed would have a lower stress-concentration factor because of reduced time for corrosive action. The results of McAdam line up in the same order as indicated in Fig. 10. The foregoing analysis would be more applicable in the case of prior-corrosion-fatigue tests, but may also have some bearing in corrosion-fatigue tests.

It is apparent from Figs. 5–10 that the stress-concentration

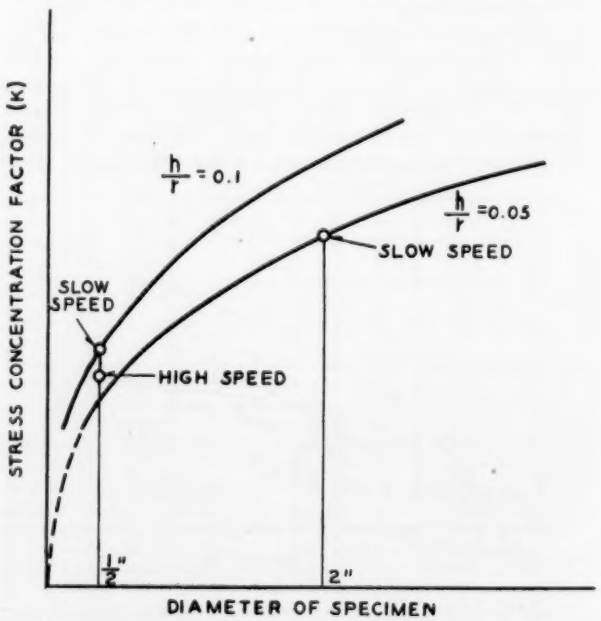


FIG. 10 POSSIBLE EXPLANATION OF CORROSION FATIGUE SIZE EFFECT

[21] McAdam, D. J., "Some Factors Involved in Corrosion and Corrosion Fatigue of Metals," Proc. Am. Soc. Test. Matls., vol. 28, Part II (1928), p. 117.

effect is less for small specimens (higher endurance strength). It is also of interest that in some cases the fatigue-stress-concentration factors for large specimens approach theoretical stress-concentration factors derived from the theory of elasticity.

The foregoing results have an important bearing on the interpretation of tests of models where fatigue is involved. This statement should not be interpreted as discouraging model tests. In many cases it is impractical to test an actual structure. A model properly made indicates the weakest section, reveals the type of fracture, and if properly interpreted gives an indication of the strength of the actual structure. In Fig. 11 is shown a torsional-fatigue model[22] of a crankshaft of an airplane engine. A fatigue model[23] of a turbo-generator rotor is shown in Fig. 12. Since the stress-concentration curves (Figs. 5, 7, 9) have a tendency to flatten out, it is important to make the models as large as the testing equipment permits.

The Impact Test

Of the common tests of materials, the impact test is perhaps the least understood from a fundamental point of view. Until the impact test can be correctly interpreted qualitatively, one cannot draw definite conclusions of a quantitative nature.

A study of impact tests of tensile specimens without notches was made by Körber and Sack,[24] who concluded that the law of similarity was applicable for tests of this type. The specimens were 5 mm and 10 mm in diameter and were made of steels varying in carbon content from 0.08 to 0.86 per cent. The weight of the hammer was 37.8 kg and the drop height 2.67 m. The striking velocity was 7.25 m per sec; a velocity of half this amount did not change the result appreciably.

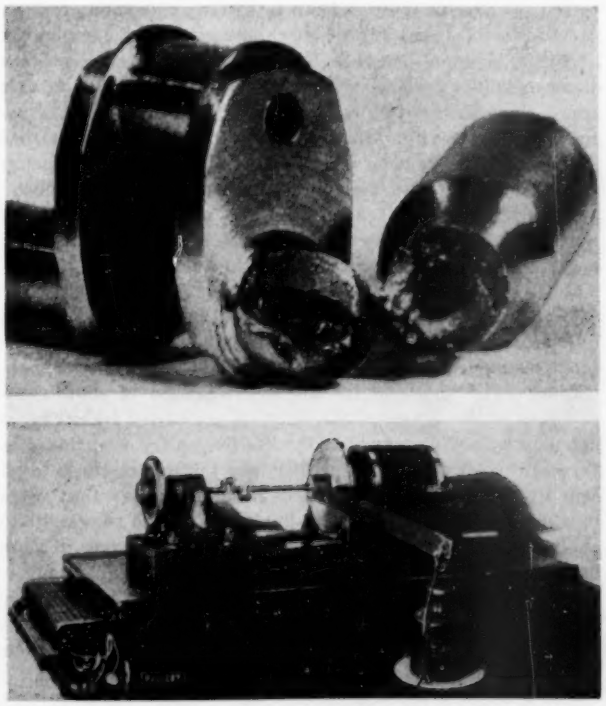


Fig. 11 Crankshaft Torsional Fatigue Model

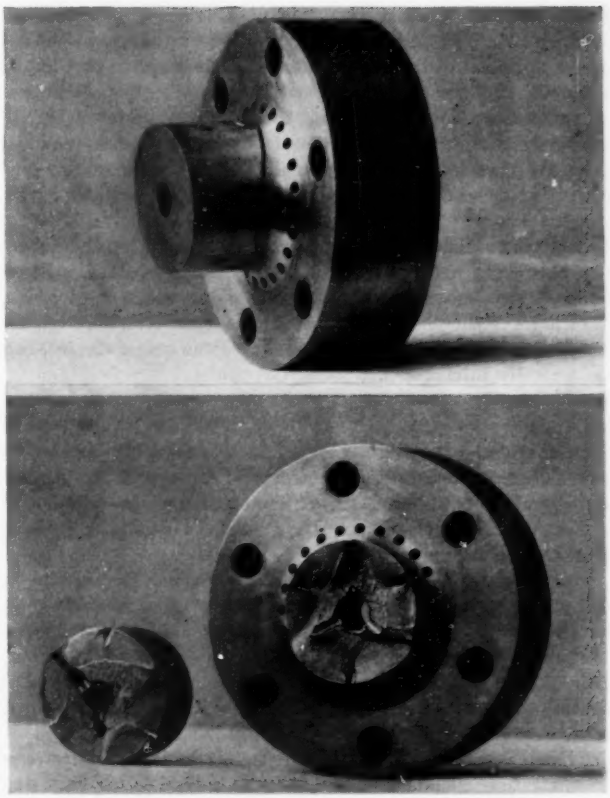


Fig. 12 Turbo-Generator Fatigue Model

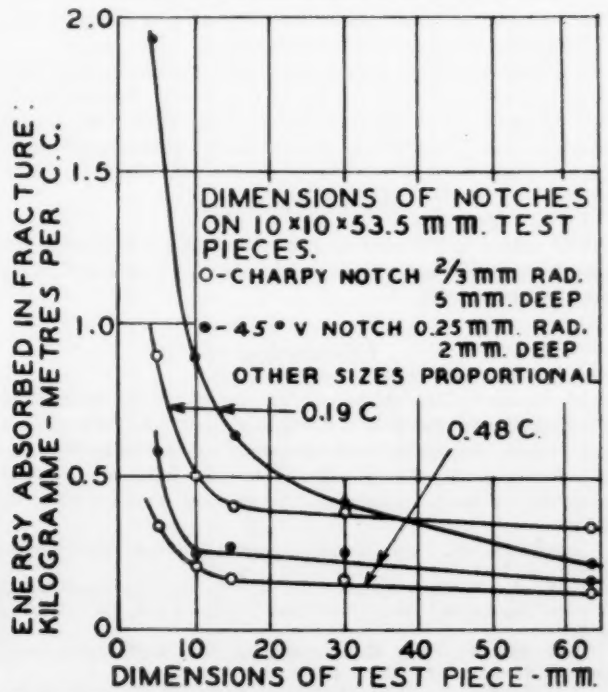


Fig. 13 Notched-Bar Tests on Similar Pieces (Stanton and Batson)

The impact tests as ordinarily performed makes use of notched-bar specimens of Charpy (simple-beam) or Izod (cantilever-beam) type. The notch introduces very marked stress concentration, which changes considerably the phenomena occurring in

[22] Matthaes, K., "Kurbelwellenbrüche und Werkstoffragen," *Luftfahrtforschung*, vol. 8 (1930), p. 91.

[23] Peterson, R. E., "Fatigue Tests of Model Turbo-Generator Rotors," *Mechanical Engineering*, vol. 53 (1931), p. 211.

[24] Körber, F., and Sack, R. H., "Vergleichende statische und dynamische Zugversuche," Mitt. Kaiser Wilhelm Inst. für Eisenforschung, vol. 4 (1922), p. 11.

the impact test. It has been shown by Stribeck,[25] Stanton and Batson,[26] Moser,[11] and others that the law of similarity does not apply to notched-bar impact tests. The curves of Fig. 13 (Stanton and Batson) illustrate the wide variation of impact value with specimens of different size. The ordinates of Fig. 13 are work per unit volume, whereas if work per unit cross-sectional area is used as ordinates (as is usually done), the curves rise with increasing size of specimen. In either case, the law of similarity is not followed.

Moser[27] concluded that the work done should be divided by the plastically deformed volume. The work of Moser, Mailänder,[28] and other investigators has done much to increase knowledge of the phenomena taking place in notched-bar impact tests, but further research work must be done before the test can be correctly interpreted.

Conclusion

In considering the data covered in this paper it seems that where stress concentration is not present the results are in most cases independent of size. An extension of the law of similarity of Barba and Kick can apparently be made to include other fields than tension testing.

However, with stress concentration present, the results of dynamic tests are quite different. Smaller specimens tend in general to give higher values. The phenomenon has not been adequately explained. It was previously suggested that different degrees of cold work due to the machining of different sized discontinuities may be an important variable affecting the results, particularly in the case of small discontinuities.

There is some evidence that materials having small stress-concentration effect in fatigue are also materials having large internal damping,[29] and vice versa. This may have some bearing on the size-effect problem, since damping capacity is a volume phenomenon, and fracture is a plane phenomenon. The relation between these phenomena varies with size.

The effect of time[30] as reflected in the speed of testing is perhaps important where stress concentration is present. Two extremely slow fatigue tests of specimens having holes were started some time ago, but a long time is required before definite conclusions can be made.

Acknowledgments

The author is indebted to Drs. Nadai and Timoshenko for their interest and suggestions and to Mr. F. Cassel for carrying out the fatigue tests.

Discussion

M. Stone.[31] The subject of the use of models in fracture determinations in the field of strength of materials is of profound importance, because our test specimens are always models, the opportunity rarely being afforded to test full-size structural elements. If we consider what the designer relies on when he analyzes a structure to withstand certain loadings, we recognize two major categories of information: (1) Elastic analysis of stresses produced by forces. (2) Interpretation of stresses in terms of failure.

The first category has received by far the major attention, the work of several decades having investigated multitudinous types of stress loadings. The second category of information remains to a large extent unexplored, the reason being that it needs must be entirely experimental, which experiments are very laborious and costly. Suffice it to recall that simple tension tests for a long time remained the basis of material application, until the phenomena of fatigue was appreciated. Then this latter test was studied thoroughly as to complete variation in stress, and gradually supplemented by superposition of variable on steady stresses. But to extend the experimental information available for such cases to actual stress conditions requires a large jump that is now bridged by resorting to analysis, based on simple elastic theory. (See "Factor of Safety and Working Stresses," by C. R. Soderberg, Trans. A.S.M.E., 1930, APM-52-2.) That this type of information will be replaced by actual experimental information as it becomes crystallized is to be expected, but now our attention is drawn to the validity of the test information itself—the translation of tests on small models to the larger actual structures.

It is evident from the paper that laws governing size effect, in all but the simplest case of tension testing, have not been definitely established. But the writer should like to emphasize the value of the paper in forcing the attention of engineers to the problem. Much work can be and has been done in the experimental investigation of failure under given stress conditions without regard to size of test specimen, and hence is of doubtful value. One particular relation is being recognized now, and that is that stress concentration factors are larger as the specimen is larger. This is in some respects disappointing, since to be certain of the results, the specimens should be as large as possible, approaching the actual size of the structure in question. However, further research will have to be done to determine how the effect increases with size and where an ostensibly constant value is obtained. In the field of weld testing, the size effect is proving itself important, and as this is an active field of investigation, the writer should like to have the author supplement his forward-looking paper with reference to welding.

Dr.-Ing. A. Nadai.[32] The various questions and points raised concerning the model-testing methods as applied to the problems of strength of materials are quite important for many reasons. The analogies for the representation of solutions of problems of the theory of elasticity are considered by engineers, not only as useful means to visualize frequently occurring solutions of certain problems, but they may serve as methods to obtain by experiment numerical information in practical cases, in which it would be difficult or impossible to find the results by analytical methods. Further attempts to work out in more detail the conditions under which mechanical similarity may be attained, either in a test or by using a model, will be of great practical interest, and particularly also the cases will have to be discussed where a given model subjected to a test cannot well represent completely the behavior of the construction element for which it was suggested.

As far as the elastic-plastic equilibrium of metallic pieces is concerned and if fracture is first excluded, a few further remarks may perhaps be in order. If two bodies of the same material and of geometrically similar shape are subjected to the action of similar systems of external loads acting solely in the surface,

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[25] Stribeck, R., "Die Kerbschlagprobe und das Ahnlichkeitsgesetz," *Z. V.D.I.*, vol. 59, Part I (1915), p. 57.

[26] Stanton, T. E., and Batson, R. G., "On the Characteristics of Notched Bar Impact Tests," Proc. Inst. C. E., vol. 211, Part I (1920), p. 67.

[27] Moser, M., "Zur Gesetzmässigkeit der Kerbschlagprobe," *Stahl u. Eisen*, vol. 42 (1922), p. 90.

[28] Mailänder, R., "Das Ahnlichkeitsgesetz bei der Kerbschlagprobe," *Kruppsche Monatshefte*, vol. 7 (1926), p. 217.

[29] von Heydekampf, G. S., "Damping Capacity of Materials," Proc. Am. Soc. Test. Matls., vol. 31, Part II (1931), p. 169.

[30] Schlechtweg, H., "Zeit und Spannung in der homogener Kontinua," *Zeit. für Physik*, vol. 72 (1931), p. 811.

[31] Engineer-in-Charge of Development, Power Engineering Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

[32] Westinghouse Research Laboratories, East Pittsburgh, Pa. Mem. A.S.M.E.

the condition of similarity of the two internal states of stress is simple. If the material was deformed in certain regions plastically, while the strains in the remaining parts of the two bodies are still purely elastic strains, the condition is simply that at corresponding points (that is, in points the coordinates of which are in the proportion $x : y : z = x' : y' : z'$) the components of stress or of unit strain must be equal in corresponding sections and directions. From this it follows that the resultant external loads acting on the two bodies must be proportional to the square of the ratio of the dimensions of the two bodies. Another very useful expression of the same fact facilitating comparison is that in the case of mechanical similarity the two pictures of the lines of principal stress must consist of two geometrically similar triple families of curves. Thus the elastic-plastic states of stress produced by compressing a set of hard, rigid balls having diameters d_1, d_2, d_3. . ., into a soft metal will be similar if the forces will be chosen according to $P_1 : P_2 : P_3 : \ldots = d_1^2 : d_2^2 : d_3^2 : \ldots$. This is the law of similarity of Barba-Kick. This rule may easily be generalized to include also the effect of speed of deformation or of rate of flow. It is known that the stresses under which a metal yields to some extent depend on the rate of flow. If rate of flow is an essential further new variable, on which the stress required to produce a given plastic strain would depend, then the necessary and sufficient condition under which again mechanical similarity of the plastic equilibria in two geometrically similar bodies, loaded similarly, would result, would be to change the forces (with time t) so that the unit strains produced by them per unit time in both bodies would be the same.

As the forces which are required to produce the same unit strains in two bodies must be in the proportion of the squares of the linear dimensions of these bodies, the test has to be conducted so that any force $P = P'(l^2/l'^2)$, where P' is the corresponding load acting on the second body and now P or P' are functions of the time t. Two similar hardness tests with a large and with a small ball have to be conducted so that if $P' = f(t)$, $P = P'(d^2/d'^2)$. If, for example, $d = 2d'$, then at equal times the load P on the large ball must be equal four times the load P acting on the small ball.

This, however, is only true in the case of comparatively slow plastic flow. If the forces increase more rapidly with time due to the acceleration of the body masses, additional inertia forces must be produced in the mass elements of the stressed bodies. This is of great importance for all questions referring to impact stresses. The case of impact of bodies needs further variables (inertia forces, mass forces), and the condition of similarity will have also to include the effect of body forces hitherto not considered.

The various observations mentioned in the report referring to fracture and particularly to fatigue strength and fracture, including also the effect of stress concentrations, are highly interesting as they seem to indicate that the conditions of similarity in fatigue tests probably depend on further mechanical or other factors not yet found or established, and it will be an interesting task in the future to search for these unknown variables or effects which seem to influence fatigue failures.

Author's Closure

Dr. Nadai mentions the time variable (rate of deformation) as a possible factor in size effect. A few comparative fatigue tests at widely different rates have not shown a marked effect to date. It is quite clear that much work remains to be done in the fatigue field before the question of size effect can be cleared up, particularly for the stress-concentration cases.

Mr. Stone suggested that some remarks be made about size effect in fatigue of welds. For welds made with bare electrodes, larger sizes gave lower endurance limits, similar to the results herein reported for specimens having holes. Since bare electrode welds contain tiny holes,[33] one might expect some similarity of results. This work is not completed as yet.

[33] Peterson, R. E., and Jennings, C. H., "Fatigue of Weld Metal," Proc. Am. Soc. Testing Matls., vol. 31 (pt. 2), p. 194 (1931).

APM-55-12

Applications of Creep Tests

By GLEASON H. MacCULLOUGH,[1] WORCESTER, MASS.

Analytical solutions of problems which involve creep phenomena and which are of practical interest are at present very limited in number. This paper discusses four specific problems for which solutions have been presented: namely, the problem of the flanged and bolted pipe joint under creep conditions, and the three problems of stress distribution and creep in thick-walled cylinders under internal pressure, in a beam subjected to pure bending, and in a solid circular shaft under torsion. These solutions will illustrate the kind of creep data which the designer desires the experimenter to furnish.



1—The Phenomenon of Creep

CREEP may be defined as the comparatively slow deformation of a material which occurs with lapse of time under the action of an externally applied stress. The exact nature of creep will not be discussed here, other than to say that it undoubtedly is due to shear. This phenomenon of creep is most pronounced at elevated temperatures. A stress which is a fraction of that required to produce failure in a few minutes as in the ordinary static test may ultimately cause failure if allowed sufficient time. Then again, while there may be stresses so small that they will not cause failure of a structural part by creep in a very long period of time, nevertheless such stresses may be harmful because of the structural distortion which they may ultimately set up. Examples of such distortion are the loosening of flanged joints caused by creep in the bolts, and the changes in clearance of turbine blading.

2—Creep Tests—The Long-Time Test

Entering into the usual creep tests of metals, there are four variables: namely, stress, deformation, time, and temperature (1).[2] Such tests may be variously classified as tension or shear (torsion) tests, as long-duration or short-duration tests, and as regards the method employed in testing. Nearly all of the tests so far made have been tensile tests, though a few torsional tests have been made by R. W. Bailey (2) and Dr. F. L. Everett (3). For a rough correlation between the results of tension and torsion tests, see Appendix No. 2.

The most valuable creep tests are those extending over months. Perhaps the most common of these tests are those where strain measurements are taken with respect to time, the stress and temperature being held constant during any one test. In this way, a family of curves like that in Fig. 1 may be obtained. From such curves (or curves where temperature is the parameter and the stress is constant), various derived curves may be obtained such as "time rate of creep vs. time," "stress vs. minimum rate of creep," "stress vs. logarithm of minimum rate of creep," "stress vs. duration to fracture," etc.

Most of the creep-time curves are characterized by an inflection point or by a linear middle portion at which a minimum rate of creep occurs. The slope of the tangent at this inflection point or along this straight portion gives the minimum rate of creep. Along this linear portion, of course, steady creep conditions prevail. In the problems of Arts. 5–8 where the effects of creep upon redistribution of stress and upon permissible deformations are considered, such a steady state of creep is assumed and plots of stress vs. minimum rate of creep consequently form the experimental basis for the solution of such problems.

Another physical property selected from creep data which is sometimes useful, is the concept of a limiting creep stress. H. J. Tapsell (4) defines this limiting creep stress as "the load per unit of the original area which will just not break the bar when allowed to remain on for a long time." Such a stress, reduced slightly by a proper factor of safety, may be used as the safe working stress in design using the ordinary formulas based on the elastic theory. This method may prove unsatisfactory, however, for although the material may not actually fail under such a limiting creep stress, there is a possibility that it will deform so much as to be unsuitable in actual practice. As to permissible tensile strain rates, K. Baumann (5) has suggested the following rates for different parts of a steam plant which seem to have gained wide acceptance:

	Inches per inch per hour
Turbine disks pressed on shafts	10^{-9}
Bolted flanges, turbine cylinders	10^{-8}
Steam piping, welded joints, boiler tubes	10^{-7}
Superheater tubes	10^{-7}

Another method of tensile creep test might be termed the "relaxation" test. This consists of a tension specimen having two turned-down portions and which is mounted between rigid supports. Provision is made for applying an initial tension to the specimen. One of the portions is surrounded by the usual electric heating element and the other portion serves as a calibration bar for measuring the force in the bar, Fig. 2. Knowing the strain in the gaged length of this second section, the corresponding stress can be found from the results of a previous calibration. So indirectly by means of these strain gages it is possible to measure the relaxation in stress due to the yielding or creep of the reduced portion within the furnace. Usually it is not convenient to measure the actual creep of the specimen within the furnace. Curves plotted from data obtained thus appear as shown in Fig. 3.

3—Short-Time or Accelerated Creep Tests

The creep tests already described have the disadvantage of requiring a long time for their performance. Much work is going on in an endeavor to develop new forms of "accelerated" creep tests which will produce in a short time the same indica-

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[1] Professor of Engineering Mechanics, Worcester Polytechnic Institute. Mem. A.S.M.E. Professor MacCullough was graduated from the Worcester Polytechnic Institute with the degree of B.S. in 1918. He also received the M.S. degree from the same institution in 1931, and in 1932 the Sc.D. degree from the University of Michigan, where he was engaged in an investigation of creep in bending. He has been successively instructor, assistant professor, and professor of mechanical engineering at the Worcester institution. He has also had outside experience with industrial concerns and in the commercial testing of materials of construction.

[2] Numbers in parentheses refer to similarly numbered items in the bibliography given at the end of the paper.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

tions regarding the fitness of a material as must occupy very long periods when tests are made by the more fundamental long-duration method.

Three of the proposals made for such accelerated tests aim to obtain a satisfactory working stress or limiting creep stress. One of the best-known methods is that devised by W. H. Hatfield (6) where he obtains a value called the "time-yield." This investigator proposed a comparatively short method of arriving at a particular creep limit, which he stated was, at high temperatures, a value comparable with the yield point at ordinary temperatures. His creep limit, or time-yield, was defined as the maximum stress which would not produce an extension exceeding 0.5 per cent

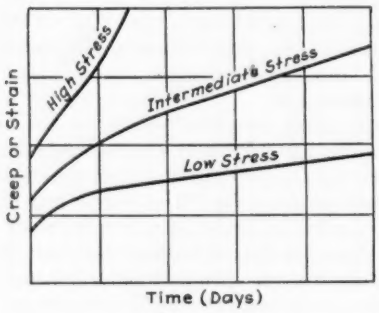


FIG. 1

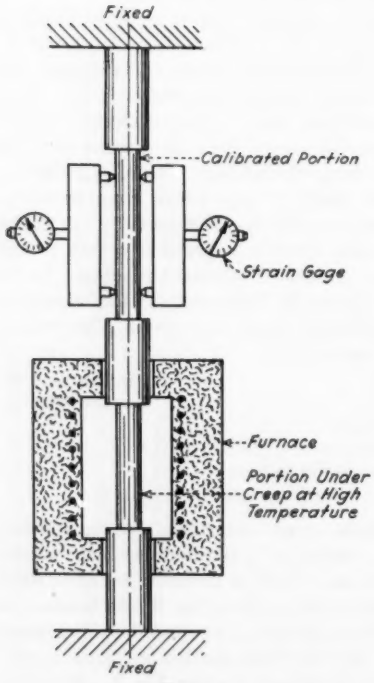


FIG. 2

in the first 24 hr, nor greater than 0.005 per cent in the next 48 hr. To determine this limit requires the making of two or three tests in the vicinity of the predicted creep limit, its actual value being found by interpolation. Such a method for the estimation of the time-yield at a particular temperature was considered to require tests occupying a total period of about one week. As a safe stress, Dr. Hatfield recommends a stress about two-thirds of this time-yield.

Another method, aiming at securing a limiting creep stress within a period of the order of 48 hr, has been devised by Messrs. W. Barr and W. E. Bardgett (7). This method makes use of the so-called "relaxation" test and consists of plotting a curve (Fig. 4) showing the relation between the decrease in stress over a 48-hr period against varying initial stresses. By extrapolation of the straight line obtained, the initial stress corresponding to a zero decrease in stress is found. This initial stress is the limiting creep stress.

For a third suggested method, the reader is referred to H. J. Tapsell's "Creep of Metals," p. 121 (1). Still another method is advocated by A. Pomp and Walter Enders (7a).

Studies have also been made looking toward reducing somewhat the time required for the usual long-time test instead of devising new tests like those already described. It is found that an initial temperature, some 50 to 100 C above the temperature for which the required result is desired, maintained through the first stages of the test and then dropped to the required temperature, shortens the time required and does not materially affect the minimum rate of creep or the limiting creep stress eventually obtained (8). Or the initial stress may be temporarily raised and later reduced to the desired value with the same results (9).

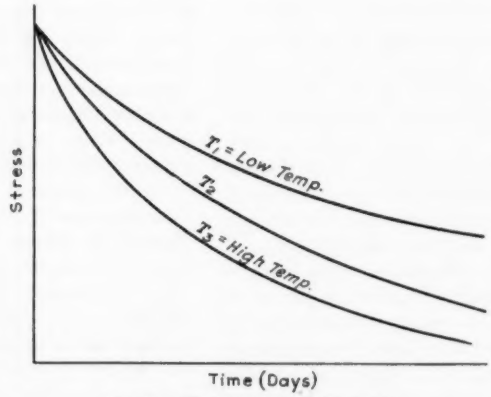


FIG. 3 "RELAXATION" CURVES

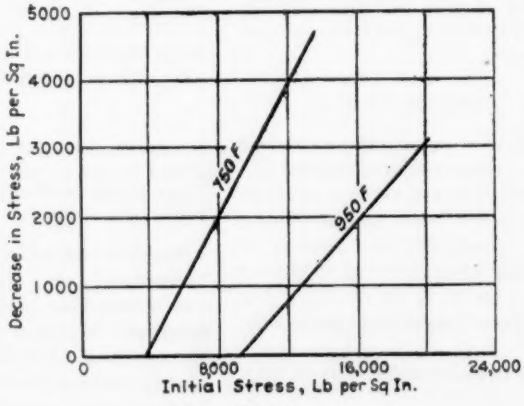


FIG. 4

4—SOME GENERAL CONSIDERATIONS AFFECTING DESIGN

In the general case of design, the most frequent problem facing engineers is that of designing a part for operation at a particular temperature for which experience exists at lower temperatures. If the material is unaltered, it is usual to base design upon existing practice by reducing the working stress in the ratio suggested by creep tests at the two temperatures.

For example, in the case of a steam power plant, considerable experience exists for an initial temperature of 750 F. In the case of design for 850 F, creep tests indicate that for the same carbon steel and the same creep rate, the stress is approximately 60 per cent of that for 750 F. The working stress for elastic design is accordingly reduced in this ratio.

This method of design takes no account of the redistribution of the stress due to creep in the cases of bending and torsion and in thick-walled cylinders. This redistribution is favorable, for it reduces the maximum intensity of stress found by the elastic theory. Neither does this practice take into direct account the permissible deformations. An elaboration of this method is discussed by Messrs. Bailey and Roberts in the article already referred to (8).

When the deformations caused by creep and their effect upon the stress distribution are taken into account, design problems acquire considerable complexity. A few problems have yielded to a rather reliable analytic treatment, among which are those dealing with parts of cylindrical shape so common in pressure vessels. In what follows, four of these solutions are presented.

5—THE PROBLEM OF THE FLANGED PIPE JOINT

This solution is essentially that presented by K. Baumann (10) in the English periodical *Engineering*.

Given a flanged joint (Fig. 5) with flanges as well as bolts subjected to creep and with the elastic flexibility of both taken into account, to find an expression for the time required for the initial stress in the bolt to fall to a given value due to creep, or, conversely, to find the stress in the bolts after an elapsed time; to show also that for any ratio

$$n = \frac{\text{flexibility of flange}}{\text{flexibility of bolt}}$$

and for any ratio

$$p = \frac{\text{creep rate of flange}}{\text{creep rate of bolt}}$$

the time required for the stress to fall from one given value to another is $q = (1 + n)/(1 + p)$ times that given by the curve of Fig. 6. This curve is a plot of the stress in the bolt against time in hours for the case of rigid flanges, $q = 1$, and is what one might call a "relaxation" curve.

The following notation will be adopted:

- L_f = flange span or distance between flanges before the bolts are tightened
- L_b = length that stressed portion of bolt would assume if relieved of load before any creep has occurred
- k_b = total bolt extension due to creep
- k_f = total diminution of flange span due to creep
- δ_b = total elastic extension of bolt
- δ_f = total elastic diminution of flange span
- σ = stress in bolt at any time
- σ_0 = initial stress in bolt before any creep has occurred
- c = creep rate of bolt material (unit strain due to creep per unit of time) taken as following the law $\sigma = a + b \log_e c$, where a and b are empirical constants, for any given temperature. For, in the case of steels at temperatures up to, say, 500 C and stresses of working magnitude, a linear relation between stress and log rate of creep expresses fairly well the results of experiment.

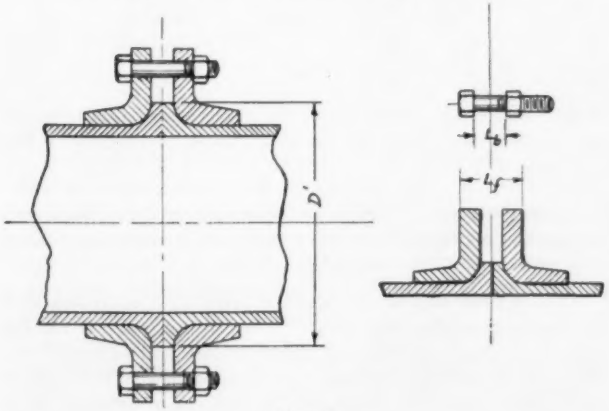


FIG. 5

After the bolt and flange are assembled and creep is taking place, the effective length of the bolt must always equal the span of the flanges. Hence

$$L_b + \delta_b + k_b = L_f - \delta_f - k_f \quad [1]$$

Let

$$s = L_f - L_b \quad [2]$$

= amount nut is screwed up with respect to the unstressed bolt
= lead × turns of nut

$$n = \frac{\text{elastic displacement of flange at bolt circle caused by any given force}}{\text{elastic stretch of bolt caused by the same force}} = \frac{\delta_f}{\delta_b} \quad [3]$$

and

$$p = \frac{\text{creep displacement of flanges at bolt circle caused by a given force}}{\text{creep stretch of bolt caused by the same force}} = \frac{k_f}{k_b} \quad [4]$$

Substituting from [2], [3], and [4] in [1],

$$s = \delta_b(1 + n) + k_b(1 + p) \quad [5]$$

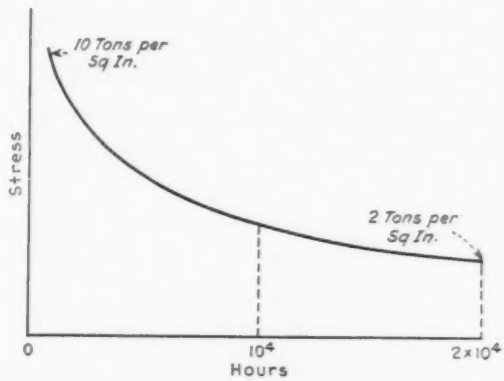


FIG. 6

No matter whether creep has occurred or not, we shall assume that the elastic properties of the bolt remain unchanged. The application of Hooke's law to the bolt, therefore, gives

$$\delta_b = \frac{\sigma L_b}{E} \quad [6]$$

where E is the modulus of elasticity of the bolt material. Substituting [6] in [5] and differentiating with respect to time,

$$0 = (1 + p)\frac{dk_b}{dt} + \frac{1 + n}{E} L_b \frac{d\sigma}{dt} \quad [7]$$

The creep rate per unit of length is

$$c = \frac{d(k_b/L_b)}{dt} = \frac{1}{L_b}\frac{dk_b}{dt} \quad [8]$$

From the assumed creep law,

$$c = e^{\frac{\sigma - a}{b}} \quad [9]$$

Eliminating c between [8] and [9] and substituting the value of dk_b/dt thus found into [7],

$$e^{\frac{\sigma - a}{b}} + \frac{q}{E}\frac{d\sigma}{dt} = 0 \quad [10]$$

where $q = (1 + n)/(1 + p)$.
The solution of the differential Equation [10] is

$$t = \frac{bq}{E}\left[e^{\frac{a - \sigma}{b}} + C\right]$$

Now at time $t = 0$, let the stress $\sigma = \sigma_0$ and hence $C = -e^{\frac{a-\sigma_0}{b}}$

$$t = \frac{bq}{E}\left[e^{\frac{a-\sigma}{b}} - e^{\frac{a-\sigma_0}{b}}\right] \quad \ldots\ldots\ldots\ldots [11]$$

Equation [11] gives the time required for the stress to fall from its initial value σ_0 to a final value σ. Since a, b, and E depend only on the bolt material and temperature and not on the dimensions, when these quantities are fixed, the time required for the stress in the bolts to fall from one given value to another is then proportional to q. It can be shown that this statement is true regardless of what the creep-stress law may be (see

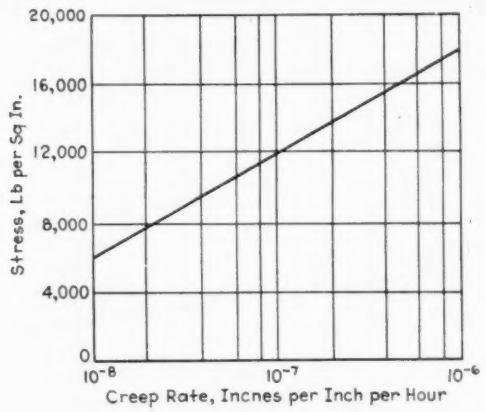


FIG. 7

Appendix No. 1). Since $q = (1 + n)/(1 + p)$, in order that the joint may remain tight for a long time, it is desirable that the flanges be very flexible, provided that this additional flexibility does not involve a proportionate increase in the creep rate of the structure.

From [11], one can obtain an expression for the stress at the end of any given time t.

$$\sigma = a - b \log_e\left[\frac{Et}{bq} + e^{\frac{a-\sigma_0}{b}}\right] \quad \ldots\ldots\ldots [12]$$

As a numerical example of the use of Equation [11], consider a 6-in. flanged pipe joint bolted together with twelve 1-in.-diam. bolts. Diameter of outside of contact surface, $D' = 8^1/_2$ in. The working steam pressure is 600 lb per sq in. For the bolt material, $E = 25{,}000{,}000$ lb per sq in., and the plot of stress against the logarithm of the minimum rate of creep is shown in Fig. 7, assumed for a temperature of 750 F.

The constants a and b in the equation of this plot $\sigma = a + b \log_e c$ are found by substituting two pairs of corresponding values of σ and c, obtained from the curve, in the above equation. Note that

$$\log_e c = 2.3026 \log_{10} c$$
$$6000 = a + 2.3026\, b \log_{10} 10^{-8}$$
$$18{,}000 = a + 2.3026\, b \log_{10} 10^{-6}$$

Solving simultaneously, $a = 54{,}000$

$$b = 2600$$

We shall make the further assumptions that for the joint, $n = 3$, $p = 1$, and that the bolts should always have a tension twice that caused by the steam pressure in order to insure tightness of joint. In calculating this tension in the bolts, it is recommended that the steam pressure be considered as acting upon an area circumscribed by the periphery of the outside of the contact surface. The area in this case is $(\pi/4)\ (8^1/_2)^2 = 56.75$ sq in., and the total steam pressure is $600 \times 56.75 = 34{,}050$ lb. The stress set up in each bolt, when doubled to insure tightness, is 7230 lb per sq in.

Since the bolts may be given an initial tension of 20,000 lb per sq in. and since the tension ought not to fall below the value of $\sigma = 7230$ lb per sq in., the time t necessary for the bolt stress to drop to the latter value is found from Equation [11]. $q = (1 + n)/(1 + p) = 2$.

$$t = \frac{2600 \times 2}{25{,}000{,}000}\left[e^{\frac{54{,}000 - 7230}{2600}} - e^{\frac{54{,}000 - 20{,}000}{2600}}\right]$$
$$= 13{,}600 \text{ hr} = 81 \text{ weeks}.$$

If this is too short a period for the life of the joint, a different bolt material may be used having greater resistance to creep, or the flexibility of the flanges may be increased, thus increasing q.

As an example of the use of Equation [12], suppose one wishes to know the stress in the bolts of the above joint at the end of 2 years or 17,520 hr. Substituting in [12], there results

$$\sigma = 54{,}000 - 2600(2.3026) \log_{10}\left[\frac{25{,}000{,}000 \times 17{,}520}{6070 \times 2} + e^{\frac{54{,}000 - 20{,}000}{2600}}\right] = 8720 \text{ lb per sq in.}$$

It is to be noted in these calculations that increasing the initial tension within safe limits will have but little effect on the life

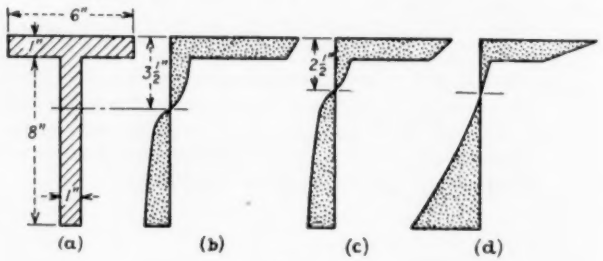


FIG. 8

of the joint. For a more complete discussion of this problem, the reader is referred to the previously mentioned article by Mr. Baumann (5).

6—The Problem of Stress Distribution and Resisting Moment for a Beam Subjected to Pure Bending and Creep Conditions (11)

This solution utilizes the results of creep tests in tension with the assumption that the rates of creep in compression are the same as in tension under the same stresses and temperatures. The general method of attack is this: Knowing the creep rate at the outside fiber of the beam and the manner of deformation of the cross-sections, we can find the creep rates for various fibers within the beam. If it is considered that the stresses producing these creep rates are equal to the stresses that would cause the same minimum rate of creep in tension or compression, then from a plot of "stress vs. minimum rate of creep in tension," the corresponding stresses in bending can be determined. Knowing the stress distribution for a few points in the cross-section of the beam, it is an easy matter to compute the moment of resistance or the bending moment acting on the beam.

The method will be illustrated by considering a numerical problem.

The T-section shown in Fig. 8 will be chosen since the method

of solution is more general than in the case of the simpler rectangular cross-section. Let us endeavor to calculate the bending moment which may act on the beam and still not cause an increase in deflection due to creep at the center of a 48-in. length greater than 0.01 in. in two years, for an assumed temperature of 750 F. A Cr-Ni-Mo alloy steel will be used, for which the plot of stress vs. logarithm of the minimum rate of creep is shown in Fig. 9.

From the fact that the sag, dy, due to creep in two years may be 0.01 in., one can find the rate of angular creep $d\theta/dt$ between plane sections 1 in. apart from the relation

$$\frac{d\theta}{dt} = \frac{8}{l^2}\frac{dy}{dt}$$

which is valid for flat circular arcs.

$$\frac{d\theta}{dt} = \frac{8}{(48)^2} \times \frac{0.01}{2(365)(24)} = 1.98 \times 10^{-9} \text{ radians per hour}$$

Let us start by assuming the neutral axis, or axis of rotation for the plane sections, to pass through the centroid of the cross-section which is $3^1/_2$ in. below the top. Then the rate of creep of any fiber equals the product of its distance from the assumed neutral axis and $d\theta/dt$. The corresponding stress causing this rate of creep can be read from the curve of Fig. 9. The distribution of stress thus found is given in the first three columns of Table 1. This stress distribution, when multiplied by the width of the cross-section at each point, is shown plotted in Fig. 8(b).

TABLE 1

Distance below top	Neutral Axis $3^1/_2$" below top — Rate of creep	Corresponding stress	Neutral Axis $2^1/_2$" below top — Rate of creep	Corresponding stress	Stress × width × xy
0	6.93×10^{-6}	18,000	4.95×10^{-9}	16,000	240,000
$^1/_2$	...	....	3.96	15,000	180,000
1	4.95	16,000	2.97	13,300	120,000
1	...	....	...	13,300	20,000
2	2.97	13,300	0.99	9,000	4,500
$2^1/_2$	...	....	0.0	0	0
3	0.99	9,000	0.99	9,000	4,500
$3^1/_2$	0.0	0	...	....	.....
4	0.99	9,000	2.97	13,300	20,000
5	2.97	13,300	4.95	16,000	40,000
6	4.95	16,000	6.93	18,000	63,000
7	6.93	18,000	8.91	19,300	86,800
8	8.91	19,300	10.9	20,300	112,000
9	10.9	20,300	12.9	21,300	138,000

If the neutral axis has been correctly chosen, the total force above the neutral axis must equal that below. This is equivalent to equating the areas above and below the neutral axis in Fig. 8(b). A rough determination shows these two areas not to be equal but rather in the ratio of about 52 to 31. It is readily seen that raising the neutral axis will reduce the upper area and increase the lower one.

Assuming the neutral axis now to be $2^1/_2$ in. below the top, a calculation similar to the above is repeated with the results shown in the 4th and 5th columns of Table 1 and the plot is shown in Fig. 8(c). The area of the portion above now works out to be about 101,500 lb and that below, 104,500 lb. This indicates that the neutral axis is but little below the $2^1/_2$-in. position assumed, and we shall consider this as satisfactory.

To calculate the moment of resistance, let y be the distance from the neutral axis to any fiber where the stress is σ and where the width of the section is b. There the total moment is expressed by

$$M = \int(\sigma b dy)(y) = \int(\sigma b y)dy$$

Therefore if the horizontal intercepts σb in Fig. 8(c) be multiplied by their respective distances from the neutral axis and a new curve plotted as in Fig. 8(d), the total shaded area represents the moment of resistance. This area gives the required moment and is found to be M = 586,000 in-lb.

If the resisting moment for this beam is figured by the usual elastic theory with the same maximum fiber stress of 21,300 lb per sq in., it is found to be but 330,000 in-lb. The effect of redistribution of stress due to creep is plainly evident. Or for the same moment of 586,000 in-lb as with creep, a maximum stress of 38,000 lb per sq in. would be necessary if the stresses were distributed according to the elastic linear law.

For a beam symmetrical with respect to the middle horizontal

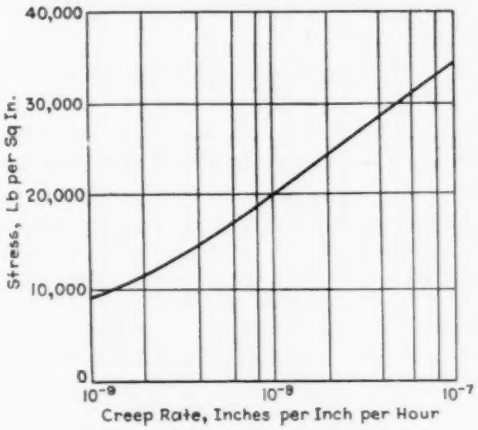


FIG. 9

axis and with like creep characteristics in tension and compression, the procedure is the same excepting that the neutral axis is known to lie at the center and need not be determined by trial. This simplifies the calculations for rectangular or circular cross-sections and for I-beams.

While the method illustrated is strictly applicable only to the case of pure bending by couples applied at the ends, it is felt that the deflection can be considered as due solely to tensile and compressive strains in the longitudinal fibers even when bending is produced by transverse loads, and the method can be used with but slight error as the basis for determining deflections. The limitation must be imposed that the ratio of the length of the beam to its depth is such that the shearing stresses are not greater than one-fourth or one-third of the flexure stresses. For the case of transverse loads, a suggested method of attack consists of subdividing the length of the beam into shorter lengths, and assuming each length to be in pure bending with a moment acting which is equal to the average moment in that length. Each of these lengths will be bent approximately into a circular arc, and the various lengths can then be "pieced together" to ascertain the total deflection at the center of the beam.

7—The Problem of Stress Distribution and Creep in Solid Circular Shafts (12)

But one example will be worked in this connection: that where the torsional stress τ is related to the minimum creep rate in shear ϕ by the relation $\tau = a + b \log_e \phi$ for any particular temperature, and where a and b are empirical constants. Other cases can be worked out similarly, either for hollow circular shafts or where different stress-creep-rate relations than the above exist.

These solutions, however, are based on a tentative hypothesis which really needs further verification experimentally. This hypothesis is stated in the following quotation from R. W. Bailey:

> The author's tests upon thin tubes in torsion indicate that a solid cylindrical test piece should behave as if composed of a number of

elementary concentric tubes undergoing rates of creep in shear proportional to their radii.

In other words, the assumption to be made is analogous to that in bending: that radial lines straight before angular creep remain straight after angular creep. Hence the analytical method to be employed in this solution is also applicable to problems in bending based on a similar assumption regarding cross-sections.

By virtue of this assumption, the shear creep rate ϕ at any point of the solid shaft of Fig. 10 is related to the shear creep rate ϕ_0 at the outside by the relation $\phi = (r/r_0)\phi_0$.

The torsional moment is given by

$$M_t = \int_0^{r_0} \tau r dA = \int_0^{r_0} (a + b \log_e \phi) r \times 2\pi r dr$$

Substituting $\phi = (r/r_0)\phi_0$ into the above and noting that $a + b \log_e \phi_0 = \tau_0$ (τ_0 being the outside shear stress), one gets

$$M_t = 2\pi \int_0^{r_0} (\tau_0 + b \log_e r - b \log_e r_0) r^2 dr$$

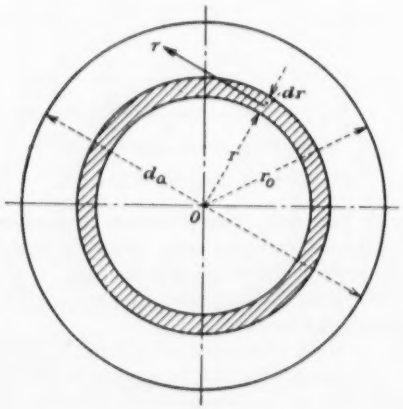


FIG. 10

Upon integration, this becomes

$$M_t = \frac{\pi d_0^3}{12}\left(\tau_0 - \frac{b}{3}\right) \quad \text{.....[I]}$$

where d_0 is the outside diameter. Knowing the external moment, τ_0 can be calculated from the above.

The stress distribution for points not near the center is

$$\tau = a + b \log_e \phi$$

$$= a + b \log_e \left(\frac{r}{r_0}\phi_0\right)$$

$$= a + b \log_e \phi_0 + b \log_e \frac{r}{r_0}$$

$$= \tau_0 - b \log_e \frac{r_0}{r} \quad \text{.....[II]}$$

The creep rate in shear ϕ_0 at the outer surface can be obtained from

$$\tau_0 = a + b \log_e \phi_0$$

or

$$\phi_0 = e^{\frac{\tau_0 - a}{b}}$$

The relation of the angular creep rate α to the shear creep rate ϕ is expressed by $\phi = r\alpha$.

Hence the rate of twist in the shaft is $\alpha = 2\phi_0/d_0$, α being in radians per hour per unit length of the shaft.

As an example of the use of the foregoing equations, let us find the moment or torque which will cause a twist due to creep of 1 deg in one year for a 1-ft length of 10-in.-diameter shaft made of 0.34 per cent carbon steel. The temperature is 400 C. The relation for this material and temperature between stress and minimum rate of creep is expressed by

$$t = 42{,}500 + 2420 \log_e \phi$$

in which $a = 42{,}500$ and $b = 2420$.

The first step is to find the shear creep rate ϕ_0 at the outside in radians per inch per hour, using the relation

$$\phi_0 = \frac{d_0 \alpha}{2}$$

$$= \frac{(10)(\pi/180)}{2(365 \times 24)(12)} = 0.83 \times 10^{-6}$$

The stress producing this creep ϕ_0 is found from the creep law above (or directly from the experimental curve) to be $t_0 = 42{,}500 + 2420(2.303) \log_{10} 0.83 \times 10^{-6} = 8600$ lb per sq in. Substituting this value in [I] gives the torque

$$M_t = \frac{\pi(10)^3}{12}\left(8600 - \frac{2420}{3}\right) = 2{,}040{,}000 \text{ in-lb}$$

Based on elastic theory and with the same maximum stress of 8600 lb per sq in., the torque is found to be 1,690,000 in-lb. The increase with creep is due to the different distribution of stress which is defined by Equation [II].

For the case where the creep law is expressible by $\log \tau = a + b \log_e \phi$, which is more nearly valid for low stresses and low creep rates than the one used above, see R. W. Bailey's paper already referred to (2).

8—The Problem of Stress Distribution and Creep in Thick-Walled Cylinders Under Internal Pressure

The solution of this problem is due to R. W. Bailey (13). It may be a little surprising that the experimental data required for its solution concern creep rates in torsion rather than in tension or compression. To show how these quantities are introduced, a portion of Mr. Bailey's analysis is summarized in Appendixes Nos. 2 and 3.

The following solution is restricted to the case where the creep rate has become constant and where the relation between the shearing stress τ and the minimum rate of creep in shear ϕ is

$$\tau = a + b \log_e \phi = a + 2.303b \log_{10} \phi$$

For cases where the creep law is given by $\log_e \tau = a + b \log_e \phi$ and where there is heat transmission through the wall, see Mr. Bailey's paper.

Let

r_1 = inside radius of cylinder
r_2 = outside radius of cylinder
r = radius to any point in cylinder wall
p = intensity of internal fluid pressure
$\sigma_r, \sigma_\theta, \sigma_z$ = normal stresses in the radial, tangential, and axial directions, respectively, at any point in the wall
b = constant appearing in the assumed creep law.

The magnitudes of the three normal stresses are given by

$$\sigma_r = -p\frac{\log_e \frac{r_2}{r}}{\log_e \frac{r_2}{r_1}} - 2b\log_e\frac{r}{r_1}\log_e\frac{r}{r_2} \quad \text{.....[IV]}$$

$$\sigma_\theta = p\frac{1-\log_e\frac{r_2}{r}}{\log_e\frac{r_2}{r_1}} - 2b\log_e\frac{r}{r_1}\log_e\frac{r}{r_2} + 4b\log_e\frac{r_1}{r} + 2b\log_e\frac{r_2}{r_1}$$

.....[V]

$$\sigma_s = \frac{p}{2}\frac{1-2\log_e\frac{r_2}{r}}{\log_e\frac{r_2}{r_1}} - 2b\log_e\frac{r}{r_1}\log_e\frac{r}{r_2} + 2b\log_e\frac{r_1}{r} + b\log_e\frac{r_2}{r_1}$$

.....[VI]

Since the creep rate is related to $\sigma_\theta - \sigma_r$, we shall deduce now the value of this useful term.

$$\sigma_\theta - \sigma_r = \frac{p}{\log_e\frac{r_2}{r_1}} + 4b\log_e\frac{r_1}{r} + 2b\log_e\frac{r_2}{r_1} \quad \text{...[VII]}$$

If, from the results of torsional creep tests, b is known, then Equations [IV], [V], and [VI] will enable one to calculate the stress distribution throughout the cylinder wall. It is significant that the stresses and creep rates set up depend only on the ratios r_2/r_1, r/r_1, etc. and not on the absolute values of r, r_1, etc.

A study of the stress distribution given by the formulas mentioned shows that the distribution of tangential stress under these creep conditions is relatively uniform across the thickness, which is in contrast with that given by the Lamé formula for purely elastic conditions. These formulas, by the way, can be used for thin-walled tubes also.

If it is desired to find the diametral creep rate of the outside diameter, Equation [VII] with r set equal to r_2 gives the value of $\sigma_\theta - \sigma_r$ at the outside. This can be substituted back into Equation [a] (Appendix No. 3) to get the shear creep rate or its equivalent $2c$, twice the diametral creep rate; that is,

$$\log_e \phi = \log_e 2c = \frac{\sigma_\theta - \sigma_r}{2b} - \frac{a}{b}$$

The inverse of this problem is of practical interest; namely, given the permissible rate of diametral creep, to determine the ratio of r_2/r_1. We have seen that the system of stress represented in Fig. 17 (Appendix No. 3) is equivalent to a pure shear with the shearing stress $\tau = (\sigma_\theta - \sigma_r)/2$. For the outside of the tube wall where $r = r_2$, [VII] reduces to

$$2\tau_2 = \frac{p}{\log_e\frac{r_2}{r_1}} - 2b\log_e\frac{r_2}{r_1}$$

where $2\tau_2$ is the value of $\sigma_\theta - \sigma_r$ at the outside.

This is a quadratic equation in terms of $\log_e (r_2/r_1)$ and its solution is

$$\log_e\frac{r_2}{r_1} = \sqrt{\frac{p}{2b} + \left(\frac{\tau_2}{2b}\right)^2} - \frac{\tau_2}{2b} \quad \text{.....[VIII]}$$

As an application, consider the case of a superheater tube, 2 in. in inside diameter, under a pressure of 1500 lb per sq in. so that the diametral creep rate shall not exceed 10^{-7} in. per in. per hr. The steel used is 0.34 per cent C steel and the temperature is 400 C. The plot of shear stress τ against the minimum rate of creep in shear for this material and temperature is expressed by the relation $\tau = a + b\log_e \phi$, where $a = 42{,}500$ and $b = 2420$.

τ_2 [or $(\sigma_\theta - \sigma_r)/2$ at the outside] for a creep rate $\phi = 2c = 2 \times 10^{-7}$ is

$$\tau_2 = a + b\log_e\phi = 42{,}500 + 2.303(2420)\log_{10}(2\times10^{-7}) = 5200 \text{ lb per sq in.}$$

Substituting in Equation [VIII],

$$\log_e\frac{r_2}{r_1} = \sqrt{\frac{1500}{2(2420)} + \left(\frac{5200}{4840}\right)^2} - \frac{5200}{4840}$$

$$2.303\log_{10}\frac{r_2}{r_1} = 0.136$$

$$\frac{r_2}{r_1} = 1.146$$

The outside diameter should therefore be 2.29 in. It is to be noted that this wall thickness of 0.146 in., based solely on creep characteristics, is less than one-half of the wall thickness of 0.312 in. recommended by the A.S.T.M. tentative specifications[1] for 1350 lb pressure, and therefore the A.S.T.M. recommended pipe will have much more creep resistance than specified in our problem.

Appendix No. 1

SHOWING THAT THE TIME REQUIRED FOR THE STRESS IN THE BOLTS OF A FLANGED JOINT TO FALL FROM ONE GIVEN VALUE TO ANOTHER IS PROPORTIONAL TO q, REGARDLESS OF WHAT THE CREEP LAW MAY BE

The various algebraic quantities are as defined in Art. 5, with the addition that

t_r = time for the stress in a bolt gripping perfectly rigid flanges to fall from one value to another, and

t = time taken for the stress in the bolt to fall through the same range when it grips flanges whose flexibility is n times, and whose linear rate of creep is p times, that of the bolts.

Let us first consider a case where the fall of stress $d\sigma$ is so small that the creep rate c for this interval dt_r may be considered as constant.

In the case of rigid flanges and elastic bolts, from Hooke's law,

$$d\sigma = \frac{E}{L_b}k_b = \frac{E}{L_b}c\,dt_r \quad \text{.....[a]}$$

since the change of stress in the bolt must cause a shortening thereof which exactly compensates for the lengthening of the bolt due to creep under tensile stress.

In the case of flexible flanges and an elastic bolt, the lengthening of the bolt relative to the flanges due to creep is nullified by the relative shortening due to fall of stress. The relative lengthening of the bolt due to creep is

$$k = k_b + k_f = (1+p)k_b$$

since $k_f = pk_b$. The relative shortening of the bolt due to fall of stress is

$$\delta = \delta_b + \delta_f = (1+n)\delta_b$$

since $\delta_f = n\delta_b$. Also

$$d\sigma = \frac{E\delta_b}{L_b} = \frac{E}{L_b}\frac{\delta}{1+n}$$

[1] See "Marks' Handbook," Third Edition, p. 1016.

elementary concentric tubes undergoing rates of creep in shear proportional to their radii.

In other words, the assumption to be made is analogous to that in bending: that radial lines straight before angular creep remain straight after angular creep. Hence the analytical method to be employed in this solution is also applicable to problems in bending based on a similar assumption regarding cross-sections.

By virtue of this assumption, the shear creep rate ϕ at any point of the solid shaft of Fig. 10 is related to the shear creep rate ϕ_0 at the outside by the relation $\phi = (r/r_0)\phi_0$.

The torsional moment is given by

$$M_t = \int_0^{r_0} \tau r dA = \int_0^{r_0} (a + b \log_e \phi) r \times 2\pi r dr$$

Substituting $\phi = (r/r_0)\phi_0$ into the above and noting that $a + b \log_e \phi_0 = \tau_0$ (τ_0 being the outside shear stress), one gets

$$M_t = 2\pi \int_0^{r_0} (\tau_0 + b \log_e r - b \log_e r_0) r^2 dr$$

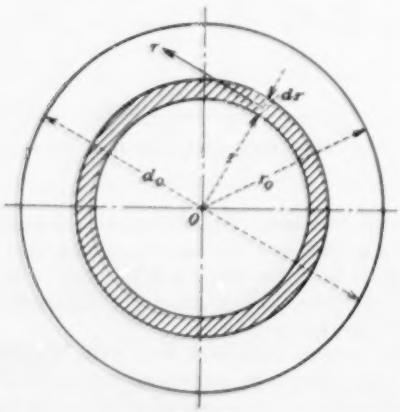


FIG. 10

Upon integration, this becomes

$$M_t = \frac{\pi d_0^3}{12}\left(\tau_0 - \frac{b}{3}\right) \quad \text{[I]}$$

where d_0 is the outside diameter. Knowing the external moment, τ_0 can be calculated from the above.

The stress distribution for points not near the center is

$$\tau = a + b \log_e \phi$$

$$= a + b \log_e \left(\frac{r}{r_0}\phi_0\right)$$

$$= a + b \log_e \phi_0 + b \log_e \frac{r}{r_0}$$

$$= \tau_0 - b \log_e \frac{r_0}{r} \quad \text{[II]}$$

The creep rate in shear ϕ_0 at the outer surface can be obtained from

$$\tau_0 = a + b \log_e \phi_0$$

or

$$\phi_0 = e^{\frac{\tau_0 - a}{b}}$$

The relation of the angular creep rate α to the shear creep rate ϕ is expressed by $\phi = r\alpha$.

Hence the rate of twist in the shaft is $\alpha = 2\phi_0/d_0$, α being in radians per hour per unit length of the shaft.

As an example of the use of the foregoing equations, let us find the moment or torque which will cause a twist due to creep of 1 deg in one year for a 1-ft length of 10-in.-diameter shaft made of 0.34 per cent carbon steel. The temperature is 400 C. The relation for this material and temperature between stress and minimum rate of creep is expressed by

$$t = 42{,}500 + 2420 \log_e \phi$$

in which $a = 42{,}500$ and $b = 2420$.

The first step is to find the shear creep rate ϕ_0 at the outside in radians per inch per hour, using the relation

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$$= \frac{(10)(\pi/180)}{2(365 \times 24)(12)} = 0.83 \times 10^{-6}$$

The stress producing this creep ϕ_0 is found from the creep law above (or directly from the experimental curve) to be $t_0 = 42{,}500 + 2420(2.303) \log_{10} 0.83 \times 10^{-6} = 8600$ lb per sq in. Substituting this value in [I] gives the torque

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Since the creep rate is related to $\sigma_\theta - \sigma_r$, we shall deduce now the value of this useful term.

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If, from the results of torsional creep tests, b is known, then Equations [IV], [V], and [VI] will enable one to calculate the stress distribution throughout the cylinder wall. It is significant that the stresses and creep rates set up depend only on the ratios r_2/r_1, r/r_1, etc. and not on the absolute values of r, r_1, etc.

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where $2r_2$ is the value of $\sigma_\theta - \sigma_r$ at the outside.

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$$\log_e\frac{r_2}{r_1} = \sqrt{\frac{p}{2b} + \left(\frac{r_2}{2b}\right)^2} - \frac{r_2}{2b} \quad \text{.....[VIII]}$$

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In the case of rigid flanges and elastic bolts, from Hooke's law,

$$d\sigma = \frac{E}{L_b}k_b = \frac{E}{L_b}cdt_r \quad \text{.....[a]}$$

since the change of stress in the bolt must cause a shortening thereof which exactly compensates for the lengthening of the bolt due to creep under tensile stress.

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since $k_f = pk_b$. The relative shortening of the bolt due to fall of stress is

$$\delta' = \delta_b + \delta_f = (1 + n)\delta_b$$

since $\delta_f = n\delta_b$. Also

$$d\sigma = \frac{E\delta_b}{L_b} = \frac{E}{L_b}\frac{\delta}{1 + n}$$

[1] See "Marks' Handbook," Third Edition, p. 1016.

Now $\delta = k = (1 + p)k_b$, and hence

$$d\sigma = \frac{E}{L_b}\frac{1+p}{1+n}k_b = \frac{E}{L_b}\frac{1+p}{1+n}cdt \ldots\ldots\ldots\ldots [b]$$

Equating $d\sigma$ between [a] and, [b] there results

$$dt = \frac{1+n}{1+p}dt_r = qdt_r$$

For any number of these intervals, $\Sigma dt = q\Sigma dt_r$ or

$$t = qt_r$$

which was to be proved. It will be observed that no assumptions have been made concerning the creep law.

If n or p is determined either experimentally or analytically so that q is known, and if a plot or data are given showing the relation between the stress in the bolts and the time for rigid flanges, the time required for the bolt stress in the flexible flange to drop a certain amount is q times that found for rigid flanges.

Appendix No. 2

RELATIONS BETWEEN SHEAR CREEP, TORSIONAL CREEP, AND TENSILE CREEP. APPROXIMATIONS FOR SHEAR-CREEP DATA OBTAINED FROM TENSILE-CREEP TESTS

Let

- r = mean radius of tube of Fig. 11 in inches
- α = angular creep rate per unit length of tube in radians per inch per hour
- ϕ = creep rate in shear in radians per hour
- c = circumferential and also diametral creep rate in inches per inch per hour. That these two creep rates are equal is evident because the circumference is proportional to the diameter. That this diagonal creep in shear is the circumferential creep of the tube under internal pressure, see R. W. Bailey's article, reference 12.

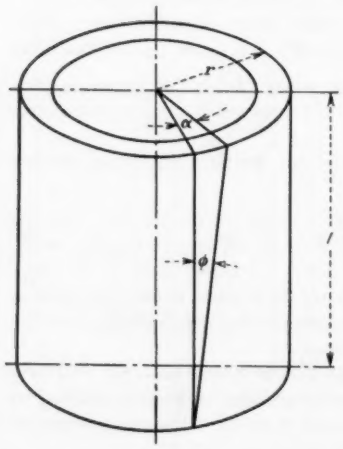


FIG. 11

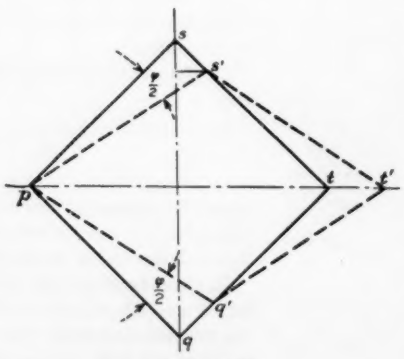


FIG. 12

The relation between α and ϕ is self-evident from Fig. 11.

$$\phi = r\alpha$$

To get the tensile-creep rate c, find how much point t (Fig. 12) moves for a shear strain θ. From the geometry of Fig. 12, the unit tensile strain over the distance pt is

$$\frac{a\theta \cos 45^\circ}{2a \cos 45^\circ} = \frac{\theta}{2}$$

where a = length of side of cube. Introducing the time element

$$c = \frac{\phi}{2}$$

Attempts have been made to connect up the experimental creep rates found in tension and torsion tests so that, from the results of tensile-creep tests, the creep rates in shear may be predicted. Messrs. Bailey, Dickenson (14), and others suggest the following procedure as giving roughly approximate results.

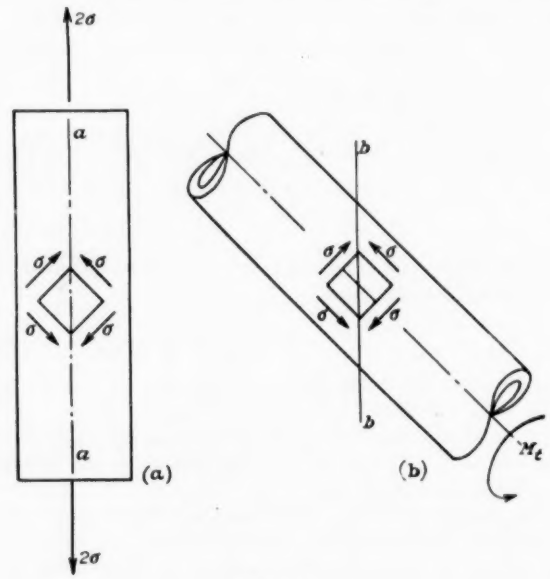


FIG. 13

According to the theory of combined stresses, a tensile stress of intensity 2σ acting on a tension specimen [Fig. 13(a)] induces shearing stresses of intensity σ on the faces of the element shown whose faces make 45 deg with the axis of the specimen. There are also normal stresses set up but, as pointed out later in Appendix No. 3, these have no effect upon the rate of creep. Let there also be a torsion specimen so stressed that the element shown, Fig. 13(b), is in pure shear with shearing stresses of intensity σ.

We have seen that the creep C' along the diagonal bb is $C' = \phi/2$. Also let the creep rate along aa be denoted by c. Now Mr. Bailey has found experimentally that c = 2 to 3 times C' = 1 to 1½ times ϕ, or, roughly $c = \phi$. Likewise a tensile stress 2σ sets up shear-stress components of magnitude σ as shown. This suggests, then, that a plot of shearing stress versus rate of creep may be obtained from tensile-creep results by halving the tensile stress to get the shear stress and by taking the value of ϕ for creep in shear equal to c as found for tension.

Appendix No. 3

MATHEMATICAL SOLUTION FOR THE PROBLEM OF STRESS DISTRIBUTION AND CREEP IN THICK-WALLED CYLINDERS UNDER INTERNAL PRESSURE

Some careful experimental work carried out by Mr. Bailey on thin-walled steel tubes and lead pipes has led to the following

conclusions which form the bases for his solution of the above problem. Since he detected no creep in the axial direction in the case of tubes undergoing creep due to torsion, he concludes that the normal stress acting on any plane has negligible influence upon the creep due to shear at that plane. Consequently, a hydrostatic system of tensile or compressive stresses may be superimposed upon any system of stress without altering the creep due to the latter. This artifice is extremely useful in converting a three-dimensional system of stress into a two-dimensional one equivalent so far as creep is concerned.

By means of this artifice, it is shown that the creep of a thin-walled tube under torsion is equivalent to that of a tube under internal fluid pressure. The application of this theory, as well as experiment, shows that there is also no axial creep in such

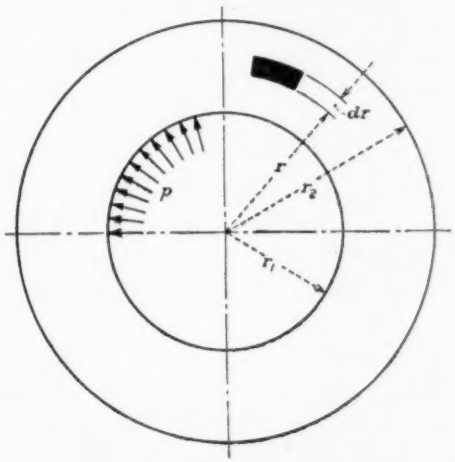


FIG. 14

thin-walled tubes subjected to internal pressure only. Interpreted in terms of stress, an absence of axial creep means that the stress condition set up is like that in thin-walled tubes under internal pressure: namely, that the circumferential stress is double the axial stress.

With the above conclusions and the relations of Appendix No. 2 at our disposal, we can now proceed to the main *problem:* to find the stress distribution and creep in thick-walled cylinders under internal fluid pressure and at a uniform temperature throughout.

Given the thick cylinder with inside radius r_1 and outside radius r_2 subjected to an internal fluid pressure of uniform intensity p as shown in Fig. 14.

Let Fig. 15 represent a curved element cut from the wall of the cylinder and located a variable distance r from the origin at the center of the cylinder. Let dr be the thickness of the element, $d\theta$ the included angle between its radial faces, and let its length in the axial direction be unity.

Let this element be acted upon by three sets of tensile or normal stress, σ_r, σ_θ, σ_z, as shown. Because of symmetry, there can be no shear stress on the surfaces of the element which are normal to σ_z and σ_θ. The absence of shear stresses on the two cylindrical surfaces is discussed later.

In order to solve for the three unknowns, σ_r, σ_θ, and σ_z, three condition equations must be set up. These conditions are:

1 There must be equilibrium of forces in the radial direction

2 There is no axial creep of the thick cylinder found experimentally

3 There is no volume change in creep phenomenon.

Several assumptions are being made either explicitly or implicitly:

1 That the relation between minimum rate of creep in shear, ϕ, and the shear stress, τ, is

$$\tau = a + b \log_e \phi$$

(A solution is also obtained by Mr. Bailey (15) on the assumption that $\log_e \tau = a' + b' \log_e \phi$.)

2 That the artifice can be employed that the addition of a hydrostatic stress does not influence creep.

3 That the stress components σ_θ and σ_z are independent of θ and z. This reduces the number of equations of equilibrium to one, that in the radial direction.

4 That plane transverse sections remain plane and parallel.

5 It is further assumed in this problem that the creep rate has become steady and has attained its minimum value.

EQUATION I. The condition for radial equilibrium of forces is

$$(\sigma_r + d\sigma_r)(r + dr)\, d\theta - \sigma_r r d\theta - 2\left(\sigma_\theta dr \cdot 1 \sin \frac{d\theta}{2}\right) = 0$$

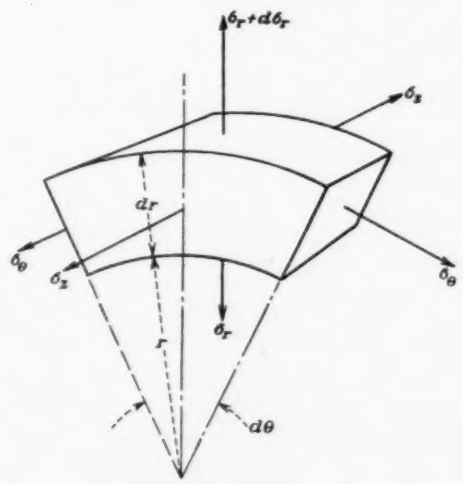


FIG. 15

Neglecting products of differentials and noting that $\sin \frac{d\theta}{2} = \frac{d\theta}{2}$, approximately, for small angles, the above equation reduces to

$$\sigma_r - \sigma_\theta + r \frac{d\sigma_r}{dr} = 0 \quad \text{..... [I]}$$

EQUATION II is based on the condition that there is found by experiment to be no axial creep in the thick cylinder. Let us first add a hydrostatic compressive stress of arbitrary magnitude $-\sigma_r$ to all faces by virtue of assumption 2; then the resulting system of stress is that pictured in Fig. 16.

On the basis of assumption 4, we are justified in dividing the thick tube up into thin tubes, the axial deformation of every one being the same, and also equal to zero, with no shear forces between the cylindrical surfaces. We can therefore imagine that Fig. 16 represents an element from a thin-walled tube subjected to an internal pressure $d\sigma_r$. The condition that there be no axial deformation or creep in this or any thin-walled tube is that the circumferential stress be twice the axial stress. Consequently,

$$\sigma_\theta - \sigma_r = 2(\sigma_z - \sigma_r) \quad \text{..... [II]}$$

EQUATION III is based on the condition that there be no volume change. Since there is no axial deformation of the cylinder,

the cross-sectional area included between any two circles must remain constant before and after creep.

Let

c_1 = circumferential or diametral creep rate at inside of cylinder in inches per inch per hour
c_2 = same at outside of cylinder
c = same at any radius r.

Between any two circles, the area after creep = area before creep.

$$\pi[(1 + c)r]^2 - \pi[(1 + c_1)r_1]^2 = \pi r^2 - \pi r_1^2$$

Neglecting the squares of the creep rates, c^2 and c_1^2,

$$c = c_1\left(\frac{r_1}{r}\right)^2$$

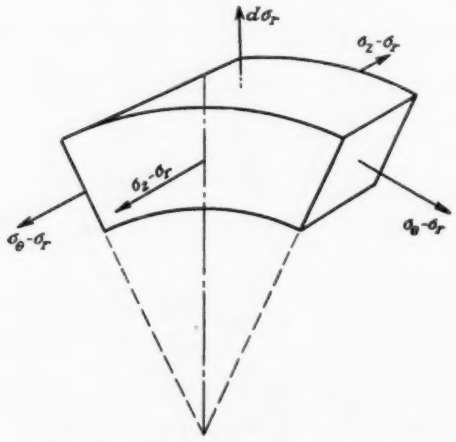


FIG. 16

This can be extended similarly to read

$$c = c_1\left(\frac{r_1}{r}\right)^2 = c_2\left(\frac{r_2}{r}\right)^2 \quad \text{[III]}$$

This equation can be expressed in terms of the creep rate in shear by substituting

$$c = \frac{\phi}{2},\ c_1 = \frac{\phi_1}{2},\ c_2 = \frac{\phi_2}{2}$$

There results

$$\phi = \phi_1\left(\frac{r_1}{r}\right)^2 = \phi_2\left(\frac{r_2}{r}\right)^2 \quad \text{[IIIa]}$$

The reason for introducing shear into this problem is because the term $\sigma_\theta - \sigma_r$ can be expressed directly in terms of the rate of creep in shear as found from experiment.

To see this, suppose, to the system of stress represented in Fig. 16, with the additional condition of Equation [II] that

$$\frac{\sigma_\theta - \sigma_r}{2} = \sigma_s - \sigma_r$$

and neglecting $d\sigma_r$ for the time being, we add a hydrostatic pressure of intensity $(\sigma_\theta - \sigma_r)/2$, we then get the system of Fig. 17. But from a preceding argument, we see this is a stress distribution resulting from pure shear. Experimentally by means of torsion tests a relationship can be established between the rate of creep in shear, ϕ, and the shear stress, $\tau = (\sigma_\theta - \sigma_r)/2$. This relationship, we are assuming as per assumption 1, is $\tau = a + b \log_e \phi$, or

$$\log_e \phi = \frac{\sigma_\theta - \sigma_r}{2b} - \frac{a}{b} \quad [a]$$

Reducing Equation [IIIa] to the logarithmic form, we have

$$\log_e \phi - \log_e \phi_1 = 2 \log_e \frac{r_1}{r}$$

$$(\sigma_\theta - \sigma_r) - (\sigma_{\theta'} - \sigma_{r'}) = 4b \log_e \frac{r_1}{r}$$

where $\sigma_{\theta'}$ and $\sigma_{r'}$ are the transverse and radial stresses at the inner surface. But at this surface, $\sigma_{r'} = -p$. Then

$$(\sigma_\theta - \sigma_r) - (\sigma_{\theta'} + p) = 4b \log_e \frac{r_1}{r} \quad \text{[IIIb]}$$

The problem now consists in solving for σ_θ, σ_r, and σ_s in terms of r by the use of Equations [I], [II], [IIIb], and the proper boundary conditions. The results are those stated by Equations [IV], [V], and [VI].

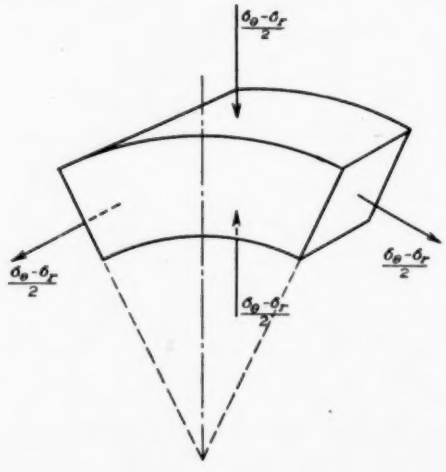


FIG. 17

BIBLIOGRAPHY AND REFERENCES

In compiling this Bibliography, no attempt has been made to give a complete list of the literature on creep of metals, a major portion of which is concerned with the experimental side, but rather to mention principally those articles of recent date which pertain to applications of creep data in design problems. A rather complete bibliography is contained in the symposium referred to in Item (14) below.

For the sake of brevity, the first fifteen items are the references appearing in parentheses in the text; a few of them refer to specific pages in an article or book mentioned.

1 Tapsell, H. J., "Creep of Metals," Oxford Univ. Press, 1931. An excellent and comprehensive work dealing primarily with the experimental side.

2 Bailey, R. W., "Creep of Steel Under Simple and Compound Stresses," *Engineering*, Jan.-June, 1930, vol. 129, pp. 265, 327.

3 Everett, F. L., "Strength of Materials Subjected to Shear at High Temperatures," Trans. A.S.M.E., APM-53-10, vol. 53, 1931.

4 See Item 1 above, p. 116.

5 Baumann, K., "Some Considerations Affecting the Future Development of the Steam Cycle," *Engineering*, July-Dec., 1930, vol. 130, pp. 597, 661, 723.

6 Hatfield, W. H., "Permanence of Dimensions Under Stress at Elevated Temperatures," *Engineer*, Oct. 10, 1930, vol. 150, p. 408. Also *Engineering*, Nov. 7, 1930, vol. 130, p. 600.

7 Barr, W., and Bardgett, W. E., "An Accelerated Test for the Determination of the Limiting Creep Stress of Metals," *Engineering*, March 4, 1932, vol. 133, p. 293.

7a Pomp, A., and Enders, Walter, "Accelerated Method for Determining Creep Limit," *Instruments*, July, 1932, vol. 5, p. 166.

8 Bailey, R. W., and Roberts, A. M., "Testing of Materials for Service in High-Temperature Steam Plants," *Engineering*, March 4, 1932, vol. 133, p. 297.

9 See Item 3 above, p. 126.
10 See Item 5 above, pp. 661, 727.
11 MacCullough, G. H., "An Experimental and Analytic Investigation of Creep in Bending," A.S.M.E. Trans. paper presented at the New Haven meeting of the Applied Mechanics Division, June, 1932. This discusses additional cases and also the general subject of creep in bending.
12 Bailey, R. W., "Thick-Walled Tubes and Cylinders Under High Pressure and Temperature," *Engineering*, June 27, 1930, vol. 129, p. 819. The solution in Art. 7 of this paper is practically that presented in the above.
13 See Item 12 above, pp. 772, 785, 818.
14 Bailey, R. W., Dickenson, J. H. S., Inglis, N. P., and Pearson, J. L., "The Trend of Progress in Great Britain on Engineering Use of Metals at Elevated Temperatures," a paper presented in a "Symposium on the Effect of Temperature on the Properties of Metals," issued jointly by A.S.T.M. and A.S.M.E., 1931.
15 See Item 12 above, p. 786.
Nadai, A., "Plasticity," McGraw-Hill Book Co., New York, 1931. A complete treatise on plastic flow, with references to the subject of creep.
Nadai, A., "The Creep of Metals." A.S.M.E. Trans. paper presented at the New Haven meeting of the Applied Mechanics Division, June, 1932. A valuable analytical treatment of idealized cases of plastic flow involving creep.
Both of the above writings of Dr. Nadai contain references to the literature of creep not mentioned here.
Allen, R. C., "Strength of Turbine Materials for High Temperatures," paper in symposium mentioned in Item 14. Discusses on p. 30 the design of bolts.
Lea, F. C., and Parker, C. F., "The Effect of Temperature on Some of the Physical Properties of Metals," *Engineering*, Jan., 1932, vol. 133, pp. 23, 54. Presents some recent experimental work, including certain experiments where torsion was employed.

Discussion

P. G. McVetty.[4] A survey of the literature on creep of metals reveals a considerable amount of experimental data on the creep of steels under stress at elevated temperatures. Up to the present time, however, relatively little has been published to show the application of creep data to specific cases of high-temperature design. Such applications require necessarily the extrapolation of creep curves and various assumptions as to the laws governing creep. The author has rendered a real service in bringing together the methods proposed by Baumann and Bailey in addition to the results of his own work. These analyses indicate methods of attack which may be applied to other problems which arise in practice.

It seems desirable to emphasize one point of great importance relative to the temperature which is assumed to represent service conditions. Experiment shows that the creep rate is particularly sensitive to temperature changes. If, for example, the temperature rises to 875 F when a value of 850 F has been assumed, we may expect an increase in the creep rate of as much as 6 to 1. This should be kept in mind in any application of creep data so that sufficient allowance is made for possible temperature fluctuations. On this account, designs are usually based on maximum rather than on average service temperatures.

In reference to the permissible creep rates suggested by Baumann for various parts of a steam plant, a distinction must be made between average and minimum creep rates. A study of Baumann's paper shows that these rates are obtained by dividing the total permissible deformation by the time in service. The resulting average rate may be quite different from the minimum rate used in the flanged joint problem. In this connection, Baumann mentions the test of a particular bolt material in which the creep in the first 200 hours is as great as that during 5000 hours at the minimum rate. He also suggests a preliminary treatment for bolts to carry them through the first stage of creep prior to their use in flanges. This has an important bearing on the relaxation problem, since the rapid initial creep causes a more rapid lowering of stress than is obtained from the assumption of a minimum creep rate. This distinction is necessary also in applying published creep rates from various sources, since the term may refer either to an average or a minimum value.

In the flanged-joint problem, the statement is made that the life of the joint may be prolonged by increasing the flexibility of the flange relative to that of the bolts above the assumed value of $q = 2$. This is true only if the change can be made without increasing the creep of the flange. Otherwise a rapid decrease of joint life may result from increased flange flexibility. Further study of the behavior of such joints at high temperature with special reference to flange creep would be of great help in the quantitative analysis of this problem.

The author makes use of the formula, $\sigma = a + b \log_e c$, to express the relation between stress and minimum creep rate at a given temperature. Perhaps it should be mentioned that this relation is only true over a limited range of creep rate. The accurate measurement of such rates below 10^{-7} in. per in. per hr is extremely difficult, and it would be interesting to know the extent of extrapolation involved in Figs. 7 and 9. It might be mentioned that Professor Norton and others have found a linear relation between logarithm of creep rate and logarithm of stress.

This lack of agreement among investigators suggests a critical attitude toward the extrapolation of curves representing such mathematical expressions. The only definite conclusion possible at the present time is that any assumed relation between stress and creep rate must be considered as tentative and subject to revision as our knowledge increases. This applies not only to the fundamental laws governing creep of metals, but also to the proof that such laws apply to commercial alloys.

T. H. Swisher.[5] The writer wishes to call attention to the fact that the value of the modulus of elasticity of steels at elevated temperatures is subject to considerable variation, depending upon the rate of loading.[6] Recently, in our laboratory, a test was made which is of interest in this connection.

A bar of chrome-molybdenum-silicon steel had been under a flow-rate test for 2300 hr at 550 C (1022 F). It had a plastic flow of 1.1 mils per inch, and with a stress of 4300 lb per sq in. was flowing at the rate of 2 per cent in 100,000 hr.

The stress was dropped to 1000 lb per sq in. and the bar immediately shortened 0.13 mil per inch, giving a modulus of

$$(4300 - 1000) \times \frac{1000}{0.13} = 25{,}400{,}000$$

Seventeen hours later the bar had shortened a total of 0.27 mil per inch, giving a modulus of 12,200,000. At the end of 180 hr the bar had shortened a total of 0.34 mil per inch, giving a modulus of 9,700,000.

From this it may be concluded that our information on the modulus of elasticity at elevated temperatures is still somewhat meager. This situation adds further complication to the calculation of flow problems.

Author's Closure

Mr. McVetty very well summarizes the purpose of the paper when he says, "These analyses indicate methods of attack which may be applied to other problems which arise in practice."

[4] Westinghouse Research Laboratories, East Pittsburgh, Pa. Mem. A.S.M.E.

[5] Research Engineer, General Electric Co., Schenectady, N. Y. Jun. A.S.M.E.

[6] "The Modulus of Elasticity of Steel at High Temperatures," by Dr. E. Honegger, *Brown-Boveri Review*, July-August, 1932.

With this as a primary purpose, some of the practical details of creep applications were underemphasized or slighted by the author. Mr. McVetty's comments therefore furnish a valuable supplement to the paper. Before applying the formulas developed, the designer should become thoroughly familiar with such limitations as suggested by the discusser and also by others to be found in many of the references cited in the footnotes and bibliography.

Mr. Swisher presents information concerning the effect of the rate of loading upon the modulus of elasticity. This has a pertinent bearing on the choice of modulus of elasticity to be used in such problems as discussed in the paper.

APM-55-13

Factors Affecting Choice of Working Stresses for High-Temperature Service

By P. G. McVETTY,[1] EAST PITTSBURGH, PA.

This paper discusses the various methods which have been proposed to determine safe working stresses for high-temperature service. The question of the stability of alloys during the test and in subsequent service is considered, with particular emphasis upon probable changes in creep characteristics during long exposure to stress and temperature. It is shown that published data in general do not admit of extrapolation, and that attempts to estimate total creep in service from such data are not usually satisfactory. The author stresses the need for more fundamental study of the laws governing creep rather than creep tests of many different materials.



IT WAS just ten years ago that a paper[2] was presented before this Society which did much to awaken interest in the advantages to be derived from the use of higher temperatures in the steam power plant. At that time it was shown that progress in power-plant design was seriously hindered by lack of reliable information regarding the properties of metals at elevated temperatures. In other fields a similar condition existed, particularly in the oil-refining and chemical industries, and attention was focused upon the tests intended to furnish the desired information. Within two years a symposium[3] was held at a joint meeting of the A.S.T.M. and the A.S.M.E. at which available information on metals and test methods was presented and discussed. The need for a definite program of research was so apparent that the two societies cooperated in the establishment of a Joint Research Committee on the Effect of Temperature Upon the Properties of Metals. Since that time many papers have been published giving the results of an immense amount of work here and abroad. Reference may be made to bibliographies published by this Society in 1928 and 1931, to the second symposium and additional bibliography also in 1931, and to the abstracts published monthly in *Metals and Alloys*.

Two books[4] on creep have helped to summarize the state of our knowledge of this subject. The latest contributions to the literature,[5] together with the extensive discussion which they brought out, are worthy of the most careful study.

It is to these sources of information that the designer naturally turns for guidance in solving the practical problems relating to choice of materials and working stresses for high-temperature applications. After all the work which has been done, he feels justified in expecting to find standardized test methods and a continually increasing fund of reliable, correlated data suitable for design purposes. Unfortunately, this expectation has not been fully realized because of the large number of variables which influence the properties of metals under service conditions, and the time required to investigate the effects of these variables.

A study of the literature for the purpose of collecting and correlating all available information soon reveals the difficulty of the task. Testing methods have not been standardized and the data from different investigators are rarely comparable. The demand for useful information has been so insistent as to stimulate tests of promising materials under prescribed conditions rather than studies of the fundamental laws governing the problem.

Some work of the latter type has been done, but the complexity of the variables has necessitated assumptions which may not be tenable in many cases. It is the purpose of this paper to discuss the validity of some of these assumptions and to show that the use of creep data requires a clear understanding of the effects of test and service conditions and methods of interpretation.

I—Methods of Estimating Safe Working Stresses

Of the many tests which have been proposed, it is now generally agreed that the long-time or "creep" test is the most reliable basis for estimates of working stresses. Whether this be made to determine the effects of stress in tension, shear, or otherwise, we have fundamentally a measurement of strain in the direction of the applied stress while the stress itself and the temperature are kept essentially constant. Up to the present time most work has been done in tension, although creep is generally considered to result from slip along shear planes. Fig. 1 shows a family of tension creep curves for various stresses at the same temperature.

A similar set of curves may be obtained from tests at the same stress and a series of temperatures. Each test requires a different test specimen, all of which must be as nearly alike as possible in order to preserve the proper relations among the several tests.

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[1] M.E., Research Laboratories, Westinghouse Electric & Manufacturing Co. Mem. A.S.M.E. Mr. McVetty was educated at Cornell University, from which he has received the degrees of M.E. and M.M.E. After graduation, he was instructor in experimental engineering and engineering research at Cornell for four years, following which he was for eighteen months a first lieutenant in the Army Air Service. After the war he was connected with the Mellon Institute of Industrial Research, and later became an assistant superintendent with the Jones & Laughlin Steel Corporation, where he gave special attention to open-hearth-furnace research and development. Since 1924 he has been with the Westinghouse Research Laboratories, specializing in the testing and application of materials, particularly at elevated temperatures. This work has been supplemented by metallurgical study at the Carnegie Institute of Technology.

[2] G. A. Orrok and W. S. Morrison, "Commercial Economy of High Pressure and Superheat in the Central Station," Trans. A.S.M.E., vol. 44 (1922), p. 1119.

[3] "The Effect of Temperature Upon the Properties of Metals," Trans. A.S.M.E., vol. 46 (1924); Proc. A.S.T.M., vol. 24, part II (1924).

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] F. H. Norton, "The Creep of Steel at High Temperatures," McGraw-Hill Book Co., Inc., 1929.

H. J. Tapsell, "Creep of Metals," Oxford University Press, 1931.

[5] R. W. Bailey and A. M. Roberts, "Testing of Materials for Service in High-Temperature Steam Plant," Proc. Inst. M. E., vol. 122 (1932), p. 209.

W. Barr and W. E. Bardgett, "An Accelerated Test for the Determination of the Limiting Creep Stress of Metals," Proc. Inst. M. E., vol. 122 (1932). p. 285.

At the higher stresses or temperatures represented by curves *d* and *e*, a transition point is found, beyond which the rate of creep increases. Experience indicates that a continuation of test *c* leads to a similar transition after a considerable time. It is reasonable to expect then that curves *a* and *b* will do the same if the test is sufficiently prolonged. Since the selection of the working stress usually depends upon keeping the deformation within a fixed limit during the expected life of the material, it is necessary either to extend the tests to times as great as 20 years or to resort to some method of extrapolation. In general, the latter is the only practicable course, and it is here that the designer must exercise great care. A test extending over a few thousand hours at most is often extrapolated to 100,000 hr or more. The term "creep rate" may mean the minimum rate at the inflection point, the minimum rate measured during the test, or the average rate from the beginning of the test. In any case, the implied constant rate is not realized in practice and a statement that the creep is 0.001 per cent in 1000 hr cannot be interpreted to mean 0.1 per cent in 100,000 hr. These facts are well known, but they are often neglected in the application of charts similar to Fig. 2 which put creep information into a form convenient for design use,

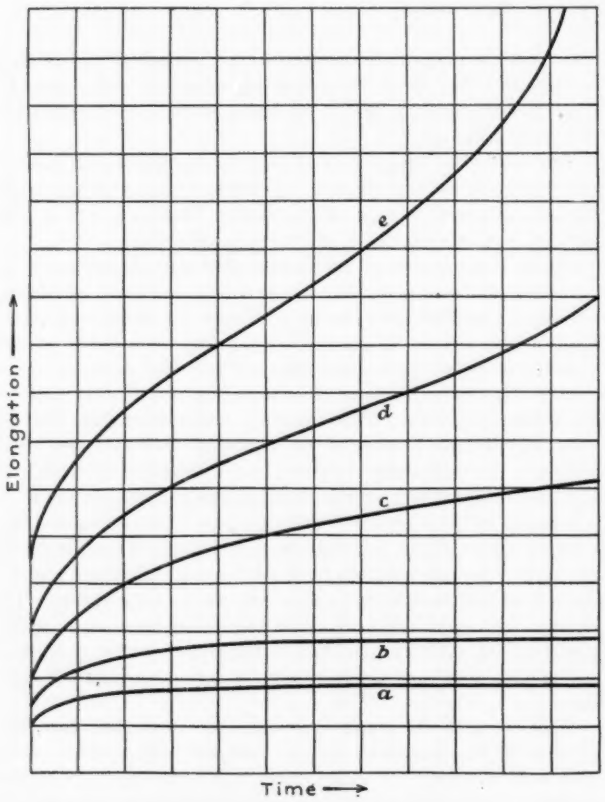


FIG. 1 CREEP CURVES FOR VARIOUS STRESSES AT SAME TEMPERATURES

When this chart is considered by itself, the inference that a definite rate of creep is associated with each combination of stress and temperatures is unmistakable. The designer is tempted to divide his total allowable deformation by the expected life to obtain an average rate similar to those represented by the curves. Whether the chart was originally based upon average or minimum creep rates, this method of extrapolation is not justified by the form of curve shown in Fig. 1.

An additional problem is presented in many applications in which the allowable creep is less than that indicated by the curves. Reference to the original data in the form of creep curves fails to show a rational basis for extrapolation which is generally applicable. By making certain assumptions, it is possible in any particular case to extrapolate into the field of smaller creep. However, the validity of each assumption is questionable and an attempt to add curves for smaller creeps to Fig. 2 requires careful consideration of the possible errors involved.

One common assumption is that a straight-line relation exists between the logarithm of the creep rate and the logarithm of the stress in tests at any one temperature. Many curves plotted in this way appear to confirm this linear relation. Other investigators plot the logarithm of the creep rate against the stress itself and again find a linear relation. In either case, extrapolation of the straight line to extremely small creep rates gives what appears to be an easy method for extension of the range of Fig. 2. Obviously the two methods cannot agree, and the error lies in the assumption that a certain creep rate is associated with a given combination of stress and temperature. Furthermore, it is evident from Fig. 1 that total creep at the end of a given time is not obtainable directly from any measured creep rate.

Since simple methods of extrapolation are not applicable to the general case, it appears necessary, in the present state of our knowledge, to go back to a study of the original creep curves in order to estimate the total creep which may be expected in any particular case. A consideration of these fundamental curves does much to explain the difficulty of correlating existing data.

II—THE CREEP TEST

Fig. 3 shows a family of creep curves similar to those in Fig. 1. These indicate the general nature of creep at the same temperature and six different stresses. The portion of the curves to the left of line *M–N* represents the relations revealed by actual tests, while that to the right represents what may happen in service in times greater than the duration of any test. At the higher stresses, transition points such as *a*, *b*, and *c* are frequently found. The measured minimum creep rates at these points are sometimes used in extrapolating to lower rates at lower stresses, as in the construction of charts similar to Fig. 2. All experimental evidence indicates that similar transitions such as *d* and *e* would be found if the tests were sufficiently prolonged. The same reasoning leads to a statement that any creep curve such as *F* or *G* will have a creep rate gradually decreasing to a minimum, beyond

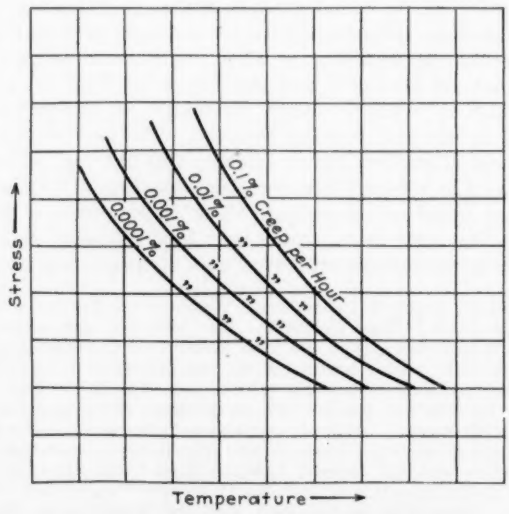


FIG. 2 CURVES OF CONSTANT CREEP RATES

which the rate will increase. If service conditions are such that this transition point is not reached within the life of the material, the estimated total creep from the curves will be on the safe side. This places considerable importance upon the location of the curve R–S joining the several transition points. On account of the time required to locate transitions at the lower stresses, very little is known about this curve. It is evident that any estimate of total creep at the end of ten to twenty years based upon creep curves extending over a few thousand hours requires consideration of such transition points.

The form of the known curves A and B in Fig. 3 indicates the action of two opposing forces, one diminishing and the other increasing in magnitude as time goes on. The transition points a and b represent a temporary balance of the two forces. In the simplest case we may assume that the adjustment of stress over the cross-section causes relatively large deformation of a few crystals in the first stage of creep. The resulting strain hardening reduces the rate of deformation toward what appears to be a constant creep rate when the stress has been uniformly distributed. As time goes on the cumulative effect of time and temperature relieves this strain hardening and causes a gradually increasing creep rate. Applying the same reasoning to curves representing lower stresses, we may conclude that curves F and G, Fig. 3, will act similarly if exposed to stress and temperature for a sufficient length of time. Under these conditions it is difficult to see how any so-called "creep rates" taken from relatively short creep tests can offer a rational method of extrapolation to times representing the service life of the material. If working stress must be based upon allowable deformation, due consideration should be given also to the first and third stages of creep.

III—Stability of Alloys in Service

It has been shown by Bailey and Roberts[6] that plain carbon steels undergo a change in the carbide from the lamellar to the spheroidized form during long exposure to elevated temperature. Their experiments show a considerable increase in the creep rate after this change has taken place. From this we may expect an increase in creep rate of such materials at the end of a time which may be greater than that represented by any ordinary creep test unless precautions are taken to stabilize the material prior to its use. Much further investigation is necessary to determine the extent of this structural change in service.

Complex alloy steels of the type usually recommended for high-temperature service are particularly subject to internal changes. Age hardening in service is a familiar example. A gradually increasing strength would tend to decrease the creep rate. An investigation of this phenomenon has shown that the hardening effect proceeds to a maximum value, after which it decreases toward a stable condition. The time required to reach the maximum may be short or long, depending upon the temperature. If the maximum is not reached during the creep test, we have an effect opposing the normal creep during the test, but tending to add to the creep effect in subsequent service. A creep rate measured under these conditions is decidedly misleading.

Another possibility is an actual volume change under the action of stress and temperature. Several investigators have found what appeared to be negative creep rates. This cannot be due to age hardening, and it must be explained by actual shrinkage of the material. When such shrinkage exceeds the creep, the negative readings serve as a warning, but when slightly less than the creep, a dangerous interpretation of the resulting low creep rate may result.

[6] R. W. Bailey and A. M. Roberts, "Testing of Materials for Service in High-Temperature Steam Plant," Proc. Inst. M.E., vol. 122 (1932), p. 209.

These disturbing influences cannot be ignored in the interpretation of creep tests. When the test is conducted at constant stress and temperature while deformation is measured as a function of time, conclusions must be based on the assumption that no other uncontrolled variables are operating. If the creep test is to have rational interpretation, the same care in controlling other variables must be used as is now employed to control temperature.

Any attempt to add high-temperature strength which makes the alloy unstable under service conditions is undesirable. This statement applies to cold work, age hardening, or other processing

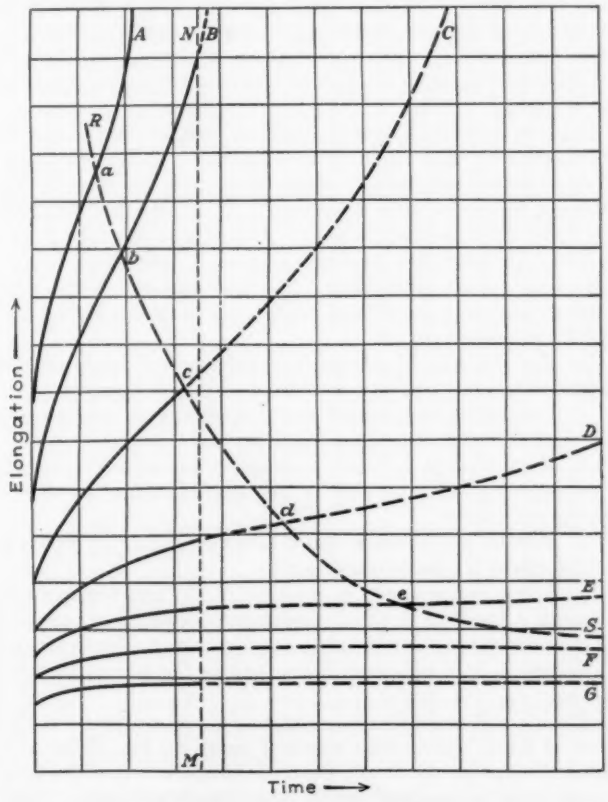


Fig. 3 Extrapolated Creep Curves

of the material. Unless great care is used, these treatments may impart a temporary creep resistance which is not maintained in service.

IV—Need for Fundamental Research

The difficulties inherent in the conduct and interpretation of creep tests have been mentioned primarily to emphasize the importance of investigating the fundamental laws involved. For example, Tapsell[7] states that "anything like a correct estimation of the total deformation in, say, 50,000 hr based on a 1000-hr test would appear to be a matter of chance." In their recent paper Bailey and Roberts[8] attack the wide use of minimum creep rates in the following statement: "It will be clear that at temperatures of interest in connection with steam power plant and in cases where small creep only is permissible, no practical extrapolation of minimum creep-rate data of the kind plotted in Fig. 40 (temperature versus logarithm of minimum creep rate,

[7] H. J. Tapsell, "Creep of Metals," p. 171.

[8] R. W. Bailey and A. M. Roberts, "Testing of Materials for Service in High-Temperature Steam Plant," Proc. Inst. M.E., vol. 122 (1932), p. 264.

curves of constant stress), or plotted against stress, or logarithm of stress, the temperature being constant, is likely to lead to satisfactory working values of stress and temperature. Such extrapolation will result in stresses and temperatures too high for safety." Statements of this kind from recognized authorities cannot fail to warn the designer against acceptance of test data without careful consideration of the methods used. They serve a further useful purpose in stimulating interest in the more fundamental aspects of the problem.

Before making extensive creep tests, it seems desirable to study the factors affecting the stability of the material not only during the test, but also in subsequent service. This is primarily a metallurgical problem involving the equilibrium of the alloy constituents under conditions existing in service. This phase of creep testing deserves much more attention than it has received in the past.

Another promising method of attack requires creep tests at temperatures higher than those of service. It is conceivable that a testing temperature might be selected at which all the changes associated with a service life of many years might be crowded into a period of 1000 hr. This suggests a study of the relation between time and temperature in producing definite amounts of total deformation. The establishment of this relation would lead to a rational accelerated creep test having the following advantages:

1 The stress and resulting total deformation would be the same in the short test as is considered allowable in service.

2 The higher temperature would accelerate all the internal changes which affect deformation in service.

3 Results would be based upon the attainment of a definite increase in length rather than on the measurement of extremely small rates of flow.

4 Tests at higher temperatures would furnish direct evidence of the effects of overheating in service.

5 Extrapolation would be confined to the time-temperature relation. As means of initial stabilization become better known, this would lead to the establishment of laws applicable to groups of materials, with the possibility of finding the general laws involved.

There is still much that is speculative in our conception of the laws of creep. The recent excellent paper by Dr. Nadai[9] has done much to express clearly the fundamental nature of the laws involved and to point the way for further analytical study. This type of study of reliable test data from carefully prepared test specimens appears to offer great promise of useful results. One of the most important phases in this field is the study of combined stresses. This includes not only the usual combinations of tension, compression, and shear, but also the effects of vibration and impact upon creep. Some work has been done on high-temperature fatigue tests, but much study is still required of the case in which the stress varies in intensity but not in direction. Another field of investigation would determine the effects of various surrounding atmospheres such as superheated steam or flue gas. A methodical study of these problems may well be considered as desirable fundamental research.

V—Conclusions and Recommendations

The growing needs of the designer furnish ample incentive for further study of the laws of creep. It is only in this way that we may cope with the problems of the future. However, an immediate pressing need confronts us in the selection of working stresses to be used in present designs. It is impossible to wait for further tests, and we have but to consider the advances in design of the past ten years to be convinced that there is real value in the data available. In attempts to summarize and correlate all these data, a few precautions may serve as a guide toward rational conclusions.

In general, the original creep curve offers the most reliable basis for estimating deformation in service. From it, allowance may be made for initial creep, together with provision for the probable effects of test and service conditions. The actual procedure involves extrapolation of the final tangent of the creep curve to the deformation corresponding with the expected service life. The possibility of a transition in the creep curve must be carefully considered in connection with the occurrence of such transitions at higher stresses and temperatures. The factor of safety must include the large effect of even slight overheating in service.

Many of the published data give creep rates such as 1 per cent in 100,000 hr. These rates appear to represent a certain stress and temperature, and their relative value depends on uniformity in the method of extrapolation. In general, they are more useful for comparing materials than for assigning working stresses. In any case, the method of measuring creep rate must be known before comparing data from different sources.

In the use of test data, preference must be given to those representing maximum care in the conduct of the test. The necessary refinement of test methods is directly related to the permissible deformation, and it is obviously wrong to accept data not suited to the particular application.

With these precautions in mind, it is possible to select from the literature the data most suitable to the problem at hand. When due regard is paid to the variables involved, extrapolation loses much of its uncertainty. For some time to come it is probable that design will err on the safe side. The prospect for increased working stresses will depend upon the extent of study of the laws involved and their application to practical problems.

VI—Acknowledgment

The author wishes to acknowledge his indebtedness to Dr. A. Nadai and Mr. R. E. Peterson for their unfailing interest and many helpful suggestions and criticisms.

Discussion

Pierre de Bethune.[10] The writer would like to emphasize the point made by the author that the creep curves indicate "the action of two opposing forces," and to present a more accurate analysis than that given in the paper, because it is his belief that the solution of the creep problem lies in a critical examination of the form of the creep curves.

Referring to Fig. 4, the fact of two opposing forces may be expressed by decomposing the curve of creep into two components, curves 1 and 2. The form of curve 1 suggests the elastic after-effect. If this were true we could apply the author's statement that "the adjustment of stress over the cross-section causes relatively large deformation of a few crystals in the first stages of creep." Of course the movement ends not only on account of the strain hardening of those grains, but also because their overstressing has simultaneously been reduced.

The writer would not consider curve 1 as due only to an elastic after-effect, because in this case the deformation is mostly elastic, only a few grains being deformed plastically, and therefore the amount d of the deformation should be small. He would more likely consider that the plastic deformation reaches every grain in the metal. In this case the discontinuance of the deformation is due only to strain hardening, this effect becoming more and more important as time progresses and reducing relatively the speed of deformation.

[9] A. Nadai, "The Creep of Metals," A.S.M.E. New Haven Meeting, June, 1932.

[10] Louvain, Belgium.

Curve 2 represents what should really be the creep of the metal. This has been drawn with an increasing rate of creep, but the changes suggested in section III of the paper, Stability of Alloys in Service, will introduce irregularities into the form of the curve.

The deformation in this case can be ascribed to an intergranular or an intragranular mechanism. The author believes that "the creep is generally considered to result from slip along shear planes," which means intragranular deformation. The writer is

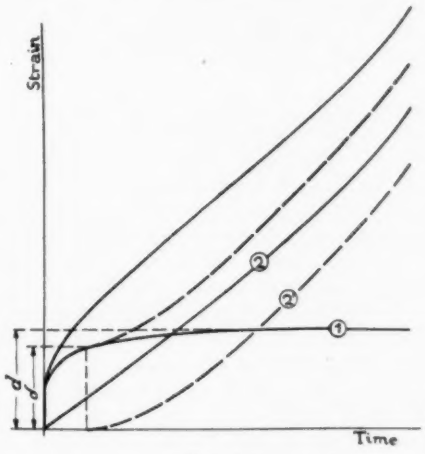


FIG. 4

much inclined toward this latter hypothesis, but would not underestimate a priori the possibility of the former.

However, in this latter hypothesis, as the author points out, the creep requires the relief of strain hardening. Since his introduction of time and temperature may seem in certain respects arbitrary, the writer will call attention to a number of facts set forth by many metallographers which seem to afford a physical explanation of this phenomenon.

The strain hardening is due mostly to the cessation of the slipping along each slip plane. The movement then continues along another slip plane, and so on up to the point where the slip lamellae become sufficiently thin. The manifestation of further slipping requires a raising of the stress. This cessation of slipping along individual slip planes has not as yet been explained satisfactorily, but it is undoubtedly related to the lattice distortion near the slip plane shown by the experiments of Joffé,[11] Dehlinger,[12] Sinkiti Sekito[13] and others.

Next to this study of cold working of metals, attention must be called to the relation existing between recrystallization and the amount of cold working. The work of many metallographers has shown that in the case of cold-worked metals the recrystallization brought about by raising the temperature begins at temperatures even lower than the general transition points.

It is thought, therefore, that this recrystallization may obliterate the distortion of the lattice due to the slipping. The crystal lattice will be restored in an unstrained state, thus allowing for further movement without increase of the stress.

This would explain curve 2,[14] the rate of creep depending on the speed of plastic deformation and of recrystallization.

No definite proof of these views has yet been given, and no relation has yet been shown to exist between creep and recrystallization limits, but the writer believes that research based on them as a working hypothesis should lead to interesting results.

D. S. JACOBUS.[15] The author points out the difficulty of interpreting data secured from creep tests and states that the growing needs of the designer furnish ample incentive for the further study of the laws of creep. There is certainly a broad field yet to be covered; in the meantime the designer is forced to proceed on the basis of our present knowledge.

Four or five years ago published results for the stress to produce creep at higher temperatures caused apprehension in regard to stresses that were in use. This led to a paper entitled "Working Stresses for Steel at High Temperatures," which the writer presented at the Annual Meeting, December, 1929. In this paper the stresses were based on the results secured in practice and the results of creep tests were used to set the stresses for higher temperatures than those for which actual operating data were available, so as to give a set of working stresses for plain carbon steel up to temperature of 950 F for drums of boilers and pressure vessels and for superheater headers.

The stresses were based on what had been found suitable through long use in boiler and superheater practice and on data secured from oil stills, the first of which was constructed about 10 years before the time of writing the paper. The superheater and oil-still data established a working stress of 8000 lb per sq in. at 800 F for steel having a minimum tensile strength of 55,000 lb per sq in.

An abstract of the paper giving the proposed working stresses was published in *The Metallurgist* (supplement to *The Engineer*, London), March 28, 1930.

The working stresses specified in the A.S.M.E. Boiler Code for plain carbon steels at high temperatures are in substantial agreement with those given in the writer's paper for the same grades of steel, and for other grades they are in line on the basis of creep-test values.

Table 1 is a copy of Tables P-6 and U-3 of the A.S.M.E. Boiler Code. Opinions regarding the correctness of the values given in this table would be appreciated.

TABLE 1 WORKING STRESSES FOR DRUMS OF BOILERS AND PRESSURE VESSELS AND FOR SUPERHEATER HEADERS CONSTRUCTED OF PLAIN CARBON STEEL

Max. temp., deg F	Minimum of specified range of the tensile strength of the material, lb per sq in.				
	45,000	50,000	55,000	60,000	75,000
700	9,000	10,000	11,000	12,000	15,000
750	8,220	9,110	10,000	11,200	13,000
800	6,550	7,330	8,000	9,000	10,200
850	5,440	6,05	6,750	7,400	8,300
900	4,330	4,830	5,500	5,600	6,000
950	3,200	3,600	4,000	4,000	4,000

It should be borne in mind, as brought out in the paper referred to, that the working stresses which are given are for the constructions specified and that they are not universally applicable. In certain types of apparatus where the life is less than in a boiler, higher stresses may be used to advantage, whereas in other types the working stress should be lower.

F. N. SPELLER.[16] The author brings out clearly the need for more fundamental research on the factors affecting the creep values of metals at high temperatures; particularly is this necessary in connection with the changes that occur in the properties

[11] Joffé, A. F., "The Physics of Crystals." New York, McGraw-Hill, 1928, p. 47.

[12] Dehlinger, U., *Zeit. f. Kristallographie*, vol. 65 (1927), p. 65, and *Naturwissenschaften*, vol. 17 (1929), p. 545.

[13] Sekito, S., Scient. Reports Tohoku Imperial Univ., vol. 16 (1927), p. 343, and vol. 17 (1928), pp. 680 and 1227.

[14] It seems to result from certain experiments that the recrystallization requires some strain, irrespective of the stress responsible for that strain. This is represented by shifting of curve 2 to curve 2', creep beginning only when the deformation along curve 1 has reached the critical value δ.

[15] Advisory Engineer, The Babcock & Wilcox Co., New York, N. Y. Past-President A.S.M.E.

[16] Director, Metallurgy and Research Department, National Tube Co., Pittsburgh, Pa. Mem. A.S.M.E.

of steel at elevated temperatures with time. In this connection, it is quite important to know how to put the metal in the most stable condition in a relatively short time so that no material change affecting its physical properties will occur during the period of the creep test or the life of the metal in service.

It seems highly desirable before undertaking a more extensive creep-test program for all concerned to agree upon a tentative standard of procedure. With this in mind the American Petroleum Institute recently appointed a small committee with powers to cooperate and act with the Joint Committee on Effect of Temperature on the Properties of Metals and the tube manufacturers, with a view to arriving at a definite specification as to the procedure. While this perhaps cannot be completely accomplished at present, it will nevertheless be a forward step. With a better understanding of the factors that are now known to affect creep, refinement of equipment, and methods of controlling and measuring temperatures and elongation, much more reliable results may be expected.

In this connection it may be stated that the Metallurgical Advisory Board of Pittsburgh expect to be able to continue next year their cooperative investigations of the effect of inclusions on the physical properties of steel. The results obtained on control of non-metallic inclusions in steel and their influence on the properties of the metal may prove of interest in connection with high-temperature testing, although the work was originally undertaken to obtain closer control of the physical properties of steel in the lower range of temperature. Samples of carbon steels with various amounts and kinds of inclusions are available for creep investigation. The creep of metals is an ideal subject for cooperative research on account of economy, efficiency of operation, and general application of results. When the procedure has been standardized and a program of tests carefully prepared in detail, there should be no trouble in getting the necessary funds from those interested to carry on work to obtain reliable data on this important subject.

Author's Closure

The author is quite in agreement with Mr. de Bethune's statement that the solution of the creep problem lies in a critical examination of the form of the creep curves. So long as practical considerations make it necessary to extrapolate such curves over intervals as great as one hundred times the length of the usual creep test, it is necessary to analyze the various factors which may influence the relations among the four variables of strain, time, stress, and temperature. The results of considerable work in this direction will be given in a later paper.

The statement that curve 2 in Fig. 4 represents the real creep curve in which velocity increases with increasing time may be somewhat disturbing to the designer. For applications in which elongation is limited, the stresses are relatively low, and the creep curve may be represented by curve *G* in Fig. 3. It is probable that a very long time would elapse before the intersection of curve *G* with curve *R-S* and a greater time before an appreciable increase in velocity. This slow approach toward a minimum velocity at low stresses helps to justify the assumption that the creep curve may be replaced by a straight line in the neighborhood of the transition point. At higher stresses, the transition occurs within shorter periods of time, but the corresponding elongation is usually greater than would be permissible in practice.

In reference to the relation between recrystallization and the amount of cold working, this depends not only on temperature, but also on the time of exposure to the temperature. This might account for the increasing creep rate shown in curve 2 in Fig. 4. Unless precautions are taken to keep unit stress instead of total load constant, a similar effect would result from the decrease in cross-section which accompanies any increase in length of the test specimen.

Dr. Jacobus rightfully emphasizes the importance of utilizing to the fullest possible extent our present knowledge of creep properties of metals. The values given in his 1929 paper and in the A.S.M.E. Boiler Code have been most useful in boiler and superheater design, but they do not provide a means of estimating the deformation which may be expected to result from the use of the suggested working stresses. The author has considered primarily those applications in which allowable deformation is the criterion of safety. Bailey[17] has discussed this problem and has found for medium-carbon steels an approximate relation among the variables, tensile stress, temperature, and creep rate. In the same paper he gives a curve which allows extrapolation from 750 F to 1000 F where reliable data are available at 750 F. The approximate agreement with the values given in Table 1 is probably less important than the fact that Boiler Code values have been found satisfactory in practice.

Dr. Speller has mentioned the desirability of adopting some standard procedure for creep tests. Agreement on testing procedure requires compromises, since one application may allow 3 per cent creep in one year while another is limited to 0.2 per cent in 20 years. To cover such a wide range, several methods of measuring creep rate, controlling temperature, etc. may be necessary, and this adds difficulty to the problem of securing agreement on a single procedure. A tentative code has been prepared and is now being discussed by the Joint Research Committee. It is expected that this will be published in the near future.

The author has followed with great interest the work of the Metallurgical Advisory Board, of Pittsburgh, in connection with the influence of non-metallic inclusions on the properties of metals. The extension of this investigation to problems involving the use of metals at high temperatures opens a field of great practical importance. Up to the present time most high-temperature tests have been confined to very uniform material, on account of the necessity of restricting the variable factors to a minimum. As our knowledge of fundamental creep phenomena increases, the importance of other metallurgical factors such as the amount and kind of non-metallic inclusions must be recognized. Exploration of this promising field may account for some of the occasional peculiar test results which cannot now be explained.

[17] A.S.T.M.-A.S.M.E. Symposium, 1931, p. 231.

APM-55-14

Notched-Bar Tests[1]

By MAX MOSER,[2] ESSEN, GERMANY

In the present paper the author discusses the influence of such factors as temperature, width, and geometrical similarity in dimensions of specimens on the impact values as obtained by means of the notched-bar test. Various shapes of specimens for notched-bar tests are compared and their impact values determined.

DURING the last few decades the notched-bar test has found application to a rapidly growing extent as one of the procedures for the mechanical testing of materials. In connection therewith, valuable research work has been performed by various investigators and this work has enabled the author of the present paper[3] to determine the relationship existing between the test results and the various conditions under which the tests have been made.

In so far as the evaluation of the results obtained from notched-bar tests is concerned, several difficulties are still being encountered, and it seems that they can be overcome only by adopting certain standards for conducting the tests. The nature of these difficulties will first be analyzed, after which will follow a description of the attempts that are being made in Germany to introduce a set of standards covering the notched-bar test, and to promote the reaching of an international agreement concerning the establishment of such standards.

The Present Situation and the Requirements to Be Met

When steel rods of different dimensions are subjected to a tensile test, the results will be the same for all specimens, provided that the latter have the same proportions. For instance, the specimens *a* and *b*, see Fig. 1 and Table 1, have the same ratio of 10 to 1 between gage length and diameter. But the tensile test offers still broader possibilities. The dimensions of the specimen *c*, Fig. 1, are not proportional to those of the specimens *a* and *b*. Specimen *c* has the same cross-section as specimen *a*, but its ratio between gage length and diameter is different, i.e., 5 to 1. Nevertheless the anticipated, comparable values are actually obtained.

TABLE 1 RESULTS OF TENSILE TEST FOR THE SPECIMENS *a*, *b*, *c* OF FIG. 1

	Specimen		
	a	*b*	*c*
Yield stress, σ_s, kg per sq mm	36	36	36
Ultimate strength, σ_B, kg per sq mm	64	64	64
Elongation, δ, per cent	$\delta_{10} = 22$	$\delta_{10} = 22$	$\delta_5 = 26$*
Contraction, ψ, per cent	47	47	47

* $\delta_5 = 1.2\delta_{10}$.

Conditions are different, however, for the notched-bar test. Each pair of notched-bar specimens in Fig. 2, either at the left or the right, has the same proportions between the dimensions of the two specimens. Nevertheless the results of the notched-bar tests obtained for the two specimens of each pair are far from being identical for the small and for the large specimen. This would not be objectionable if the impact values for the large and for the small specimens bore at least a definite ratio to each other. However, the quoted values indicate that even this is by no means the case. In consequence no conclusions can be drawn from the results of one notched-bar test that will be of any assistance in predicting the values to be obtained from another, not even when the specimens have proportional dimensions. (Compare the impact values given in Table 2 for the specimens shown in Fig. 2.) For the same reason it is not possible either to compare the results of tests made with different specimens. And this in spite of the fact that every governmental commission, every association, and every manufacturing company has its own particular type of specimen.

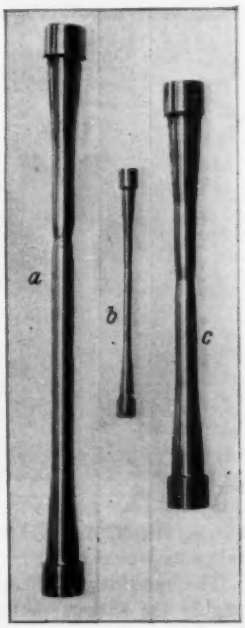


FIG. 1 LARGE AND SMALL TENSION TEST SPECIMENS (For test values see Table 1.)

A standardized method of dealing with problems relating to the strength of notched specimens is still lacking, not only in international testing procedure but also in that of the various countries.

What now is the significance of the present situation as set forth in the previous paragraphs?

It is true that when a higher or lower figure for the impact value is found, this is not a reason for reinforcing or weakening our machine or structural parts, nor does it constitute a criterion for judging the strength of those parts, because impact values are not a basis for designs. In other words, the notched-bar test does not give any practical design information. Nevertheless it is of value, because it makes possible, for instance, a comparison of the ductilities (toughnesses) of various grades of steel, and more

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[1] Although questions concerning standardization of methods in testing materials do not strictly belong to the subjects of this symposium, it was felt that the present excellent review of some of the main factors which have a pronounced influence on the impact values as determined by notched-bar tests, kindly prepared by Dr.-Ing. Moser, an authority on the impact testing in Germany, should be brought before the American engineer. The Applied Mechanics Division is particularly calling attention of the engineers of this country to this subject because of the extended use the impact test has found in Europe as a quick method for controlling certain treatments of materials.

[2] Head of the Testing Laboratory (Probieranstalt) and the Material Acceptance Department of the Fried. Krupp Aktiengesellschaft, Gussstahlfabrik, Essen. Dr.-Ing. Moser was born at Brumath, Alsace, in 1873; he studied at the Universities of Strasbourg, Stuttgart, and Berlin, was graduated as a mechanical engineer, became assistant at the laboratories for testing materials at Stuttgart and Neubabelsberg, and later entered the service of Fried. Krupp at Essen, where he has remained ever since. He has published a great number of scientific papers on important problems, such as those concerning strength and rupture of materials, and is one of the first authorities on standardization of testing methods in Europe. He has particularly investigated the conditions determining the yield point of steel and the impact test. He is a co-author of the German Handbook on the rolling process, and is chairman of the committee on impact tests which presented a report before the International Association for Testing Materials at Zurich in 1931.

[3] Translated from the German by A. van Niekerk, Pittsburgh, Pa.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., Dec. 5 to 9, 1932, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

particularly of steels at low temperatures. Further, being a typical aging test, it leads to certain conclusions concerning the metallurgical characteristics of steels. And finally, it affords the possibility of making a technological check test for the purpose of determining whether the material, especially steel, actually has the structure which it was assumed to have when it was selected for a certain machine or structural part.

TABLE 2 IMPACT VALUES (WORK PER UNIT OF AREA OF CROSS-SECTION) FOR THE NOTCHED-BAR TEST SPECIMENS *a*, *b*, *c*, *d* OF FIG. 2

Specimen	Impact value, kg per sq mm	Ratio
a	22.3	*a*:*b* = 2.1
b	10.6	
c	0.6	*c*:*d* = 0.75
d	0.8	

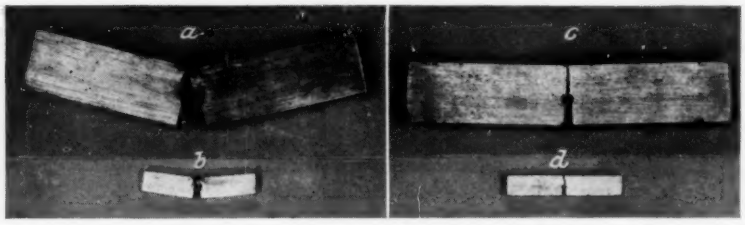


FIG. 2 LARGE AND SMALL NOTCHED-BAR TEST SPECIMENS
(For impact values of specimens see Table 2.)

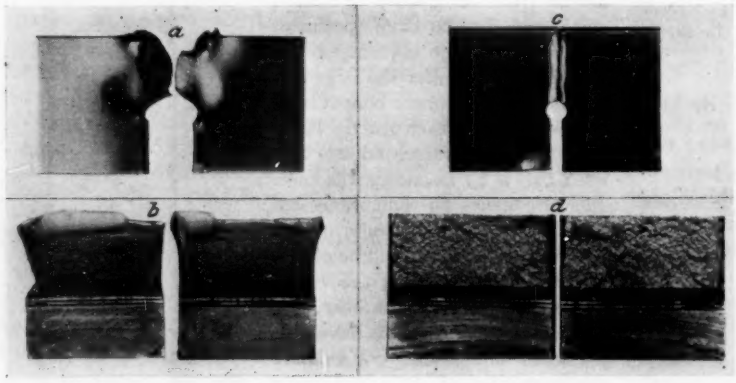


FIG. 3 SLIDING FAILURE (*a*, *b*) AND SEPARATION FAILURE (*c*, *d*)

The tendency, noticeable everywhere, to investigate thoroughly the characteristics and behavior of materials from the standpoint of impact testing, has its origin in the recognition that the notched-bar test is of decided value in connection with our attempts to make our designs perfectly safe.[4]

But the results of the research work carried out by individual investigators cannot be of general benefit, and their discussion cannot be of any value, unless the regrettable situation prevailing at present, as pointed out in the foregoing, is improved to such an extent that we shall be able to express all proceedings relating to notched-bar testing in a *uniform* language that will be readily understood by all those concerned, and will not leave the slightest doubt as to the true meaning of the terms and designations employed.

This means that all reports publishing test results, all comparisons, and all conclusions must be made on the basis of a definite, generally accepted, and uniform testing procedure; or, expressed in other words, we must *standardize* the impact or notched-bar test.

[4] See F. Rötscher and M. Fink, "The Notched-Bar Impact Values of High-Grade Construction Steels," *Z.V.D.I.*, vol. 76 (1932), no. 8, p. 173.

The adoption of a local standard for each individual country will be satisfactory when the organizations of cooperating interests do not expand beyond the frontiers of those countries. However, the iron and steel business is international in extent. For this reason it is quite interesting to note that the New International Association for Testing Materials, which held its First International Congress at Zurich in September, 1931, has designated the problem of standardizing the notched-bar test as one of the first problems that should be studied.[5]

It is well known that we have a specimen, the so-called Charpy specimen, which has been more or less considered as an international standard specimen. This Charpy specimen was very satisfactory from a testing standpoint, but on account of its size its use has fallen off. With the development of the use of high-grade steels to the present stage, it has become of increasing importance to utilize specimens that can be taken even from forgings of very small size, or from portions of large forgings where the designer anticipates that the stress conditions will be the most dangerous. And finally, when it is necessary to examine broken parts, it is desirable to take specimens exactly from those points where the failure has occurred. None of these things is possible, however, when specimens having the size of a Charpy specimen must be procured; and this explains why there is everywhere a search being made for suitable standardized specimens of small size.[6]

Committees have also been organized in Germany by the German Association for Testing Materials as well as by the German Institute of Steel Engineers (Verein Deutscher Eisenhüttenleute), which have for years been attempting to solve the problem of determining the best type of small specimen for notched-bar tests. As a result, we were able to present to the Congress at Zurich a unanimous German proposal which apparently satisfies all the requirements that must be met by a small-size standard specimen. This proposal was intended, first, to provide a tentative standard for use in Germany, and, in addition, to serve as a basis for seeking an international agreement.

FACTS AND POSSIBILITIES

In order to be able to present a proposal having the highest possible degree of perfection, it was first necessary to make a final check of all available knowledge concerning the factors which influence the results of a notched-bar test, and to fill in the remaining gaps in this knowledge. We finally succeeded in putting our knowledge of the fundamentals in a satisfactory and fairly complete shape, and an attempt will now be made to describe briefly the principal facts; a more elaborate description, including a bibliography of recent researches relating to the subject, was presented to the Congress at Zurich in the form of a report, to which reference has already been made.

To begin with, a structure can be occasionally overloaded

[5] The author was requested by the German Association for Testing Materials to report on the situation of the problems relating to notched-bar testing. The report was written with the cooperation of Dr. Fettweis and Dr. Mailänder, and has been published in the "First Series of Communications of the First International Congress held by the New International Association for Testing Materials at Zurich, Group A, Zurich, 1931." An abstract, not explicitly mentioning the source, can be found in *Z.V.D.I.*, vol. 75 (1931), p. 1401, in an article by E. Lehr on the testing of materials.

[6] See: Fr. F. Fischer, "How Must a Small Standard Specimen for Notched-Bar Tests Be Selected?" *Kruppsche Monatshefte*, vol. 9 (1928), p. 53. See also *Stahl u. Eisen*, vol. 48 (1928), p. 541.

without any harm, even by impact loads, when its design or the properties of its material are such that the overloading will be reduced automatically by a yielding of the structure, that is, by a deformation of a comparatively large part of it. However, such overloads can be extremely dangerous when the deformation remains restricted to a small part of the structure. The occasional overloading that the structure is able to withstand will be much higher in the first case, and an examination of a broken part will reveal the fact that the fracture did not occur until the susceptibility of the material to deform under stresses had reached the limit; in this case the designation "sliding failure" (*Verformungsbruch*) is used.[7] But in the second case, failure will have been caused by a comparatively small overload, and the evidence exhibited by the broken parts of having been deformed considerably will not be very pronounced. In this case we have the so-called "separation failure" or "cohesion failure" (*Trennungsbruch*); due to the absence of any stress-reducing reaction, the increased stress has been able to overcome rapidly the resistance offered by the material against separation, thus causing the abrupt destruction of the structure.[8]

In the author's opinion, the notched-bar test is important from a test standpoint mainly because, by performing this test, we are able to force the material to disclose its resistance against failure due to separation when it is subjected to impact loads. In order to have the material reveal this fact, we compel it to remain in a condition of restricted stress equalization, by providing the notch with adjoining stiff walls. Fig. 3 shows specimens of a steel that was available in two forms with different structures, after they had been subjected to notched-bar tests. In its first form (see *a* and *b*) the steel does not show a great tendency toward separation failure, whereas in its second form (see *c* and *d*) this tendency is strongly pronounced; the test conditions were the same for

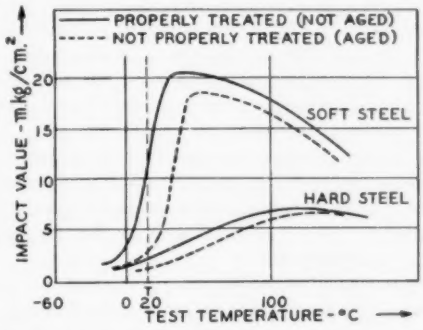


FIG. 4 HIGH AND LOW IMPACT VALUES AT VARIOUS TEST TEMPERATURES (MAILÄNDER)

(Impact value = work per unit of area of cross-section.)

both specimens. Notwithstanding the condition of restricted stress equalization during the notched-bar test, the steel in its first form shows a "sliding failure;" in its second form it shows a "separation failure." The measured impact values (the work per unit of area of the cross-section along which the test pieces break) are high in the first case and low in the second. In this connection the designations *high* impact value and *low* impact value (computed per unit of area of the cross-section along which the rupture occurred) will be used.

The state of stress or the strained condition enforced upon the steel during the notched-bar test can be influenced and, incidentally, if desired, can be intensified in three different ways:

1 By changing the temperature prevailing during the test

2 By changing the ratio between height and width of the cross-section subjected to the impact

3 By changing the shape of the notch, i.e., its sharpness and depth.

A further possibility of exerting the influence just mentioned consists in changing the working speed of the impact testing machine; this possibility hardly needs to be mentioned, since it does not find application in practice.

1 Fig. 4, taken from a paper prepared by Dr. Mailänder,[9]

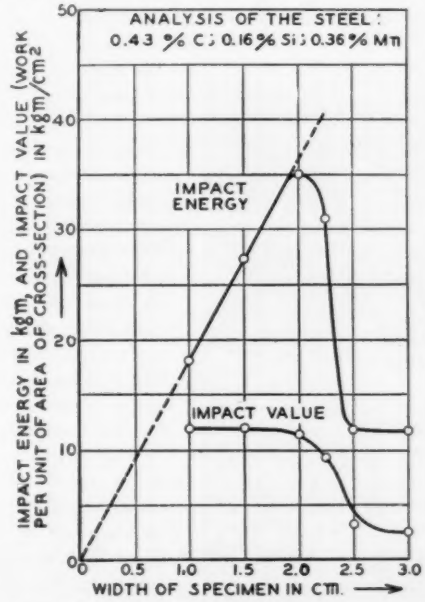


FIG. 5 IMPACT ENERGY AND IMPACT VALUES FOR VARIOUS WIDTHS OF SPECIMEN (MOSER)

(Material, medium-soft steel. Up to the point of critical specimen width, the impact energy increases proportionately with the width; thereafter the impact energy and impact values suddenly start to drop.)

clearly reveals the influence of the temperature prevailing during the test. At higher temperatures the impact values, as defined above, are high. When the temperature drops, the impact values decrease. The temperature at which the impact values start to decrease is different for the various structural conditions of a steel. As a consequence, the curve of impact value plotted against temperature will disclose whether the material, in so far as its structure is concerned, is in the most favorable or in a less desirable condition. In the latter case, the drop to low impact values will have already started at a higher temperature. Comparisons between different materials can be made in a similar manner by using this curve.

Evidently the statement just made applies only under the assumption that the two other factors which influence the strained condition of the material remain unchanged, that is, the shapes of the specimens and of the notches must remain the same in all cases. It should be emphasized that the influences of the three factors mentioned, i.e., test temperature, shape of specimen, and shape of notch, are not independent of one another; as a matter of fact, the influence of any one of these factors will be modified by the other two. For instance, in the temperature diagram, the temperature at which the drop in impact value occurs will also depend on the shape of specimen and notch. It will be shown later how this fact has been utilized in establishing a tentative standard specimen. The existence of such a relationship between

[7] W. Schwinning, "The Notched-Bar Test as a Criterion for the Properties of Materials," *Z.V.D.I.*, vol. 73 (1929), p. 321.

[8] F. László, "The Notched Bar," *Z.V.D.I.*, vol. 72 (1928), p. 851.

[9] R. Mailänder, "The Influence of Temperature, Shape of Specimen, and Test Speed on the Notched-Bar Impact Values of Iron and Steel," *Die Wärme*, vol. 48 (1925), p. 283.

the three influencing factors forces us to recognize that standardization of the notched-bar test is also badly needed from the standpoint of scientific research.

2 In Fig. 5 have been plotted the results of a series of tests made with specimens which had been given different values for the ratio between height and width of cross-section by changing the width of the specimens in steps, the height remaining unchanged.[10] The upper curve shows the relationship between the amount of energy absorbed during impact and the width of the specimen, and the lower curve the impact values or the work per unit of cross-section, calculated from the impact energy by dividing it by the area of the smallest cross-section of the bar. When the width of the specimen exceeds a certain critical value, the impact values suddenly start to decrease, exactly as in the case of the temperature curve. And here again the point at which the drop in impact value starts is different and characteristic for the various conditions of the structure as well as for various materials.

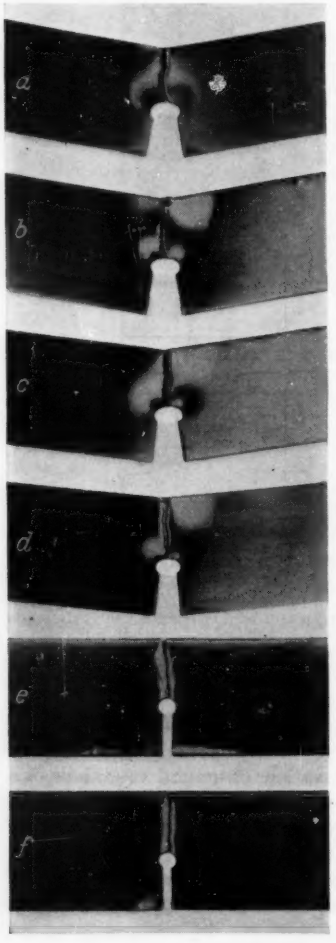


FIG. 6 DISAPPEARANCE OF THE REGION IN WHICH A PLASTIC FLOW OCCURS. CHANGE INTO A SEPARATION FAILURE
(Width of specimens in centimeters: *a*, 1; *b*, 1.5; *c*, 2; *d*, 2.25; *e*, 2.5; *f*, 3.)

The drop in impact value must be demonstrated by the appearance of the fracture of specimens that have been subjected to the impact test, i.e., the fracture must appear in the form of a transition from a sliding failure to a separation failure. This transition is very clearly shown by the polished specimens of Fig. 6; for this particular steel the transition occurred at a critical width of the specimen equal to 2.25 cm.

3 When polished specimens are used and the extent of the region in which plastic flow occurs is considered, it will be found that the deformation of a specimen under the influence of stresses will decrease up to the point where finally a separation failure is unavoidable, when the radius of the notch is made smaller and smaller, that is, when the sharpness of the notch is increased, all other test conditions remaining unchanged.[10]

VARIOUS DESIGNS OF SPECIMENS—TENTATIVE STANDARDS

It is recommended that the complete set of characteristic curves be determined, when materials must be tested for which the notched-bar characteristics are still unknown or when the influences of changing conditions must be investigated. The impact temperature curve, Fig. 4, is undoubtedly one of the most instructive curves of this set. But for the particular purpose that we are attempting to achieve by establishing a standard test, the procedure involved in determining the complete curves would require too much time and material. It should be possible to meet the specification of a practical notched-bar test by making tests at but one temperature, and it should not be necessary to determine the entire series of temperature curves in each individual case.

It appears logical to choose for this purpose exactly that temperature at which the material is still able to maintain its high impact value when its structure has the desired qualities, because in this case the impact value will immediately drop to its low point as soon as the qualities of the structure are impaired in any respect whatever. From a practical standpoint it would be most desirable if the temperature thus defined could be invariably the same for all grades of steel that will ever have to be subjected to this test, and the same as the room temperature. For it hardly needs to be emphasized that, in practice, difficulties and legal tangles can be avoided only if tests are made at room temperature.

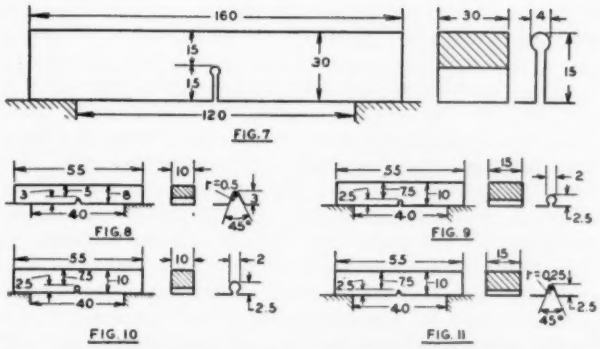
It was realized that it should be possible, by properly choosing the dimensional ratio of the cross-section of the specimen, and by giving the notch a suitable shape, to shift the almost vertically dropping parts of the temperature curves to such an extent that the drop in impact value would occur at practically room temperature. That this possibility exists is evidenced by what was stated in a previous paragraph regarding the existence of a relationship between the three factors influencing the strained condition of the specimen. However, it was recognized at the same time that in case the characteristics of the various grades of steel to be covered by the test should widely differ, it would hardly be possible to have the drop in impact value for all grades occur at room temperature by shifting all temperature curves in the same manner, that is, by using one standard specimen for all grades. For this reason it was thought that, although a single standard specimen would be sufficient for the majority of grades of steel to be covered by the test, it would eventually be necessary to adopt additional specimens for other grades not included in this majority. Nevertheless the standard specimen would have to be designed in such a manner as to make its field of application as broad as possible.

Finally, the problem to be dealt with by the committees took the following practical form:

A small standard specimen must be developed, on the basis of our present knowledge of this particular subject which has been described in the foregoing; and the shape of this specimen and the form of its notch must be such that, at room temperature, it will be possible to distinguish the various grades of steel and their structures in a satisfactory manner. When the adoption of subsidiary specimens should be necessary, their number should be restricted to a minimum.

The most important specimens that have been designed and tested by the two German committees in line with the foregoing considerations are listed in Table 3. The ratio between width and height of the cross-section was varied systematically, and both sharp and round notches were tried. Several series of tests were made with these specimens, employing all the various grades of steel and their different structures that are to be covered by this particular test, and it was found that several of the specimens could be eliminated from further consideration. The remaining designs are shown in Figs. 7–11, from which a comparison of their dimensions with those of the Charpy specimen, Fig. 7, can be readily made. The efficacy of the various specimens in producing the desired test results was determined by using the former standard specimen, i.e., the Charpy specimen with the dimensions 160 × 30 × 30 mm, as a basis of comparison. In

[10] M. Moser, "The Laws of the Notched-Bar Test," *Kruppsche Monatshefte*, vol. 2 (1921), p. 225.



FIGS. 7–11 DIMENSIONS OF VARIOUS SUITABLE DESIGNS OF SPECIMENS FOR THE NOTCHED-BAR TEST AS COMPARED WITH DIMENSIONS OF THE CHARPY SPECIMEN, FIG. 7
(All dimensions in millimeters.)

other words, all grades of steel and their various structures had also to be tested with the Charpy specimen.

TABLE 3 THE MOST IMPORTANT SMALL NOTCHED-BAR TEST SPECIMENS EXAMINED
(All dimensions in millimeters)

Ref. No.	Width	Height	Length	Notch diam.	Notch depth	Cross-section subjected to impact	Distance between supports
1	30	30	160	4	15	30 × 15	120
2	10	10	55	2	2.5	10 × 7.5	40
3	10	10	55	1	2.5	10 × 7.5	40
4	10	10	55	0.5	2.5	10 × 7.5	40
5	10	12	55	2	4.5	10 × 7.5	45
6	10	10	55	1	3	10 × 7	40
7	10	10	55	2	5	10 × 5	40
8	15	10	55	2	2.5	15 × 7.5	40
9	15	10	55	1	2.5	15 × 7.5	40
10	15	10	55	0.5	2.5	15 × 7.5	40
11	15	10	55	2	3	15 × 7	40
12	10	8	55	1	3	10 × 5	40

The question now was, Which of these specimens will best serve to positively identify the various grades of steel and their different structures in practice?

The answer was as follows: It was found that, of all the specimens examined, the Charpy specimen still differentiates most distinctly between the various steel grades and steel structures; at the same time it was found, however, that one of the newly designed specimens, i.e., the 10 × 10-mm specimen with the 2-mm round notch as represented in Fig. 10 (see Table 3, No. 2), was almost as efficient as the Charpy specimen. This may be seen by comparing the results obtained with some of those of a series of tests made by Dr. László (see Fig. 12 and Tables 4–6); these latter data serve at the same time to give an idea of the method of investigation followed by us.

A steel, the analysis of which is indicated in Table 4 (1 per cent nickel steel), had been subjected to eight different heat treatments, and consequently was available in eight different structural conditions. Table 6 gives particulars of the heat treatments. The eight different structures of the steel were subjected to the notched-bar test in the form of the specimens under investigation; details regarding three of these specimens are given in Table 5. Fig. 12 shows that the Charpy specimen and the 10 × 10-mm specimen with the 2-mm round notch (curve 2) of almost the same efficiency, produce a well-nigh perfect separation of the different structural conditions. It is seen that these conditions can be very clearly identified, the absolute values being sufficiently high.

After another very careful check of all factors entering into the problem, the decision was reached that the new specimen just referred to, i.e., No. 2 of Table 3, which had been found to be the best substitute for the Charpy specimen, should be proposed as a tentative German standard. At the same time, however, for several reasons, it was decided to increase the depth of the notch to 3 mm. One reason was that with the original notch depth of 2.5 mm the deformation would occasionally spread to a greater or less extent into regions on both sides of the notch, thus affecting the positiveness of the results. This difficulty was avoided when the depth of the notch was increased to 3 mm. This increase in depth will further lessen the influence of small discrepancies in the notch depth that can always occur when the specimens are made in a commercial shop, and thereby result in smaller discrepancies between the impact values.

However, it was found, as had been anticipated from previous knowledge of the subject, that the positive range of differentiation of this particular specimen does not include the very ductile steel grades. These can be differentiated only when a specimen

TABLE 4 MATERIAL SUBJECTED TO TESTS

Material: 1% Nickel steel; Ac_1 at 780 °C; Ar_1 at 710 °C

Analysis (per cent):	C	Si	Mn	P	S	Ni	Cr
	0.36	0.25	0.77	0.020	0.017	1.04	0.14%

TABLE 5 NOTCHED-BAR TEST SPECIMENS

Specimen	Cross-section, sq mm	Notch diam., mm	Cross-section subjected to impact, sq mm
(1) Charpy	30 × 30	4	30 × 15
(2) New	10 × 10	2	10 × 7.5
(3) New	15 × 12	sharp 45° r = 0.5 mm	15 × 7.5

TABLE 6 HEAT TREATMENT AND APPEARANCE OF FRACTURE

Condition	Temp. at which specimen was held 4 hr, deg C	Speed of cooling process: Avg. down to 20°, °C/min	Maximum Range, °C	Maximum °C/min	Appearance of fracture (du. = ductile) (br. = brittle): 30 × 30	10 × 10	15 × 12
a	None	..	.....	...	1/2 du.	3/4 du.	3/4 du.
b	725	0.80	600– 430	2.2	1/2 du.	3/4 du.	1/2 du.
c	850	0.80	850– 815	3.7	1/4 du.	1/2 du	1/2 du.
d	950	0.63	715– 610	2.2	1/4 du.	1/4 du.	1/4 du.
e	1000	0.67	785– 635	2.7	1/4 du.	1/4 du.	1/4 du.
f	1050	0.75	875– 785	3.6	br.	br.	br.
g	1150	0.80	830– 760	3.5	br.	br.	br.
h	1250	0.83	1090–1040	5.0	br.	br.	br.

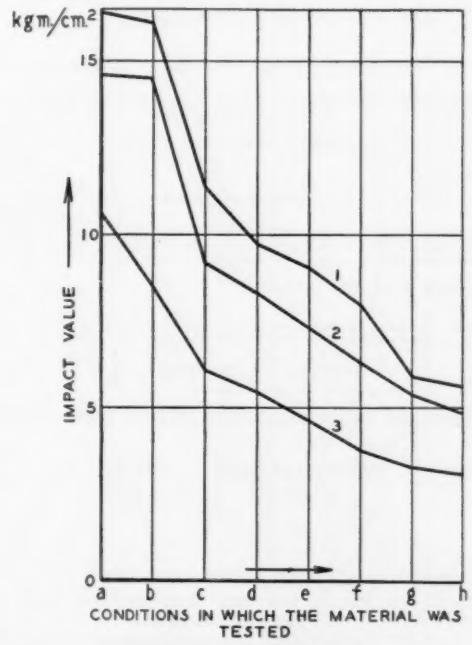


FIG. 12 EXAMPLE OF A COMPARISON OF IMPACT TESTS MADE WITH THE CHARPY SPECIMEN AND TWO NEWLY DESIGNED SPECIMENS (LASZLO)
(For data on analysis and conditions of material and shape of specimens, see Tables 4–6.)

with a sharp notch is used. As a consequence, it was found necessary to adopt, in addition to the main specimen with round notch to be proposed as a tentative standard, a subsidiary tentative standard with a sharp notch. In the author's opinion, the adoption of this additional specimen would be sufficient to cover the entire range.

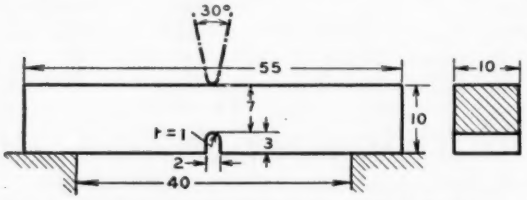


FIG. 13:- SMALL STANDARD PROPOSED BY DVM (GERMAN ASSOCIATION FOR TESTING MATERIALS).

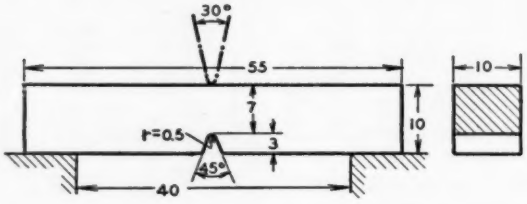


FIG. 14:- ADDITIONAL SMALL STANDARD PROPOSED BY DVM.

FIGS. 13 AND 14 PROPOSED GERMAN STANDARDS FOR NOTCHED-BAR TEST SPECIMENS
(All dimensions in millimeters.)

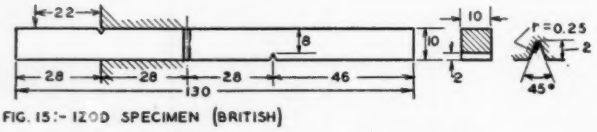


FIG. 15:- IZOD SPECIMEN (BRITISH)

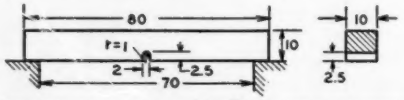


FIG. 16:- SPECIMEN OF THE ASSOCIATION OF SWISS MACHINE COMPANIES

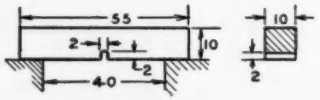


FIG. 17:- MESNAGER SPECIMEN. (FRANCE AND ITALY)

FIGS. 15–17 SPECIMENS FOR THE NOTCHED-BAR TEST USED IN VARIOUS COUNTRIES
(All dimensions in millimeters.)

The new tentative German standard specimen, to be designated as "Small Standard Proposed by the DVM (German Association for Testing Materials)," is shown in Fig. 13. No restrictions are placed on the method of machining the notch, that is, whether it must be drilled, milled, or ground. It is important, however, to avoid any cross-scratches, and when these appear, they must be removed. Fig. 14 shows the subsidiary tentative standard specimen, with the sharp notch, to be designated as "Additional Small Standard Proposed by the DVM." The specified radius of 0.5 mm at the bottom of the notch will facilitate the manufacture of this standard specimen in the shop. It has been found from experiments that the exact form of the notch can no longer be guaranteed when a smaller radius is employed.

It is interesting to note that the dimensions of the Charpy specimen and of the proposed tentative standards are such that the two halves of a Charpy specimen which has been subjected to the impact test, can be used afterward for making small DVM specimens. This means that it is relatively simple to make comparative series of tests with both specimens.

Conclusions

The small specimen, Fig. 13, and its supplementary specimen, Fig. 14, constitute the two tentative standards that will be submitted later to the German producing and consuming interests by the German Association for Testing Materials.

In the future it is intended to have all statements relating to impact values based on this standard and on the subsidiary standard specimen, on the latter in those cases where the toughness of the material to be tested is so great that the round-notch specimen would not lead to conclusive results.

In a few exceptional cases, the standard specimens will not apply—for instance, when the material and the form in which it is delivered are such that the specimens used at present must be retained as special specimens. An example of such a special case is cast steel. Here it will be more desirable to use the old Charpy specimen, because, on account of its larger size, it contains a larger amount of the coarse-crystalline material. And, for similar reasons, when we must test materials which are manufactured in the form of plates, we cannot, at least not at the present time, do without the customary specimens now in use.

But nevertheless, with the introduction of the standard specimens described in the foregoing, a uniform language, which will not leave the slightest doubt as to its true meaning, will become available to those who must deal with scientific and practical problems relating to the properties of materials as revealed by notched-bar tests. And this is certainly an important step forward!

The accomplishments of the Congress at Zurich are such that an international agreement looking toward the establishment of a standard specimen does not seem unlikely.

Several specimens for notched-bar testing, much used in other countries, and differing but slightly from the tentative German standard and subsidiary standard specimens, are reproduced in Figs. 15–17.

APM-55-15

Theories of Strength

By A. NADAI,[1] EAST PITTSBURGH, PA.

The conditions under which yielding starts at lower temperatures in metallic and also in other materials are discussed. Recent tests, particularly those made in England, Germany, Switzerland, and the United States, are compared and the results of a number of new ones concerning the conditions of yielding of steel are reported.

More detailed reference to these tests is made, and conclusions as to the best expressions available at present are drawn from them and utilized in rules for computing working stresses in practical cases. In conclusion, certain suggestions are made as to the conditions of ordinary rupture in a static test.

I—INTRODUCTION



AT THE suggestion of the Applied Mechanics Division, a brief review of the so-called theories of strength is offered in what follows. The discussion will be confined mainly to two principal questions which may be raised in connection with the various possibilities under which failure in general may be observed in engineering materials. No attempt will be made to define the term "failure" with greater accuracy, because of the great variety of factors which would have to be considered if the conditions should be described under which the safety of a given construction member has to be determined. The problems to be discussed in the following may perhaps be reduced to two: (1) What are the mechanical conditions, if any exist, under which engineering materials start to deform permanently, that is, under which they start to yield plastically? And (2) what are the corresponding conditions under which rupture or visible separation of the parts of a construction member occurs? However, before entering into a detailed discussion of the answers which at the present state of our knowledge may be advanced concerning these questions, a few experimental facts about the behavior of materials under stress must be briefly recalled.

If a solid body is subjected to a homogeneous state of stress such as pure tension, compression, or pure shear, the strains produced in the body will be partly elastic and partly plastic. There are materials, such as mild steel, glass, etc., which at a sufficiently low temperature behave perfectly elastically within a certain range of stress. They deform slightly under stress, but the deformations disappear completely if the loads are removed. In other cases, however, no range of stress can be detected within which an initially unstressed material would behave perfectly elastically. If a stress which has been acting for a certain time and has produced the strain ϵ (point A, Fig. 1) is removed, the material will exhibit a certain strain ϵ'' (point B). The portion ϵ' of ϵ may be called the elastic, and the portion ϵ'' the permanent, part of the strain ϵ. The curves $\sigma = f(\epsilon)$ for loading and for unloading will in general not coincide, and if unloading and loading are repeated between the limits of stress σ (point A) and the stress zero (point B), in general a loop of elastic hysteresis will be observed in the corresponding "stress-strain curves," such as the one indicated in Fig. 1 between the curves $A2B$ and $B3A$. If the stresses are increased above the stress σ (branch AC), the curve AC will continue about in the direction of the extension of the first branch OA. These details may be characteristic for the shape of the "stress-strain curves" for initial loading in certain engineering materials. If the initial stress-strain curve is of the shape shown in Fig. 2, with a distinct elastic line OA and with permanent deformations starting quite suddenly as soon as the stress $\sigma = \sigma_0$ is reached (point A), the material has a sharply defined "yield point" or a sharp limit of plasticity σ_0 under which the material starts to deform permanently or plastically (line AB). In this case an elastic

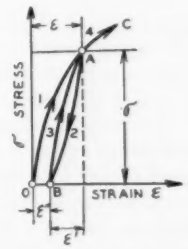


FIG. 1 :- ELASTIC AND PERMANENT STRAIN

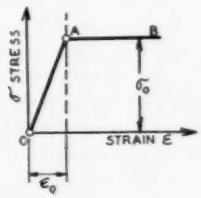


FIG. 2 :- SHARPLY DEFINED YIELD STRESS

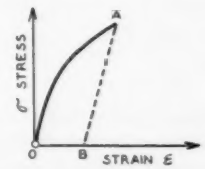


FIG. 3 :- STRESS-STRAIN DIAGRAM WITH GRADUAL TRANSITION CURVE

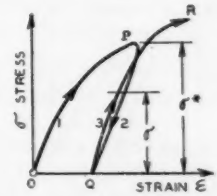


FIG. 4 :- DEFINITION OF YIELD STRESS, ASSUMING STRAINHARDENING

range $0 < \epsilon < \epsilon_0$ of the strains may well be distinguished from the plastic range of strains $\epsilon > \epsilon_0$. In other cases the transition from a purely elastic initial stage into the plastic range is less sharply marked, as, for example, in the case of the stress-strain curve shown in Fig. 3. Is there any justification in speaking of a yield stress also in this case, and if so, what stress, if any, can be called the "yield stress?" The answer will be given by interrupting the test at a point P (Fig. 4). If the material were to be unloaded from the stress σ^* and again loaded in the same manner but to a stress σ not exceeding σ^*, only small elastic changes of the strains would be noticeable, corresponding to the curves $P2Q$ and $Q3P$. If, however, $\sigma \geqq \sigma^*$, then yielding starts along PR again. For this reason σ^* may be called the new "yield stress." In most of the polycrystalline metallic materials, the yield stresses σ^* producing new permanent plastic deformations increase with ϵ (strain hardening). A stable condition of yielding requires always that $d\sigma/d\epsilon > 0$. In materials exhibiting

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[1] Research Laboratories of the Westinghouse Elec. & Mfg. Co. Mem. A.S.M.E.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

considerable strain hardening, an elastic range of the strains may be distinguished; they behave more or less elastically after the yielding is interrupted and provided that the subsequently applied stresses σ remain smaller than the previous yield stress ($\sigma < \sigma^*$, curve $P2Q$ or $Q3P$ in Fig. 4).

In what follows it will furthermore be useful to distinguish between (*a*) brittle, (*b*) tough, and (*c*) ductile materials. (Fig. 5.) This refers to the shapes of the stress-strain curves of the materials in pure tension. A brittle material will not exhibit much permanent deformation until rupture occurs (glass, cast iron); a tough material will show a considerable increase of stress while it deforms to a few per cent elongation (hard steels), and a ductile material can be stretched 20 to 40 per cent or more before it breaks (soft copper). It is, however, important to note that this distinction is an artificial one, because materials which under ordinary conditions are brittle, under other circumstances may be brought into a tough or even a ductile condition. This can be done by repeating the tensile or the compression test and by applying a heavy hydraulic pressure at the same time on the bounding surfaces of the test piece; and under such circumstances brittle materials such as marble have been deformed considerably in plastic distortion. Another example is the plastic impression obtained in cast iron under a hard steel ball, although cast iron is a brittle material.

For this and other reasons, not explained here, it is perhaps more correct not to speak of brittle or of ductile materials, but rather of the brittle and the plastic states of a given material.

When speaking of the conditions under which yielding as just defined (including strain or work hardening) starts, it is furthermore assumed that the stresses are applied slowly and at normal temperatures. It is known that the speed of deformation has an influence on the magnitude of the yield stress producing a given strain ϵ such that a bar of a ductile metal stretches permanently. Recent tests,[2] in which the speed of stretching was varied in the ratio of 1 to 1,000,000, seem to confirm results obtained earlier by Cassebaum and P. Ludwik that for the ductile metals the yield stress σ which is required to produce a *given* elongation ϵ at normal temperatures increases with the rate of elongation $d\epsilon/dt$ according to a logarithmic expression of the form

$$\sigma = \sigma_1 + \sigma_2 \log \frac{v}{v_0}$$

where σ_1, σ_2, and v_0 are material constants and $v = d\epsilon/dt$. These latter changes will not amount to more than a few per cent under ordinary testing conditions, and the stress conditions discussed below therefore apply to ordinary testing speeds but not to impact or in cases of very rapid plastic stretching. The second and more important restriction to the use of the criteria to be discussed refers, as has already been mentioned, to the temperature. It is assumed that the temperatures are sufficiently low not to cause gradual creep and very slow plastic distortions, which, under continued application of a load, could become appreciable even under comparatively small stresses. All such slow permanent deformations which depend on time, that is, the phenomena of creep or of recrystallization as effected by elevated temperatures, are excluded.

As far as the phenomena of rupture are concerned, only stationary or prolonged cases of loading are assumed, and particularly failure due to fatigue is excluded, this case being treated by another paper of the present symposium. The observations made below refer only to polycrystalline materials. Although single crystals have exhibited some of the phenomena of yielding and rupture in a more perfect way than polycrystalline materials, their low strength properties make it unnecessary to deal with them separately here. Special reference should, however, be made to the extensive experimental work by means of which the mechanical properties of single metal crystals have been established in recent years, and which has contributed so much to a better understanding of the mechanism of yielding, and of rupture also, in the polycrystalline metals.

A homogeneous state of stress is determined by six quantities, for example, the three normal stresses σ_x, σ_y, σ_z, and the three components of shearing stress τ_{xy}, τ_{xy}, τ_{zx} acting in the faces of a small parallelopiped cut out of the stressed material. Instead of using these six components, however, it is more convenient to deal with the principal stresses σ_1, σ_2, σ_3, and to assume that the three mutually perpendicular directions are given in which they act. In a similar way the state of strain may be given either by the unit strains ϵ_x, ϵ_y, ϵ_z, and the unit shears γ_{xy}, γ_{yz}, γ_{zx}, or by the three principal strains ϵ_1, ϵ_2, ϵ_3, and by their directions. When speaking of strains it is assumed that they are small quantities compared with unity and that reference is made only to the permanent strains. It is assumed also that, while the material yields, the directions of the principal stresses do not *rotate* with respect to the body itself. This limits the discussion to certain types of loads or modes of plastic yielding, and makes it possible to neglect the elastic parts of strains at first. A more general discussion of the conditions of yielding, and particularly of the transition curves from an elastic state of strain to a plastic, continuously changing state of strain with both elastic and plastic parts, would require the abandonment of one of the assumptions made implicitly, namely, that both the tensor of the elastic strains and that of the permanent strains during a transition curve are two *coaxial* tensors.

II—EXPERIMENTAL DETERMINATION OF THE CONDITION OF YIELDING

In this section will be described briefly the method which has been used by the author and his collaborator, Dr. W. Lode, in their tests (1925–1928)[3] to find the stress relation which seems to express the condition of yielding in ductile metals.[4] A principal consequence of the assumptions made above is that yielding in the ductile metals may be considered as depending on a relation or expression in which the principal stresses σ_1, σ_2, σ_3 appear only as variables. Previous tests with metallic materials, as well as those made on single metal crystals, have shown that if a system of principal stresses σ_1, σ_2, σ_3 is at a limit of plasticity, by the addition or subtraction of a three-axial uniform tensile stress or of a hydraulic compression σ^*, a system $\sigma_1 - \sigma^*$, $\sigma_2 - \sigma^*$, $\sigma_3 - \sigma^*$ is obtained which is also at the limit of plasticity. Hence yielding can depend only on the principal-stress differences $\sigma_1 - \sigma_2$, $\sigma_2 - \sigma_3$, and $\sigma_3 - \sigma_1$, and, at least under not too large an average or mean pressure $\sigma = (\sigma_1 + \sigma_2 + \sigma_3)/3$, yielding is independent of the mean pressure. Until now this could obviously be established only under comparatively low values of σ; it is true mainly for the ductile metals, while the

[2] Deutler, *Phys. Zeit.*, 1932, p. 347.

[3] Cf. the author's papers: "Zur Mechanik der bildsamen Formänderungen," Werkstoff ausschussbericht 56 des Vereines deutscher Eisenhüttenleute, Verlag, *Stahl u. Eisen*, Düsseldorf, 1925; and "Versuche über die Fliessgrenze des Eisens," Proc. 2 Intern. Congress for Applied Mechanics, Zurich, 1926. Also, W. Lode: "Versuche über den Einfluss der mittleren Hauptspannung auf das Fliessen der Metalle Eisen, Kupfer und Nickel," *Zeit. für Physik*, Bd. 36, p. 913–939, 1926; and "Der Einfluss der mittleren Hauptspannung auf das Fliessen der Metalle," Forschungsarbeiten des Vereines deutscher Ingenieure, Berlin, Heft 303, 1928.

[4] As a bibliography of the tests made under combined stress previous to these tests, with a critical discussion of their results, is contained in Dr. Lode's second paper mentioned in the previous footnote, further reference to the earlier work on the subject will be omitted here.

yielding of other polycrystalline materials, particularly of those commonly called brittle materials, may depend on σ.

With reference to the transition part of a stress-strain curve from the elastic to a plastic state, particularly when repeated loading and unloading curves are compared, it is worthy of note that in a tensile test with mild steel an almost perfect straight line is obtained as the unloading curve (CD in Fig. 6) and that this will also coincide with the lower portion of the succeeding loading curve if the yielding is suddenly interrupted. If the unloading occurs very slowly on account of some afterflow, the corner at C will be somewhat rounded, but later the unloading curve will still be nearly a perfectly straight line, as CD.

Dr. Lode and G. J. Taylor and H. Quinney[5] have shown that a test piece of a ductile metal, such as copper, aluminum, or iron, which has been stretched in pure tension with frequent interruptions, furnishes a diagram similar to that shown in Fig. 7. Each subsequent loading curve leaves the straight line B_1A_1, B_2A_2, B_3A_3 at a certain point C_1, C_2, C_3, and bends more or less gradually into a new direction which closely coincides with a curve passing through the points A_1, A_2, A_3. The same curve would have been obtained, if the test piece had been stretched continuously and without interruption in a single test.

In test pieces in the form of tubes having not too small or great a wall thickness, distributions of stress may be produced by means of internal pressure and axial loads in which two of the three principal stresses σ_1, σ_2, σ_3 are tensile stresses. As the third principal stress in such a tube is either zero or very nearly equal thereto, distributions of stress can be obtained in which

$$\sigma_1 \geqq \sigma_2 \geqq \sigma_3 = 0$$

Thus the intermediate principal stress σ_2 may be chosen equal to any value between $\sigma_1 \geqq \sigma_2 \geqq 0$, and the influence of the intermediate principal stress σ_2 on the yielding be determined. If such a tube is first tested in pure tension in the axial direction, then, by a combination of internal hydraulic pressure and axial tension, similar curves to those shown in Fig. 7 can be recorded. In obtaining these diagrams it is assumed that the principal stresses in the hollow cylinder increase according to the proportion $\sigma_1 : \sigma_2 = \sigma_1' : \sigma_2'$. The points A_1, A_2, A_3 . . . obtained for a set of consecutive tests made on the same tube under a number of different distributions of stress applicable to the tube may now serve to determine the condition of yielding (Fig. 8). For example, if a hollow cylinder is subjected first to pure (axial) tension σ_1, $\sigma_2 = \sigma_3 = 0$, then unloaded and again stressed by an axial load and in addition by an internal pressure until it yields, and such tests with various loads and pressures are repeated, curves for the axial principal stress σ_1 as a function of the axial principal strain ϵ_1 will be obtained such as those shown in Fig. 8. Each of the subsequent curves C_1A_2, C_2A_3, C_3A_4 . . . will correspond to another principal stress system σ_1, σ_2, σ_3, and the height of the new points A_1, A_2 . . . may now be compared with the ordinates of the projected curve A_1, A_3 . . . shown by the broken line, which line would be the stress-strain curve for pure tension.

In this manner Dr. Lode and the author in their tests have eliminated the influence which the strain hardening of a ductile metal must have on the condition of yielding. Using a number of mild steel, copper, and nickel tubes, all of which in their original condition were annealed, the following test results were obtained:

1 The condition of yielding for the ductile metals can, within errors of about 3 per cent, be expressed by the equation:

$$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 = 2\sigma_0^2 = \text{const.} \ldots [1]$$

[5] "The Plastic Distortion of Metals," Phil. Trans. Roy. Soc., London, Series A, vol. 230.

first suggested by R. von Mises[6] and later by H. Hencky.[7] Here σ_0 is the yield stress for pure tension in the strain-hardened condition of the material. This equation, expressing the condition for uniform yielding, was checked within about 10 per cent of the total axial extension and verified, with the exception of the first yielding in the case of mild steel.

It is known that the characteristic phenomenon of the upper and lower yield points for the first yielding in tensile tests of mild steel may be observed on smooth cylinders or tubes which have heads with very gradually changing cross-sections. In such bars or tubes an instability of the static equilibrium develops until the first signs of yielding become visible. The load then drops suddenly, while at the same time yielding starts in a narrow oblique layer, this latter becoming visible as a strain or flow figure on the surface of the bar. As the "upper yield stress" of mild steel is a stress under which the bar is in an unstable equilibrium, such a limit has no greater practical significance as a limit of plasticity. It is therefore important, as far as the behavior of mild steel is concerned, not to include tensile tests in which an upper yield point has been observed in the comparison and which refer to the first yielding. Various discrepancies in the interpretation of the results of former tests made under combined stresses may probably be due to this cause. As soon as the lower yield point is observed or when uniform yielding starts in test pieces of mild steel, Equation [1] is found to agree with the results of the tests.

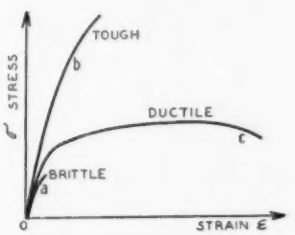


FIG. 5 :- STRESS-STRAIN CURVES FOR TOUGH, DUCTILE, AND BRITTLE MATERIALS

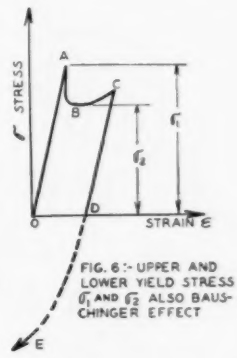


FIG. 6 :- UPPER AND LOWER YIELD STRESS σ_1 AND σ_2 ALSO BAUSCHINGER EFFECT

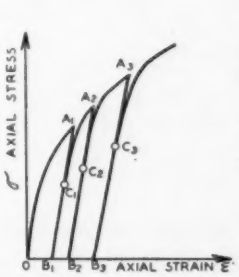


FIG. 7 :- STRESS-STRAIN CURVE FOR TENSILE TESTS WITH INTERRUPTIONS

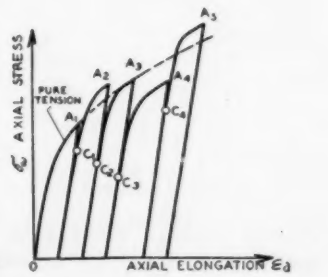


FIG. 8 :- STRESS-STRAIN CURVE FOR TESTS WITH TUBE SUBJECTED TO COMBINED ACTION OF VARIOUS AXIAL LOADS P AND INTERNAL PRESSURES p

[6] Mechanik der festen Körper im plastisch deformablen Zustand," *Göttinger Nachrichten*, 1913.

[7] "Zur Theorie plastischer Deformationen, etc.," *Zeit. für angew. Math. u. Mech.*, vol. 4 (1924), p. 323; "Uber das Wesen. der plastischen Verformung," *Z.V.D.I.*, vol. 69 (1925), p. 695. In the discussion of this condition, Equation [1], at the 1st International Congress for Applied Mechanics in Delft (1924) it developed that Prof. M. J. Huber had also independently suggested the same equation as the condition of yielding.

2 For the *infinitesimal* permanent parts of the resulting deformations, the following three laws express the mode of yielding:

a The volume remains constant:

$$\epsilon_1 + \epsilon_2 + \epsilon_3 = 0 \quad \ldots\ldots [2]$$

b The axes of the three principal strains coincide with the axes of the three principal stresses σ_1, σ_2, σ_3.

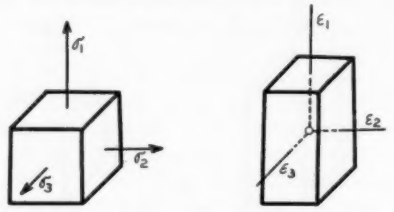


FIG. 9 :- PRINCIPAL STRESSES $\sigma_1, \sigma_2, \sigma_3$ AND PRINCIPAL PLASTIC STRAINS $\epsilon_1 \epsilon_2 \epsilon_3$.

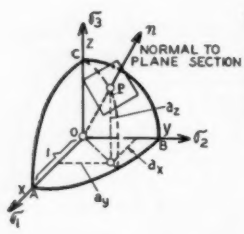


FIG. 10

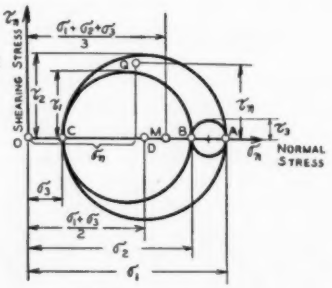


FIG. 11

MOHR'S REPRESENTATION OF STRESS

c Rule of yielding: Although in the tests of Lode certain deviations from the law expressed by Equation [3] were clearly observable, for practical purposes it can be assumed that if the strains ϵ_1, ϵ_2, ϵ_3 change uniformly,

$$\frac{\epsilon_1 - \epsilon_2}{\sigma_1 - \sigma_2} = \frac{\epsilon_2 - \epsilon_3}{\sigma_2 - \sigma_3} = \frac{\epsilon_3 - \epsilon_1}{\sigma_3 - \sigma_1} \quad \ldots\ldots [3]$$

Since

$$\gamma_1 = \epsilon_2 - \epsilon_3, \quad \gamma_2 = \epsilon_3 - \epsilon_1, \quad \gamma_3 = \epsilon_1 - \epsilon_2 \quad \ldots\ldots [4]$$

are the principal shearing strains and

$$\tau_1 = \frac{\sigma_2 - \sigma_3}{2}, \quad \tau_2 = \frac{\sigma_3 - \sigma_1}{2}, \quad \tau_3 = \frac{\sigma_1 - \sigma_2}{2} \quad \ldots\ldots [5]$$

are the principal shearing stresses, Equation [3] is identical with

$$\frac{\gamma_1}{\tau_1} = \frac{\gamma_2}{\tau_2} = \frac{\gamma_3}{\tau_3} \quad \ldots\ldots [6]$$

or, expressed in words, when a metal yields, the principal shears change in proportion to the principal shearing stresses which cause the plastic distortion.

Introducing the principal shearing stresses τ_1, τ_2, τ_3 and the maximum shearing stress $k = \sigma_0/2$ for yielding in pure tension, the condition of plasticity, Equation [1], can be rewritten in the form:

$$\tau_1^2 + \tau_2^2 + \tau_3^2 = 2k^2 = \text{const} \quad \ldots\ldots [7]$$

Another convenient way of expressing the meaning of Equations [2] and [3] suggested by Dr. Lode and the author utilizes Mohr's representation of stress by means of the three principal stress and strain circles. In an octant of a small sphere of material (Fig. 10), the edges x, y, z of which are parallel to the principal stress directions, any point P on the surface represents the position of a plane section in the stressed material. If the radius of the sphere is taken equal to unity, then the direction cosines, a_x, a_y, a_z of the radius OP or of the normal n to the tangent plane of the sphere are the rectangular coordinates of the point P. In Mohr's plane (Fig. 11) a second point Q corresponds to any point P in Fig. 10, the rectangular coordinates of which are equal to the normal stress σ_n and the shearing stress τ_n acting in the plane section with the normal n. The three principal circles in the octant of the sphere, AB, BC, CA in Fig. 10, correspond to the three principal circles of stress indicated by the same letters in Fig. 11. The abscissas of the three points A, B, C in Fig. 11 determine the principal stresses σ_1, σ_2, σ_3, and the radii of these three principal circles determine the principal shearing stresses τ_1, τ_2, τ_3. If now a variable μ is introduced for the ratio of the lengths DB and $AD = DC$, then

$$\mu = \frac{BD}{AD} = \frac{\sigma_2 - \frac{\sigma_1 + \sigma_3}{2}}{\frac{\sigma_1 - \sigma_3}{2}} = \frac{2\sigma_2 - \sigma_1 - \sigma_3}{\sigma_1 - \sigma_3} \quad \ldots\ldots [8]$$

determines the intermediate principal stress σ_2 provided that $\sigma_1 > \sigma_2 > \sigma_3$. The point D is the center of the largest of the three principal-stress circles.

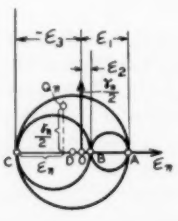


FIG. 12 :- PRINCIPAL STRAIN CIRCLES WHEN $\epsilon_1+\epsilon_2+\epsilon_3=0$

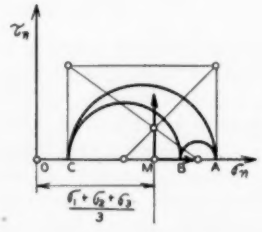


FIG. 13 :- CONSTRUCTION OF ORIGIN M (WHICH IS CENTER OF GRAVITY OF POINTS ABC, ASSUMING EQUAL MASSES)

As yielding in the case of ductile metals does not depend appreciably on the mean pressure $\sigma = (\sigma_1 + \sigma_2 + \sigma_3)/3$, it is frequently convenient to introduce in Fig. 11 a new origin M which is obtained by making $OM = (\sigma_1 + \sigma_2 + \sigma_3)/3 = \sigma$. Obviously M is the center of gravity of the three points A, B, C, assuming equal masses concentrated in them, and can be constructed as indicated in Fig. 13. The principal stresses for this new origin M are then

$$\sigma_1^* = \sigma_1 - \sigma, \quad \sigma_2^* = \sigma_2 - \sigma, \quad \sigma_3^* = \sigma_3 - \sigma$$

In a similar way the state of plastic strain ϵ_x, ϵ_y, ϵ_z, γ_{xy}, γ_{yz}, γ_{zx} may be represented by Mohr's circles. If ϵ_1, ϵ_2, ϵ_3 are the principal strains, then Mohr's circles in Fig. 12 represent the state of strain. As for the state of plastic strain,

$$\epsilon_1 + \epsilon_2 + \epsilon_3 = 0$$

the origin O of the three principal plastic-strain circles must always coincide with the center of gravity of the points A, B, C on the ϵ-axis. The ordinates in the strain diagram, Fig. 12, are equal to $\gamma_n/2$.

In quite an analogous way a second variable ν referring to the principal strains ϵ_1, ϵ_2, ϵ_3 and determining the mean or intermediate plastic principal strains ϵ_2 may now be introduced by defining ν as

$$\nu = \frac{\epsilon_2 - \frac{\epsilon_1 + \epsilon_3}{2}}{\frac{\epsilon_1 - \epsilon_3}{2}} = \frac{2\epsilon_2 - \epsilon_1 - \epsilon_3}{\epsilon_1 - \epsilon_3} \quad \ldots\ldots [9]$$

The third law of plastic yielding can then be expressed by the equation

$$\mu = \nu \quad \ldots\ldots [10]$$

which indicates that a plastic state of stress produces such strains that the principal-strain circles (Fig. 12) consist of a group of circles similar to the principal-stress circles (Fig. 11 or 13). Both sets of circles are furthermore oriented similarly to their respective origins M (Fig. 11) and O (Fig. 12).

Two points Q and Q_n which are chosen in each of these two figures in a similar position determine by their abscissas and ordinates the normal and shearing stress σ_n, τ, and the unit elongation ϵ_n and half of the shear $\gamma_n/2$, respectively corresponding to the same normal n in Fig. 10. As according to Equation [6] for the principal shears

$$\tau_1 = c\gamma_1, \quad \tau_2 = c\gamma_2, \quad \tau_3 = c\gamma_3 \quad \ldots\ldots [11]$$

then quite generally for any direction of a given n section,

$$\tau_n = c\gamma_n \quad \ldots\ldots [12]$$

This shows that plastic slip or shearing strains occur naturally not only in one, two, or all the three pairs of planes in which the principal shearing stresses τ_1, τ_2, τ_3 act, but *on all planes oblique* to the principal-strain directions. This may be mentioned here, because some authors have been inclined to assume "that slip or gliding occurs in the planes of maximum shear," while others assume that it occurs "in the planes of maximum and minimum shear."[8] As, on the other hand, visible signs of initial plastic distortion can frequently be detected on the surfaces of test pieces of mild steel along thin layers which nearly coincide with a pair of the principal shearing planes, it is evident that here two quite different things are frequently confused in such statements as those just quoted, namely, *simple slip or shear* as one of the six components of a uniform strain or of a tensor—and *the temporary localization of a shear* or of a *plastic strain in a thin layer throughout the body*. In the former sense and according to Equation [12], plastic slip occurs in *all* oblique planes. In the second sense it refers to a temporary infinitesimal tangential movement of a certain part of the body as a rigid whole with respect to the remainder of the body, during which both portions are separated by a thin plastic layer. (Strain or flow figure, Lüders line.)

It is perhaps noteworthy that the first and second invariants of a stress tensor have a significance for the laws of the yielding of perfectly plastic materials. The first invariant,

$$\sigma = \frac{\sigma_1 + \sigma_2 + \sigma_3}{3}$$

determines the origin M for the three principal-stress circles. A second invariant expressing the von Mises-Hencky condition of yielding for the ductile metals may also be interpreted as a certain invariant of a tensor using scalar multiplications.

III—USING OCTAHEDRAL SHEARING STRESS IN PLASTICITY CONDITIONS

If one computes the stress components σ_n and τ_n for a special oblique plane the normal n of which makes equal angles of 54° 44′ with the three principal stress directions and the principal stresses are referred to the origin M [taken as $\sigma_1^* = \sigma_1 - \sigma$, $\sigma_2^* = \sigma_2 - \sigma$, $\sigma_3^* = \sigma_3 - \sigma$, where $\sigma = (\sigma_1 + \sigma_2 + \sigma_3)/3$] so that

$$\sigma_1^* + \sigma_2^* + \sigma_3^* = 0$$

then the normal stress σ_n^* in this oblique plane vanishes. The normal stress σ_n in this plane is accordingly given by $\sigma_n = (\sigma_1 + \sigma_2 + \sigma_3)/3$ for any stress distribution.

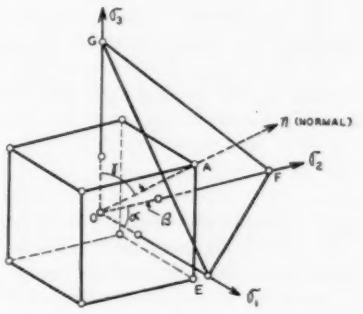


FIG 14.

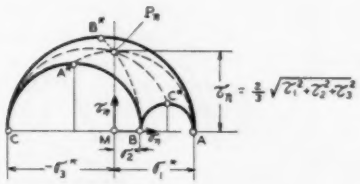


FIG 15 - CONSTRUCTION OF SHEAR STRESS τ_n

If one orients a cube with the faces according to the principal-stress axes, then there are eight planes which have a similar orientation with respect to the cube as the oblique plane, the normal of which bisects the first octant. They round off the eight corners of the cube and are the planes of an octahedron coaxial with the cube. The shearing stress τ_n in the octahedral plane *EFG*, Fig. 14, having for its normal the line *OA* or n, with the direction cosines $\cos\alpha = \cos\beta = \cos\gamma = 1/\sqrt{3}$, is given by

$$\tau_n^2 = \frac{\sigma_1^2 + \sigma_2^2 + \sigma_3^2}{3} - \frac{(\sigma_1 + \sigma_2 + \sigma_3)^2}{9} \quad \ldots\ldots [13]$$

or

$$\tau_n^2 = \frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{9} \quad \ldots\ldots [13a]$$

This shows that evidently in an octahedral plane (as *EFG*, Fig. 14)

$$\tau_n = \frac{2}{3}\sqrt{\tau_1^2 + \tau_2^2 + \tau_3^2} \quad \ldots\ldots [14]$$

or according to the condition of yielding (Equation [7]) in a plastic state, the shearing stress τ_n in an octahedral plane is

$$\tau_n = \frac{2\sqrt{2}}{3}k = \frac{\sqrt{2}}{3}\sigma_0 = 0.47\sigma_0 = \text{const} \quad \ldots\ldots [15]$$

To have a short designation for the physical facts underlying Equation [15], the author suggests calling Equation [15] the *condition of constant octahedral shearing stress* $\tau_n = const.$ It is

[8] In a paper by M. Ros and A. Eichinger: "Versuche zur Klärung der Frage der Bruchgefahr," Eidgen. Material prüfungsanstalt, Zürich, Sept., 1926, for example, it is said on page 10 (and italicized) that "the permanent deformations occur through slip in slip planes, which coincide with the planes of the maximum and minimum shearing stresses."

seen, therefore, that yielding in a perfectly plastic material according to *the theory of constant octahedral shearing stress* would start when the shearing stress τ_n in any one of the eight octahedral planes reaches a certain value (which, according to Equation [15], is equal to $\sqrt{2} \cdot \sigma_0/3 = 0.47\,\sigma_0$, where σ_0 is the yield stress for pure tension), the octahedron being oriented coaxially with the principal stress directions.

In Mohr's representation of stress a point P_n is easily found which furnishes the shearing stress τ_n. One has only to draw any one of the three bisecting circles AA^*, BB^*, CC^* indicated by broken lines in Fig. 15. Their common intersection or the intersection of one of them with the ordinate axis furnishes the point P_n and the ordinate $MP_n = \tau_n$ determines the shearing stress τ_n acting in one of the eight octahedral planes. The normal stress in the eight planes of the octahedron vanishes, provided

$$\sigma_1^* = \sigma_1 - \sigma, \ldots$$

To determine finally also the direction of the shearing stress τ_n acting in one of the eight octahedron planes it is not necessary to consider the reduced system σ_1^*, σ_2^*, σ_3^*, for we can construct τ_n for any system σ_1, σ_2, σ_3. If only the principal stress σ_2 would act alone (Fig. 16), then this stress σ_2 would produce a resultant stress σ_r in the plane *EFG* indicated in Fig. 16, making equal

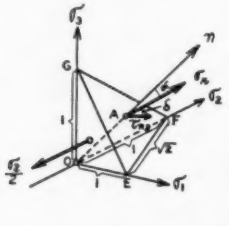


FIG. 16:- THE SHEARING STRESS τ_{n_2} PRODUCED BY σ_2

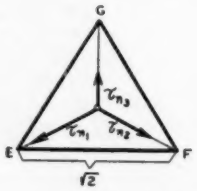


FIG. 17:- CONSTRUCTING OF $\tau_n = \frac{1}{3}\sqrt{\tau_1^2 + \tau_2^2 + \tau_3^2}$ FROM THREE COMPONENTS $\tau_{n_1}, \tau_{n_2}, \tau_{n_3}$

angles with the three directions σ_1, σ_2, σ_3, and which must be equal to $\sigma_2/\sqrt{3}$. (The area of the equilateral triangle *EFG* is $\sqrt{3}/2$ if the lengths of the three edges $OE = OF = OG$ of the tetrahedron *OEFG* are chosen equal to unity.[9]) Thus from equilibrium,

$$\frac{\sigma_2}{2} = \sigma_r \cdot \frac{\sqrt{3}}{2} \quad \text{or} \quad \sigma_r = \frac{\sigma_2}{\sqrt{3}} \ldots\ldots\ldots [16]$$

The stress σ_r must be parallel to σ_2, hence its tangential component or the shearing stress $\tau_{n2} = \sigma_r \times \cos\delta$, where δ is the angle of the σ_r axis with the plane *EFG*. As $\cos\delta = \sin\alpha$ and $\cos\alpha = 1/\sqrt{3}$, $\cos\delta$ becomes $= \sqrt{\frac{2}{3}}$. Thus

$$\tau_{n2} = \frac{\sqrt{2}}{3}\sigma_2 \ldots\ldots\ldots\ldots\ldots [17]$$

and the vector representing τ_{n2} must for reasons of symmetry coincide with the bisecting line of the angle at *F*. Similarly the other two principal stresses σ_1 and σ_3 will produce shearing stress components in the triangle *EFG* equal to

$$\tau_{n1} = \frac{\sqrt{2}}{3}\sigma_1, \quad \tau_{n3} = \frac{\sqrt{2}}{3}\sigma_3 \ldots\ldots\ldots\ldots [18]$$

The three coplanar vectors τ_{n1}, τ_{n2}, τ_{n3} are situated symmetrically as indicated in Fig. 17. Their resultant finally is the octahedral shearing stress τ_n given by Equation [15] and can thus be easily constructed.

As the elastic energy of distortion or the amount of energy required to distort the material purely elastically just to the yield point (the total elastic energy of deformation minus the work done in changing the volume) is equal to

$$A_d = \frac{1}{3G}\left(\tau_1^2 + \tau_2^2 + \tau_3^2\right) \ldots\ldots\ldots [19]$$

and for simple tension $\sigma_1 = \sigma_0$, $\sigma_2 = \sigma_3 = 0$, $\tau_1 = 0$, $\tau_2 = -\frac{\sigma_0}{2}$, $\tau_3 = \frac{\sigma_0}{2}$

$$A_d = \frac{\sigma_0^2}{6G} = \frac{2k^2}{3G} \ldots\ldots\ldots\ldots [20]$$

Equation [1] or [7] may therefore also be expressed by saying that yielding starts as soon as an amount of energy of distortion A_d given by [20] is reached in the material. This is the same amount of distortion energy the material contains in pure tension just when yielding starts.

Summing up the foregoing statements, it can be seen that the conditions of yielding, Equation [1] or [15], may be interpreted physically in several easily expressible ways. Mathematically they are simple and show a close connection with the fundamental variables of tensors, and it may perhaps be of interest to state briefly some additional reasons or facts which might be considered in their favor.

From a mechanical point of view it is hardly conceivable why the intermediate principal stress σ_2 (assuming that algebraically $\sigma_1 > \sigma_2 > \sigma_3$) should in polycrystalline materials have no effect when yielding starts. The conditions of yielding as stated in Equations [1] and [7] indicate that yielding is influenced by σ_2. According to the so-called "maximum-shear theory" and to its generalization by O. Mohr, yielding should not depend on σ_2. The experimental evidence based on more recently made tests, however, is opposed to this. (Cf. discussion of test results given later.) According to the observations which led to the condition $\mu = \nu$, the dimensions of a body which is plastically deformed change in a direction parallel to the direction of the intermediate principal stress σ_2. The only exception is when $\sigma_2 = (\sigma_1 + \sigma_3)/2$. In this case the radii of the two smaller circles in Mohr's representation become equal and the mean stress equal to the intermediate stress: $\sigma = (\sigma_1 + \sigma_2 + \sigma_3)/3 = \sigma_2$. This is the case of a pure shear or a pure shear combined with a pure hydraulic-pressure distribution. According to Mohr's hypothesis or to its special case included in the theory of maximum shear, one should expect, however, that in all other cases of stress or strain the intermediate dimension of a body parallel to ϵ_2 or σ_2 should not change, except only in the case when two of the three principal stresses become equal to each other. This has not been verified by tests.

The condition of yielding as an equation $F(\sigma_1, \sigma_2, \sigma_3) = 0$ containing the three principal stresses σ_1, σ_2, σ_3 may be represented or visualized graphically in space as a surface taking σ_1, σ_2, σ_3 as rectangular coordinates. This representation has been introduced by H. M. Westergaard and B. P. Haigh. The author suggested that this surface might be called the "surface

[9] Ros and Eichinger explain the shearing stress (or the corresponding unit shear γ_n) acting in an octahedral plane as "the resultant shearing stress" (or the "resultant shear") of the given stress or strain distribution. A stress or strain tensor having in general six components cannot be reduced or replaced by one "resultant shearing stress or strain." If the six components of a strain tensor are all finite and one relation is prescribed among them, as is the case above, they can be reduced to not less than five simple shears, but not to a single "resultant shear."

of yielding." The general problem of describing the conditions of yielding would then consist in determining empirically by tests the shapes of the surfaces of yielding $F(\sigma_1, \sigma_2, \sigma_3) = 0$ for the principal materials of construction.[10]

Taking $\sigma_1 > \sigma_2 > \sigma_3$, the maximum-shear theory is expressed by

$$\tau_2 = \frac{\sigma_1 - \sigma_3}{2} = \text{const} \quad \ldots\ldots\ldots\ldots\ldots [21]$$

which is the equation of a plane, and Mohr's theory by

$$\frac{\sigma_1 - \sigma_3}{2} = f\left(\frac{\sigma_1 + \sigma_3}{2}\right) \quad \ldots\ldots\ldots\ldots [22]$$

which is the equation of a cylinder. By interchanging the order of $\sigma_1, \sigma_2, \sigma_3$, six such planes and six such cylinders will be obtained. The shape of the surface of yielding for the various possibilities which may exist for $\sigma_1, \sigma_2, \sigma_3$ is therefore given in the former case by a hexagonal prism, the axis of which bisects the first octant of the principal directions. According to Mohr's theory also, a sort of hexagonal body is obtained having six different curved faces with sharp edges oriented symmetrically around a straight axis. This leads to insurmountable difficulties from a mathematical point of view, if the stresses in a plastic body change from point to point and are functions of the coordinates x, y, z. It would be necessary to introduce inequalities, which would limit the stresses to one or the other of the faces of the surface of yielding. Such difficulties are at once eliminated by introducing the von Mises-Hencky condition of yielding:

$$(\sigma_1 + \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 = 2\sigma_0^2 = \text{const.} \quad [23]$$

which is a circular cylinder. This cylinder circumscribes the hexagonal prism of the maximum-shear theory, and it is evidently a simpler condition to satisfy that a point may be situated on a circular cylinder than to prescribe that the point may move on six different planes. It is true that in certain cases it may be advantageous to replace parts of this cylindrical surface by a plane, by which the second-order expression (Equation [23]) in the stresses is reduced to a linear equation. In general, however, to introduce a hexagonal cylinder or a body having regular hexagons with changing side lengths as sections, as in Mohr's case, seems too artificial. Certain other objections against Mohr's theory have been recently raised by W. V. Burzynski[11] which make it probable that Mohr's enveloping curve does not envelop truly all the possible limiting largest stress circles which determine the critical states of stress.

IV—TESTS ON THE YIELDING OF DUCTILE METALS UNDER COMBINED STRESS

(a) W. Lode's Recent Tests

Since the tests described in the papers mentioned in footnote No. 3 were made, W. Lode and the author have attempted to control the former experiments with improved apparatus.[12] This was done with two different equipments.

1 *Axial Tension and Internal Pressure.* A 20-ton universal Losenhausen machine for tension and a high-pressure pump designed by A. Amsler, of Schaffhausen, which could produce hydraulic pressures up to 1200 atm where simultaneously used. Hollow cylinders were tested, which were stretched in the axial direction and subjected to internal pressure. Particular care was taken in cutting these cylinders from various steels in the form of solid bars, these being first drilled out for the bore, to avoid the inevitable anisotropy encountered in drawn tubes (such as were used by J. Guest and others and in earlier tests by Dr. Lode). The steel tubes had a gage length of 180 to 200 mm, an internal diameter of 20 to 30 mm, and wall thicknesses of 3 to 3.8 mm. The wall thickness did not vary more than 0.1 mm, this being controlled by a special micrometer apparatus. The bars had heavy heads with gradually changing cross-sections and large fillets.

The tests were conducted so that the high-pressure pump furnished an internal pressure p, which increased in a given proportion simultaneously with the increasing axial load P. To secure this simultaneous proportional increase of the axial and tangential principal stresses in the hollow cylinder, an auxiliary pointer was designed which indicated the pressure p but also moved along the scale on which the axial loads were read. By adjusting the valve of the pump for the high pressure the auxiliary pointer could be moved with such a speed that it would advance simultaneously with the pointer of the load, thus securing strictly proportional increase of both p and P. The pressures of the Amsler pump were measured by a high-pressure pendulum manometer, and the indications for the axial loads of the tensile machine were controlled by a balance beam. The changes of shape of the tubes were carefully measured after each test by micrometers, and during the test by means of a Bauschinger roller apparatus and a lateral extensometer.

From the theory of elasticity the stresses in a tube closed at its ends (r_1, r_2, internal and outer radii) and simultaneously subjected to internal pressure p and an axial load P, are given by

Radial stress: $$\sigma_r = -\frac{pr_1^2}{r_2^2 - r_1^2}\left(\frac{r_2^2}{r^2} - 1\right) \quad \ldots\ldots [I]$$

Tangential stress: $$\sigma_t = \frac{pr_1^2}{r_2^2 - r_1^2}\left(\frac{r_2^2}{r^2} + 1\right) \quad \ldots\ldots [II]$$

Axial stress: $$\sigma_z = \frac{pr_1^2}{r_2^2 - r_1^2} + \frac{P}{\pi(r_2^2 - r_1^2)} \quad \ldots\ldots [III]$$

Taking for r the mean radius

$$r = \frac{r_1 + r_2}{2} \quad \ldots\ldots [IV]$$

and designating the wall thickness by

$$\delta = r_2 - r_1 \quad \ldots\ldots [V]$$

the formulas above may be rewritten, after expanding in powers of δ and neglecting higher powers of δ than the first, as follows:

$$\left.\begin{aligned} \sigma_r &= -\frac{p}{2} \\ \sigma_t &= -\frac{p}{2} + \frac{pr}{\delta} \\ \sigma_z &= -\frac{p}{2} + \frac{pr}{2\delta} + \frac{P}{2\pi r\delta} \end{aligned}\right\} \quad \ldots [VI]$$

The condition of yielding at the mean radius of a hollow tube then takes according to Equation [23] the form

[10] A very instructive discussion of the shapes of this surface for the various theories of strength is given in the paper by H. M. Westergaard, "On the Resistance of Ductile Materials to Combined Stresses," *Jl. Franklin Inst.*, May, 1920, p. 627.

[11] "Über die Anstrengungshypothesen," *Schweiz. Bauzeitung*, Zürich, Bd. 94 (1929), Nov. 23.

[12] After the author left Germany at the end of 1927, Dr. Lode continued his tests until 1929. As the latter has until now also not published the results of his recent tests, a brief résumé of a part of his tests is given above.

$$\left[\frac{P}{2\pi r\delta\sigma_0}\right]^2 + \left[\frac{\sqrt{3}rp}{2\delta\sigma_0}\right]^2 = 1 \ldots\ldots\ldots\ldots [VII]$$

With regard to the terms included in the two brackets and which are taken as variables, this is the equation of a circle having unity for its radius. σ_0 is the yield stress for pure tension. The results

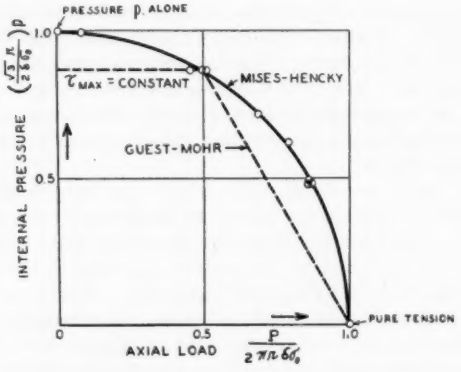


FIG. 18-LODE'S TESTS OF 1926. (INTERNAL PRESSURE & AXIAL LOAD)

of Lode's recent tests are given in Fig. 18, in which the quantity in the first bracket of [VII] is the abscissa, and the quantity in the second bracket the ordinate. The points referring to the tests

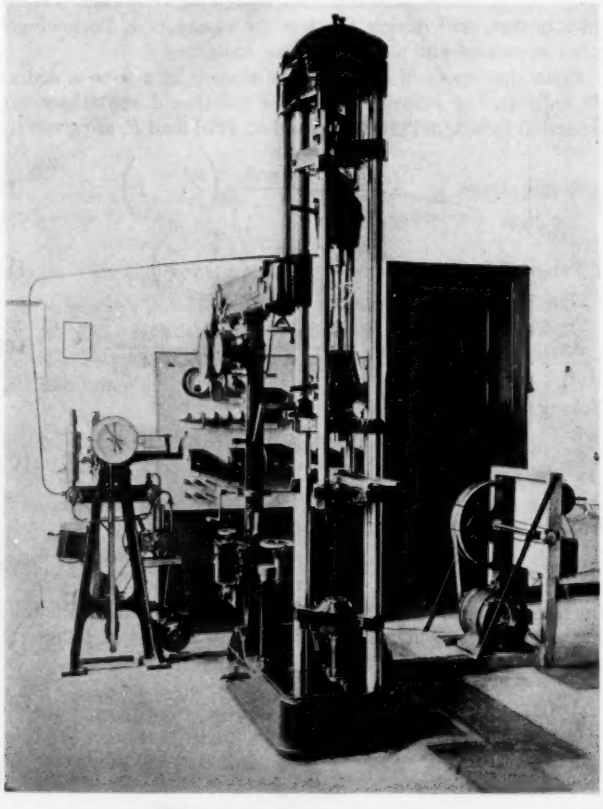


FIG. 19 COMBINED STRESS TESTING

fall close to the circle corresponding to the condition of yielding (Equation [VII]) or the condition of constant energy of distortion. The two straight lines in Fig. 18 correspond to the maximum-shear theory. Fig. 19 shows the testing equipment.

At the left is the Amsler high-pressure pump mounted on a pendulum manometer. In the center is the 20-ton Losenhausen testing machine shown with the test tube in position. The motor drive of the machine will be seen at the right of the figure. Fig. 20 is a closer view of the high-pressure apparatus.

2 *Tension-Torsion Machine.* This machine was built by

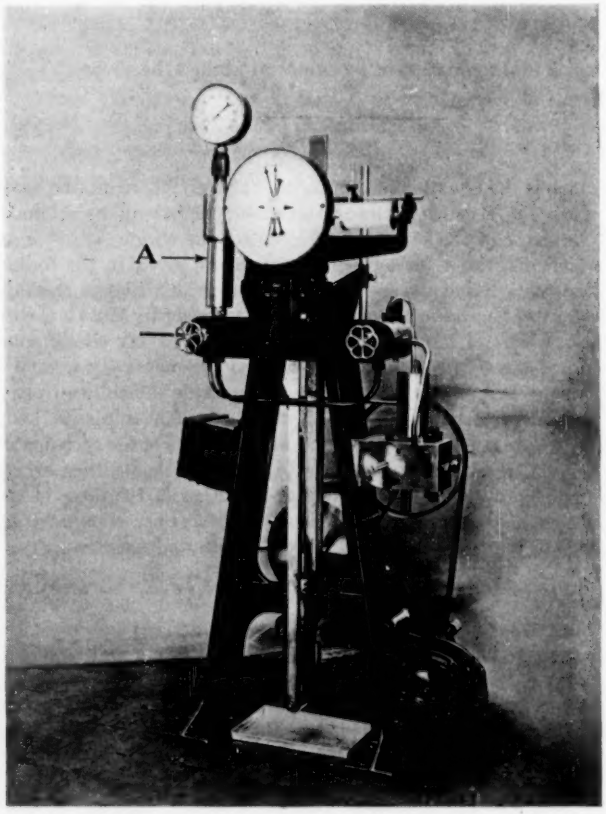


FIG. 20 AMSLER HIGH-PRESSURE PUMP

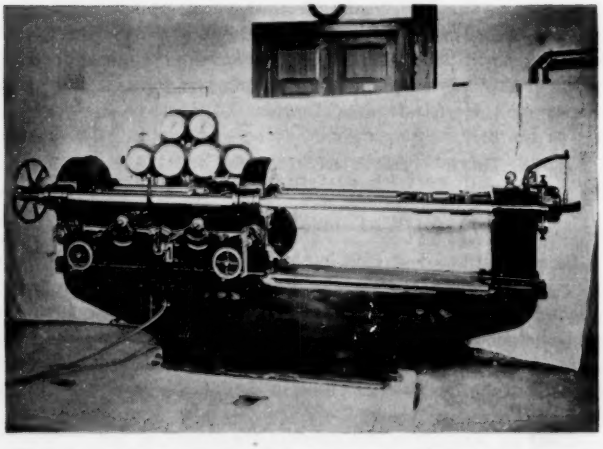


FIG. 21 LOSENHAUSEN TENSION-TORSION TESTING MACHINE

the Losenhausenwerk in Düsseldorf.[13] In it hollow specimens can be simultaneously distorted by applying axially a tensile load and by twisting the tube by a torque. Fig. 21 shows a view of the machine. It consists of two screw-type drives com-

[13] Designed by Baurat v. Bohuczevitz and Eng. Sontag.

bined into one machine. An electric motor through two friction drives operates the tension and torsion parts of the machine, which can be coupled together or used separately. It is possible to adjust the speed for axial stretching independently of that for twisting or also in a given ratio of the two. The axial load and the torque are measured by means of calibrated hydraulic-pressure gages (*messdose*) of the Losenhausen type. The grips and the attachments of the tube specimen to the machine are

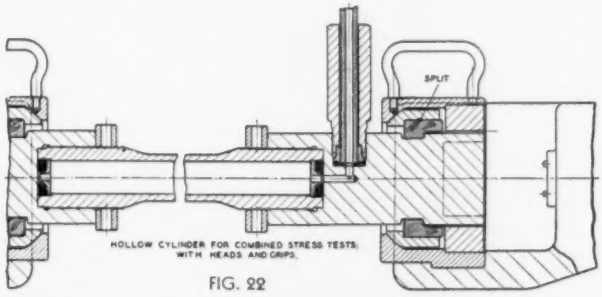


FIG. 22

shown in Fig. 22. The attachments are so designed that internal pressure can also be applied to the tubes or combined with tension or torsion or both. The capacity of the machine is 15 tons for tension and 150 kg-m for torsion. The machine has been used in various tests since the author left Germany, among others for one of the purposes for which he had it designed, namely, for experiments in which the axis of the applied-stress tensor is rotated relatively to the stressed body.

In this case of yielding there is no justification in assuming that the tensor of the elastic strains will at all times remain coaxial with the tensor of the permanent strains.

By combining the results of his tests described under (*a*) with a few torsion and combined torsion-tension tests, Dr. Lode attempted to investigate whether an appreciable effect of the mean tensile stress $\sigma = (\sigma_1 + \sigma_2 + \sigma_3)/3$ might be ob-

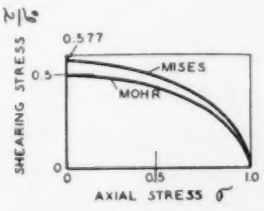


FIG. 23. - σ AND τ FOR TENSION-TORSION

served on the constant k in Equation [15] or on the radius of a section of the surface of yielding normal to its axis. The results were negative for steel, thus indicating that no appreciable effect of the mean pressure on the constant appearing on the right side of the condition of plasticity, Equation [1] or [15], might be present.

(*b*) Tests by G. I. Taylor and H. Quinney

In a recent paper[14] the first-named author has extended his most careful observations on the plastic deformations of single metallic crystals[15] to polycrystalline ductile metals. For obtaining the critical values of the principal stresses at the plastic limit, Taylor and Quinney subjected tubes to a direct axial load and to torsion. By measuring the change of the internal volume of the tube during the strain by filling it with water and reading the movement of a column in a capillary tube, they were able to determine also the relation existing between the quantities μ and ν in a more perfect way than was the case in Lode's early tests. The volume of the bore of a tube stretched plastically in axial tension should not change. The same is true for pure torsion, and hence also for all combinations of both, provided that $\mu = \nu$. Hence the possible changes of volume of the water

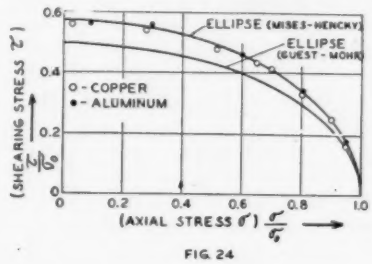


FIG. 24

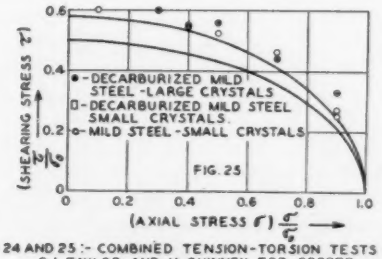


FIG. 24 AND 25 :- COMBINED TENSION-TORSION TESTS BY G.I. TAYLOR AND H. QUINNEY FOR COPPER, ALUMINUM AND MILD STEEL.
σ_0 = YIELD STRESS FOR PURE TENSION

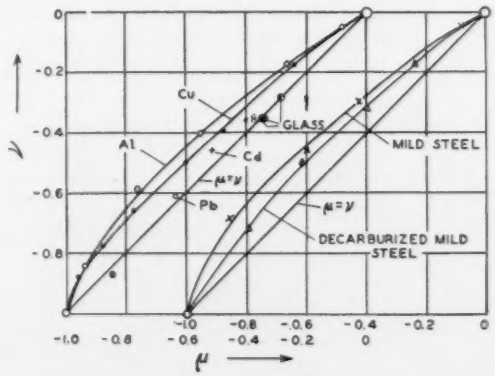


FIG. 26 - $\nu = f(\mu)$ ACCORDING TO TESTS BY G.I. TAYLOR AND H. QUINNEY.

inside the bore could be utilized to find any observable deviations from the law $\mu = \nu$. These tests led to the results reproduced in Figs. 24, 25, and 26. According to Equation [1], yielding in a tube subjected to tension-torsion should start when

$$\sigma^2 + 3\tau^2 = \sigma_0^2 = \text{const.} \quad \text{[VIII]}$$

where σ is the axial tensile and τ the shearing stress, and σ_0 the yield stress for pure tension. However, according to the theory of maximum shear (Guest-Mohr), yielding would start as soon as

$$\sigma^2 + 4\tau^2 = \sigma_0^2 = \text{const.} \quad \text{[IX]}$$

Plotting σ and τ as rectangular coordinates, the expressions above give ellipses with the semiaxes σ_0, $\sigma_0/\sqrt{3}$ and σ_0, $\sigma_0/2$, respec-

[14] "The Plastic Distortion of Metals," Phil. Trans. Roy. Soc. London, Series A, vol. 230, pp. 323–362.

[15] "The Distortion of an Aluminum Crystal During a Tensile Test," Bakerian Lecture, Proc. Roy. Soc. London, vol. 102 (1923), p. 643, and subsequent papers.

tively (Fig. 23). Figures 24, 25, and 26 taken from Taylor and Quinney's report show that for aluminum or copper the test points very closely follow the upper ellipse corresponding to Equation [VIII], disproving Equation [IX]. For mild steel, however, the agreement between the observations and the curves was by no means as good. All points were found lying above the von Mises ellipse. In any case they were closer to this than to the other ellipse. As the observations of the volume filled with water showed anomalies which appeared closely connected with the breaks of the stress-strain curve, most probably the different behavior of mild steel in these tests must be attributed to the non-uniformity of the first yielding over the length of the specimen (yielding in thin layers) due to the phenomenon of upper and lower yield points. The previous and varying thermal heat treatment of the steel specimen may have also been instrumental in causing some of these discrepancies.

Finally, Fig. 26 shows the test results of Taylor and Quinney with regard to the kinematics of the plastic flow. The scattering of the points in Lode's μ, ν diagram appears to be eliminated here, and for the relation $\nu = f(\mu)$ very distinct curves were obtained, although not very different from the straight line $\mu = \nu$.

One very interesting result was that the curves for the hexagonal metal cadmium and for lead approached the straight line $\mu = \nu$ much closer than did those for metals (copper, aluminum) with a regular lattice. The points for the amorphous heated glass were almost exactly on the line $\nu = \mu$, as was to be expected for viscous flow of a substance. It seems probable that the type of lattice structure within the individual grains has something to do with the shape of the $\nu = f(\mu)$-curve. This was suspected by Lode, who tried to explain the shape of the actual curve $\nu = f(\mu)$ by the statistical effect of a great number of grains oriented at random with their axes, but yielding according to the well-known modes of the single metallic crystals.

Summing up the conclusions from the preceding tests by Taylor, Quinney, and Lode, it appears (1) that the condition of yielding for ductile metals is, within experimental errors, well expressed by the condition

$$\tau_1^2 + \tau_2^2 + \tau_3^2 = \frac{\sigma_0^2}{2} = \text{const} \ldots\ldots\ldots [X]$$

and (2) that when suppressing the clearly observable deviation of the curves $\nu = f(\mu)$ from the straight line $\nu = \mu$ in the first approximation and for the purpose of a simple mechanics of the plastic state, the relation

$$\nu = \mu \ldots\ldots\ldots\ldots\ldots\ldots [XI]$$

is sufficiently accurate to express the third law of yielding for the ductile metals.

(c) Tests by M. Ros and A. Eichinger[16]

Among a number of other tests, Ros and Eichinger compared the behavior of various carbon steels and of cast steels in hollow or solid specimens subjected to tension or compression combined with internal pressure, and also to torsion. For each material they plot three kinds of stress-strain curves, namely, (1) the maximum stress σ_1 versus the maximum strain ϵ_1, (2) the maximum shearing stress τ_{max} as a function of the maximum shear γ_{max}, and (3) the shearing stress τ_n acting in the octahedral planes of the stress or strain tensor as a function of the corresponding strain γ_n (see Equations [12] and [14]). The various stress-strain curves which they obtained for different stress distributions differed from each other considerably when plotted according to the schemes (1) and (2), but these curves came closer together in the third diagram. Their conclusion, therefore, was that Equation [X] is the best representation of the condition of yielding.

By plotting the continuous stress-strain curves over a longer interval of the strains observed on various bars the comparison of the curves becomes less exact, because on account of the small differences in the speed of testing or in the structure of the metals which cannot be avoided, the deviations will accumulate gradually until finally the curves diverge. These and other tests by Ros Eichinger not discussed here indicate the superiority of the condition of yielding (Equation [X]) over the Guest-Mohr condition. If they had used the testing method described under (1) and (2) above (Lode, Taylor, and Quinney) they would have arrived probably at a still closer agreement between theory and tests.

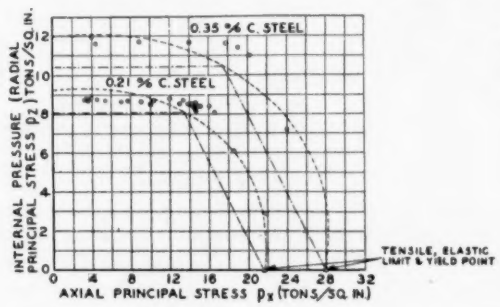


FIG. 27.—TESTS WITH THICK-WALLED TUBES BY G. COOK

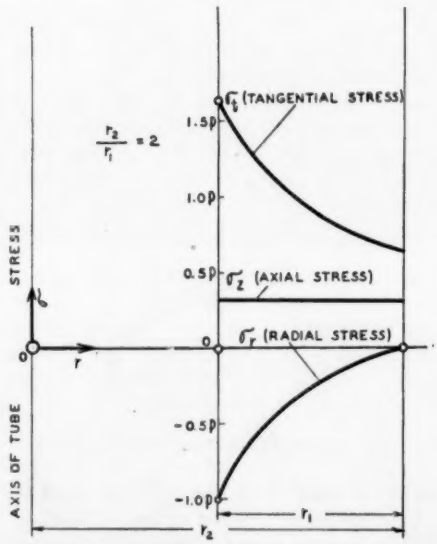


FIG. 28.—ELASTIC STRESS DISTRIBUTION IN THICK-WALLED TUBE PRODUCED BY INTERNAL PRESSURE

In all these tests, yielding of the ductile metals was observed parallel to the direction of the intermediate stress σ_2 [except in the case when $\sigma_2 = (\sigma_1 + \sigma_3)/2$], thus disproving Mohr's conclusions about the mode of yielding.

Corroborating Lode's conclusions, they found no appreciable influence of the mean stress $\sigma = (\sigma_1 + \sigma_2 + \sigma_3)/3$ on the maximum shearing stress τ_{max} during yielding within the comparatively wider range of stress distributions tested here compared to Lode's tests.

[16] "Versuche zur Klärung der Frage der Bruchgefahr. III. Metalle." Diskussions-bericht No. 34. Eidgenossische Materialprüfungsanstalt, Zürich, 1929.

(*d*) Tests by G. Cook

While the recent tests described under (*a*), (*b*), (*c*) were all made with regard to yield stresses, G. Cook has in a recent paper[17] attempted to investigate the elastic limit under combined stress in order to discover whether a similar effect is observable in regard to the elastic limit in previously unstressed materials. Thick-walled tubes of two steels (with 0.35 and 0.21 per cent carbon, respectively) were tested by simultaneously applying to them internal pressure and an axial load. The ratio of the outer to the inner diameter of the tubes, however, was chosen extremely large, namely, $r_2 : r_1 = 2$. This leads to the elastic stress distribution shown in Fig. 28 in case the tube is subjected to internal pressure p alone and is closed. To this distribution is added axial tension σ by which σ_z is changed into $\sigma_z + \sigma$. The tests were conducted by applying an axial load first and then increasing the pressure until the first slight signs of permanent set became observable. For this purpose, also, the radial and axial extensions were measured. The observed points were then plotted in a diagram similar to that indicated by Equation [VII] or Fig. 18, which is reproduced here in Fig. 27. In this last figure the axial principal stresses $\sigma_z + \sigma$ or the quantities

$$\sigma + \frac{pr_1^2}{r_2^2 - r_1^2}$$

are plotted as abscissas, and the radial principal stresses at the highest-stressed inner surface $r = r_1$ of the hollow tube, that is, $\sigma_r = -p$ or the pressures p, are used as ordinates.

To the condition of yielding, Equation [X], there corresponds here also an ellipse which is indicated in Fig. 27, and to Guest-Mohr's condition, the two straight lines. The points reproduced from Cook's tests (Fig. 7, p. 572) do not seem to fit either of these two curves. The whole comparison is unfortunately obscured by introducing an "elastic limit" in the case of a rapidly changing stress distribution in the radial direction, such as that present in thick-walled tubes of such large ratio of outer to inner radius ($= 2$).

In a case of non-homogeneous stress distribution like the one tested by Cook in which the tangential stress σ_t at the instant of first yielding at the inner surface $r = r_1$ of the tube is more than 2.5 times larger than at the outer surface $r = r_2$, the danger is great that the critical instant of first yielding may have escaped observation.

It must be difficult to observe in a stress-strain curve the first instant when the yield stress is just reached in the infinitesimal layer coinciding with the *inner* surface of a thick-walled tube, when all readings of the extensions have been made at the *outer* surface. The possible presence of an upper and lower yield stress in tension (the two diagrams shown in Fig. 5 of the paper by Cook seem to indicate that a drop of stress was observed in the tensile tests) might have further obscured the comparison. In some of the published stress-strain curves in which the pressures p are plotted above the radial extensions of the tubes, more or less sharp breaks also appear which seem to support the view of the author that at these states of stress probably larger portions of the tubes were already yielding. Assuming that the yield stress for pure tension was possibly overestimated (by taking an upper yield stress), this would reduce the ellipse in Fig. 27 to a smaller-sized one and would bring the observation points of the pressures p *above* the ellipse, as should be expected if higher pressures were observed than those producing first yielding.

The cylinders in these tests were first axially loaded with a stress such that, with the addition of the stress due to internal pressure, yielding would start. When internal pressure was then added, the ratio of the principal stresses $\sigma_1 : \sigma_2 : \sigma_3$ during the raising of the pressure p until yielding started was not kept constant. It is possible that yielding under such circumstances starts at different values of σ_1, σ_2, σ_3 from those when the three principal stresses $\sigma_1 : \sigma_2 : \sigma_3$ are increasing proportionally until they reach the plastic limit. This is a point that needs further testing.

Summing up the preceding remarks, it seems probable that some systematic influences not taken into account might have been the reasons for the deviations observed in these last tests.

(*e*) Concerning the "Elastic Limit" Under Combined Stress

A word in regard to this may perhaps be in order. More recent tests, particularly tests in which both the principal stresses increase according to some law and at the same time the principal axes of the stress tensor gradually change their position in space with regard to the material, show that the directions of the three axes of the tensor of the elastic parts of the total strains at a given instant do not coincide with the directions of the three axes of the stress tensor. This has the consequence that, particularly after a yield test has been made under a given ratio of $\sigma_1 : \sigma_2 : \sigma_3$ and then another yield test is made on the same test piece in which $\sigma' : \sigma_2' : \sigma_3' \neq \sigma_1 : \sigma_2 : \sigma_3$, a transition curve will be observed starting from the elastic straight line, combining any two corresponding stresses and strains σ_n and ϵ_n and depending on the previous operations until the plastic limit is again reached. In other words, the stresses σ_{1e}, σ_{2e}, σ_{3e} at the point where a transition curve starts, or the stresses referring to the "elastic limit," depend not only on the amount of strain hardening, but particularly also on the *change* of the ratios of the critical values of $\sigma_1 : \sigma_2 : \sigma_3$ from one test to another. It has already been stated that the reversing of the stress directions (for example, change from tension into compression or reversing of the directions of a pure shear) has a large effect on the shape of the transition curve and brings the elastic limit practically down to zero. For these and other reasons not mentioned here, it must be conceded that the "elastic limit" for a given material will in general depend also on the previous history of the straining of the material, and therefore lose much of its significance as a limiting state of stress.

V—THE MORE GENERAL KIND OF YIELDING OF POLYCRYSTALLINE MATERIALS

For single metallic crystals as well as for the polycrystalline ductile metals, it has been repeatedly shown that the addition of a uniform tensile or compressive stress will not change the magnitude of the principal shearing stresses τ_1, τ_2, τ_3 at a plastic limit.

For other less compact materials, in which the grains are not well cemented together, this may be different. For such materials the diameters of the three principal-stress circles for which μ has the same value but the mean stress

$$\sigma = \frac{\sigma_1 + \sigma_2 + \sigma_3}{3} \quad \text{[XII]}$$

varies, will depend on σ. For any given type of stress $\sigma_1 > \sigma_2 > \sigma_3$ corresponding to a given μ, the radius of the largest stress circles

$$\tau_2 = \frac{\sigma_1 - \sigma_3}{2}$$

will continuously change, either with $\frac{\sigma_1 + \sigma_3}{2}$ or with

$$\sigma = \frac{\sigma_1 + \sigma_2 + \sigma_3}{3}$$

[17] "The Elastic Limit of Metals Exposed to Triaxial Stress," Proc. Roy. Soc. London, Ser. A., vol. 137 (1932), p. 559.

Introducing at the plastic limit

$$\tau_2 = \frac{\sigma_1 - \sigma_3}{2} = f\left(\frac{\sigma_1 + \sigma_3}{2}\right) \ldots\ldots\ldots [XIII]$$

would be what Mohr assumed. As it is scarcely probable that the intermediate principal stress σ_2 would remain without any influence on the value of τ_2, other mathematical relations have recently been suggested by means of which the experimental facts just mentioned might find more adequate expression.

While F. Schleicher[18] for this or similar reasons has suggested that the total elastic-strain energy might be assumed as a function of the mean pressure, $\sigma = (\sigma_1 + \sigma_2 + \sigma_3)/3$, that is,

$$\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \frac{2}{m}[\sigma_2\sigma_3 + \sigma_3\sigma_1 + \sigma_1\sigma_2] = f(\sigma) \ldots [XIV]$$

where m is Poisson's ratio, W. Lode[19] and W. T. Burzynski[20] suggest the expression of the condition of this more general kind of yielding by assuming that the energy of distortion is a function of σ. Referring to what has been said above with regard to the significance of the octahedral shearing stress τ_n (Equation [14]) it is here suggested that this last condition might now be expressed by assuming that "the octahedral shearing stress τ_n (acting in the octahedron planes of the stress tensor), or

$$\tau_n = \frac{2}{3}\sqrt{\tau_1^2 + \tau_2^2 + \tau_3^2} = f(\sigma) \ldots\ldots\ldots [XV]$$

shall now be considered as depending on the mean stress $\sigma = \frac{\sigma_1 + \sigma_2 + \sigma_3}{3}$."

Although both conditions, Equations [XIV] and [XV], at a closer examination do not seem to differ much mathematically and can be transformed into each other under certain conditions, the second expression must receive the preference for various reasons. One is that Equation [XIV] contains Poisson's ratio m, which does not appear explicitly or implicitly in Equation [XV]. As Burzynski points out, it is possible when adjusting the constants in Equation [XIV] to fit the tests, to be led to absurd conclusions with regard to the value of Poisson's ratio m.

Reasons for the superiority of the relation Equation [XV] over the old Mohr relation Equation [XIII] become more obvious when discussing the shapes of the surface of yielding corresponding to these relations. They are the same as those advanced in favor of the Mises condition of Equation [X]. If the order of σ_1, σ_2, σ_3 is interchanged, the Mohr relation, Equation [XIII], leads to three sets of cylindrical surfaces intersecting each other along sharp edges, while Equation [X] is that of a single continuous surface of rotation having the diagonal of the first octant of the σ_1, σ_2, σ_3 planes as its axis. The improbable exceptional behavior of the intermediate stress has also been eliminated by assuming the validity of Equation [XV] instead of that of Equation [XIII].

The tests made by T. v. Kármán, Böker, Ros and Eichinger, and Brandtzaeg on brittle materials, such as marble, sandstone, or concrete, tend to show that the yielding of such materials, which under the ordinary tensile test would behave as brittle materials, will, under increasing mean pressures

$$\sigma = -\left|\frac{p_1 + p_2 + p_3}{3}\right|$$

approach the laws of the yielding of the ductile metals. Experimental evidence shows that the surface of yielding must approach asymptotically a cylindrical surface at high hydraulic pressures. This is a consequence of the intercrystalline mechanism of gliding in the individual crystal grains of any polycrystalline material, even though under ordinary conditions (low mean pressures σ) it may be as weak as some of the less strong materials mentioned above. By compressing the aggregate of loose crystals under a high mean pressure, the former starts to yield provided that the pressures in certain directions attain the required magnitude, and in a way similar to that in which a ductile metal would act under low mean pressures.

[18] *Zeit. für angew. Math. u. Mech.*, 1925, p. 478 and 1926, p. 199.

[19] Loc. cit., Forschungsheft No. 303, p. 2.

[20] *Schweiz. Bauzeitung*, loc. cit. (Cf. also several of his papers published in the Polish language in Lwow.)

VI—RUPTURE

We come finally to a brief discussion of some of the mechanical conditions which may express or be used to determine the danger of a material separating into fragments. It must be said, however, that it cannot be anticipated here with the same certainty or experimental evidence that in the conditions of rupture the principal stresses σ_1, σ_2, σ_3 will figure as the only or the principal variables as expressed in the relations of yielding. In amorphous solid and strong materials not having a pronounced crystalline grain structure or with a submicroscopic crystalline structure, however, this may, to a certain extent, be the case. In such materials one kind of rupture is known, the circumstances of which have perhaps been the best described, although they may still be obscure enough.

(*a*) Brittle Fracture Due to Tensile Stress

In its original form, the theory of strength advanced by O. Mohr left it quite undecided whether the critical conditions stipulated in this theory might be applied to failure due to yielding (plastic flow) or to rupture. This point has been brought out particularly by von Kármán's tests with brittle crystalline materials, such as marble or sandstone, subjected to combined stress. In general, tests seem to indicate that if at least one of the three principal stresses is a tensile stress, failure due to tensile fracture in such materials occurs when the largest of the tensile principal stresses reaches a value equal to the tensile strength in pure tension. The surface of rupture is perpendicular to the maximum tensile stress. Thus in this region neither Mohr's enveloping curve nor the surface of yielding can have any significance. For example, using the representation of the surface of yielding in its general form given by Equation [XV], this would mean that this surface cannot be extended beyond the region where one of the three principal stresses is a tensile stress and has at least reached the tensile strength of rupture. Thus for the rupture due to tensile stresses, Rankine's theory of maximum principal stress may be taken as the hypothesis representing the experimental facts. The states of stress producing the tensile failure when represented by points in the system of rectangular coordinates σ_1, σ_2, σ_3 engender also a certain surface $H(\sigma_1, \sigma_2, \sigma_3) = 0$ which might be called "the surface of rupture." According to the present hypothesis this surface consists of the three mutually perpendicular planes:

$$\sigma_1 = c, \quad \sigma_2 = c, \quad \sigma_3 = c \ldots\ldots\ldots\ldots [XVI]$$

where c is an essentially positive stress (equal to the tensile strength in pure tension). The "surface of yielding" has thus to be limited toward its tensile region by the "surface of rupture" consisting here of the three mutually perpendicular planes as just described.

(*b*) Sliding Fracture

Another kind of fracture is known in brittle crystalline materials, and can also be observed when ductile materials break

after considerable permanent strain. The surface of rupture is a plane inclined at a certain angle to one of the principal-stress planes. This kind of fracture occurs, for example, in compression tests made with cylinders of brittle materials (cast iron), in which cases nearly plane oblique surfaces of rupture may sometimes be observed.

A closer examination of such fractures may, however, disclose that the phenomenon is not altogether so simple as it may appear at the first glance. Long before a surface of rupture finally forms, it will be seen that the structure of the material has undergone changes. The crystalline structure is gradually loosened, minute cracks appear in the grains or the latter exhibit discoloration in thin samples. This is particularly the case in compression tests when in the perpendicular direction to the principal compressive stress, no external stresses act. A more or less solid crystalline material, the grains of which have been first cemented together by their boundary substances, when subjected to a gradually increasing uniaxial compressive stress, will show a disintegration of the grain structure or a gradual weakening of its strength and its internal structure. Certain grains which will be oriented favorably with regard to the stress field, will start to yield and deform by slip in their crystallographically determined gliding planes. Other grains which are in an unfavorable direction for slip may be ruptured. If soft inclusions are present (that is, points of weakness, as, for example, the graphite inclusions of a cast iron), the result will be that the whole structure of the material will be changed under the increasing stress. Another important fact may be mentioned, to which Griffith seems to have first called attention, namely, that in pure uniaxial compression, failure may still be due to local tensile stresses.

For example, the graphite inclusions in cast iron are equivalent to minute cracks, and if such cracks are oriented at random in a field of compressive stress, tensile stresses will be developed along certain of the cracks. The final surface of rupture will be inclined to the principal compressive stress. For this kind of fracture perhaps Mohr's theory may be sufficient as a working one upon which to predict the fracture and its orientation. Assuming that the radii of Mohr's largest principal-stress circles would be known for pure shear τ_s and for pure compression τ_c (see Fig. 29) so that τ_s would be the strength for pure shear and

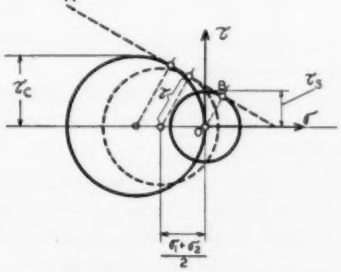


FIG. 29- CONDITION OF SLIDING RUPTURE

$2\tau_c$ the strength for pure compression, according to Mohr and Duguet a tangent AB to these two circles may be drawn. For intermediate cases of stress, the maximum shearing stress τ_{max} or the radius of the largest principal-stress circle

$$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} \quad \text{[XVII]}$$

at the instant of a sliding fracture would have to be chosen so that this circle becomes tangent to AB (circle shown by broken lines in Fig. 29). As condition of rupture, the relation

$$\tau = \frac{\sigma_1 - \sigma_3}{2} = \tau_s + \left(1 - \frac{\tau_s}{\tau_c}\right) \cdot \frac{(\sigma_1 + \sigma_3)}{2} \quad \text{[XVIII]}$$

may be used, provided that $\sigma_1 > \sigma_2 > \sigma_3$, which can be read from Fig. 29.

(c) Fracture of Ductile Materials

In general, in ductile materials the state of stress at the instant of final rupture is almost without exception of a more or less complex nature, even if the material has first been subjected to a simple and homogeneous state of stress. This is particularly true for the rupture in an ordinary tensile or compression test. Consequently the fracture in these tests is also not of a simple nature, and more exact rules cannot be given beyond the empirical proceeding of taking a certain load in the stress-strain diagram and dividing it by the characteristical area of the original or the actual cross-section.

Discussion

Pierre F. de Bethune.[21] In section IV(d) of the paper, Dr. Nadai discusses the discrepancy between the von Mises-Hencky law and the tests of G. Cook.

The writer agrees entirely with the author's opinion on the previous history of the material. It is certain that a prior plastic deformation induces in the material an anisotropy which cannot be overlooked in a further deformation, especially if that deformation has not the same anisotropy. This can be physically expressed in terms of slip planes in every crystalline grain, and the writer thinks that it would not be difficult to find examples of slipping along a previously established set of planes in a specimen subjected to stresses which would induce movement along another set of planes if the material had not been previously deformed.

However, this explanation implies a previous plastic deformation, and Dr. Nadai refers to that when he speaks of transition curves. Now the writer would remark that in the case of a first yielding the former history of the material would have no effect, or at least a trifling one.

Another point he would make is that, next to the principal stresses σ_1, σ_2, σ_3, account should perhaps be taken of the partial derivatives with regard to space: $\partial\sigma/\partial x$, $\partial\sigma/\partial y$, $\partial\sigma/\partial z$. These are nil in the case of the work by Lode, Quinney, and Ros and Eichinger, where the stress distribution is nearly homogeneous, but have some value in the case of the experiments by Cook.

The writer bases this opinion on the fact that the plastic yielding of the grains occurs to the same scale as that of the individual grains, and that at that scale the metal is no longer homogeneous. The response of that heterogeneity to the stress must be different if at that scale the stress is also heterogeneous, i.e., if the terms of the form $\partial\sigma/\partial x$ have a sufficiently high value.

H. Hencky.[22] The connection between the so-called "testing of materials" and the methods of applied mechanics is not a very satisfactory one at the present time. One principal reason is that the problem of strength is not only a problem of mechanics but also one of colloidal chemistry. Therefore theoretical statements seeking to explain matters along purely mechanical lines are often overthrown by new experimental results. However, the author's critical exposition of the problem contains many suggestions for further research and helps to counteract any discouraging influence resulting from theoretical shortcomings.

In connection with further research, and referring to the em-

[21] Louvain, Belgium.

[22] Massachusetts Institute of Technology, Cambridge, Mass.

pirical formula for creep recommended by the author in some of his papers, namely,

$$\sigma = \sigma_1 + \sigma_2 \log_e (v/v_0)$$

which connects the strain velocity v with the stress, the writer would suggest changing it to

$$\sigma = \sigma_1 + \sigma_2 \log_e [1 + (v/v_0)]$$

because in this form it would be applicable to all strain velocities and could be easily generalized for triaxial stress. This last formula has important consequences as regards the theory of the rolling mill, which cannot be elaborated here, and the writer would ask whether the author has any objections to making the change.

Another point that the writer would discuss concerns the general theory of yielding of polycrystalline material. As is well known, the state of stress in a crystalline aggregate is turbulent. This is explained very clearly by Moore and Kommers in their treatise on "Fatigue" and was known to Heyn a long time ago, as the author has shown in some of his papers.

Now the stresses entering the plasticity condition are obviously only the average stresses. There must be, however, a plasticity condition limiting the amount of possible stress fluctuations. The hidden elastic energy connected with the stress fluctuations at the grain boundaries seems much more important for the plastic behavior than any possible influence of the hydrostatic stress, and the author is requested to state whether methods of measuring the amount of stored-up elastic energy have already been developed. Obviously this energy behaves in a manner quite analogous to that of heat energy.

P. G. McVETTY.[23] In this paper the author has brought into agreement theoretical analysis and actual test data relating to the behavior of metals under the action of combined stresses. It is of interest to note that this very complete discussion is restricted to temperatures low enough to avoid creep effects. In a previous paper[24] the author has discussed the problem of creep under the action of simple stresses at elevated temperatures. Up to the present time, however, very little has been done on the problem in which creep occurs under the action of combined stresses. The introduction of the concept that the stress-strain relations depend also upon the temperature and the rate of application of the stress adds complexity to the analysis of the problem and to the experimental work necessary to effect its solution.

There may be some question as to the desirability of attacking the problem of combined stresses at high temperature until more is known about the laws governing the action of creep under simple stress. Practical considerations indicate the urgent need of such an investigation to establish a more direct relation between creep tests and actual design problems. In many cases a rational basis for design awaits further analysis of creep laws and further verification of such laws under conditions simulating those of service.

Creep tests are necessarily tedious, and the amount of information obtained per unit of time is extremely small. It is generally agreed that creep tests can seldom, if ever, extend to times approaching the expected life of materials in service. In this respect all creep tests may be considered accelerated tests. By extending our knowledge of the laws governing creep under various conditions we may extrapolate the results of short tests made either at higher stresses or higher temperatures with greater assurance of accuracy. To do this for simple stresses only is valuable, but more rapid progress may be made if investigation of the combined stress problem is carried out at the same time.

A careful study of these papers relating to creep and combined stresses will do much to guide future experimental work. In this way much time may be saved by following a definite program of correlated tests. It is hoped that this paper will stimulate further analytical study of creep under the action of combined stresses, together with the experimental work necessary to establish and verify the laws involved.

BENJAMIN MILLER.[25] The author sees fit to discuss only two of the theories of strength: the maximum-shear theory and the maximum-energy-of-distortion theory. Since the maximum-strain theory is still the most popular in the handbooks which are used by the engineer who has not made a special study of the subject, it would be extremely valuable to have a definite statement that the maximum-strain theory has been disproved appear in a paper such as this. Furthermore, if the maximum-energy-of-distortion theory has had general acceptance by those who are specialists, it should be recognized in those publications of the Society, such as the Boiler Code, which are used as a guide by governmental and other regulatory bodies.

The writer is particularly interested in the subject of pressure piping. The Society is now engaged in preparing a Code for Pressure Piping, and one of the things on which agreement has not been reached is the relation between the dimensions of a pipe, the strength of the material in the pipe, and the load required to cause failure. The several theories of strength lead to strikingly different forms for this relation. These are usually presented as pipe-thickness formulas, or as curves showing the ratio of failure pressure to critical stress as a function of the ratio of wall thickness to diameter. Since no such formulas or curves based on the maximum-energy-of-distortion theory have been published, so far as the writer knows, he has prepared those given below.

According to all theories of strength, the inner skin of a tube subjected to internal pressure is first to reach the critical condition as the internal pressure is increased. By the maximum-energy-of-distortion theory the relation is

$$S_0^2 = 3\left(\frac{P}{4K(1-K)}\right)^2 + \left(\frac{(1-2K)^2P}{4K(1-K)} - S_z\right)^2 \quad \text{[A-1]}$$

in which

S_0 = limiting stress in the tension test (yield point)
S_z = average axial tension
K = ratio of wall thickness to external diameter
P = pressure when the inner skin reaches the critical condition.

Two cases which are of practical importance yield expressions which are relatively simple. One is the case in which there is no axial stress, $S_z = 0$. The expression in this case is

$$P_1 = 2K\,S_0\,\frac{2(1-K)}{\sqrt{3 + (1-2K)^4}} \quad \text{[A-2]}$$

As K decreases, the fraction approaches unity, so that for thin tubes Equation [A-2] can be approximated by

$$P_1 = 2KS_0 \quad \text{[A-3]}$$

Equation [A-3] is the familiar Barlow's formula, which is thus seen to be theoretically correct for thin tubes with no axial stress. For showing the difference between the exact and the approximate forms, it is convenient to rewrite [A-2] as

$$P_1 = 2KS_0(1-J) \quad \text{[A-4]}$$

[23] Westinghouse Research Laboratories, East Pittsburgh, Pa. Mem. A.S.M.E.

[24] A. Nadai, "The Creep of Metals," A.S.M.E. New Haven Meeting, June, 1932.

[25] Henry L. Doherty & Co., New York, N. Y.

in which J is a correction to Barlow's formula. J is a function of K; the function is plotted in Fig. 30 for values of K up to 0.1, which includes most commercially important pipe. It is of interest to note that the correction is only $2^1/_2$ per cent for a pipe whose thickness is one-tenth the external diameter.

The second case is that in which the pipe wall carries the load due to the internal pressure on unstayed closed ends. The expression in this case is

$$P_2 = 2KS_0 \frac{2}{\sqrt{3}} (1 - K) \quad \text{[A-5]}$$

For thin tubes the factor $(1 - K)$ differs by a negligible amount from unity, so that approximately

$$P_2 = 1.155\, P_1 \quad \text{[A-6]}$$

Equation [A-6] indicates that the pressure required to bring the inner skin to the plastic state is about 15 per cent greater if the tube has unstayed closed ends than if there is no axial stress. The engineer concerned with pressure piping cannot neglect this important difference between the conclusion reached on the basis of the maximum-energy-of-distortion theory and that on the basis of the maximum-shear theory, which is that the pressure for failure should be the same in the two cases.

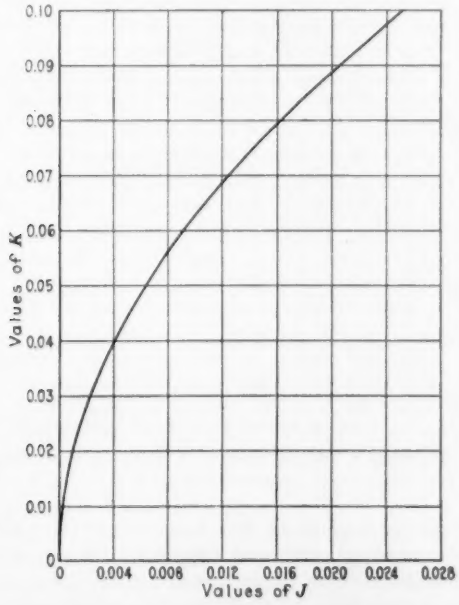


FIG. 30 DEVIATION (J) OF YIELD PRESSURE CALCULATED BY BARLOW'S FORMULA FROM YIELD PRESSURE FOR INNER SKIN CALCULATED BY MAXIMUM-ENERGY-OF-DISTORTION THEORY FOR VARIOUS VALUES OF THE THICKNESS, EXTERNAL-DIAMETER RATIO (K) FOR TUBES WITH NO AXIAL STRESS

Since engineers are reluctant to use theories which are subjects of controversy, the recent practice has been to run internal-pressure failure tests on samples of the pipe to be used in important pipe lines. Such tests have been made by welding heads to the ends of the sample and loading to failure by internal hydrostatic pressure. Observations of the volume of water pumped in and the corresponding pressures could be used to plot a stress-strain curve from which the yield pressure could be determined. The operating pressure was then set at some fraction of the yield pressure. It has been suggested that such a method of determining safe operating pressure be inserted in the Pressure Piping Code referred to above.

In purchasing pipe to be used at 800 lb per sq in., for example, it might be specified that the yield-pressure in such a test be not less than 1200 lb per sq in., which would involve a factor of safety of 1.5. But according to Equation [A-6] the true failure pressure under the operating conditions of no axial stress would be not 1200 lb per sq in., but rather 1040 lb per sq in., and the factor of safety not 1.5 but 1.3.

Further, it might be specified that every length of pipe be tested to 1000 lb per sq in. in the regular pipe-testing machine, in which the pipe is subjected to axial compression. Since axial compression lowers the yield pressure according to the theory, pipe might fail in the routine test which had safely passed the special high-pressure test.

In order to show graphically the change in yield pressure with axial loading, it is convenient to transform Equation [A-1] to

$$(3 + a^4)n^2 - 2(a^2 + a^3)qn + (1 + a)^2(q^2 - 1) = 0 \quad \text{[A-7]}$$

in which

n = ratio of yield pressure to that given by Barlow's formula
q = ratio of axial tension to yield tension
$a = (1 - 2K)$ = ratio of inside to outside diameter.

Equation [A-7] may be represented by a family of ellipses, one for each value of K. Two of these are shown in Fig. 31,

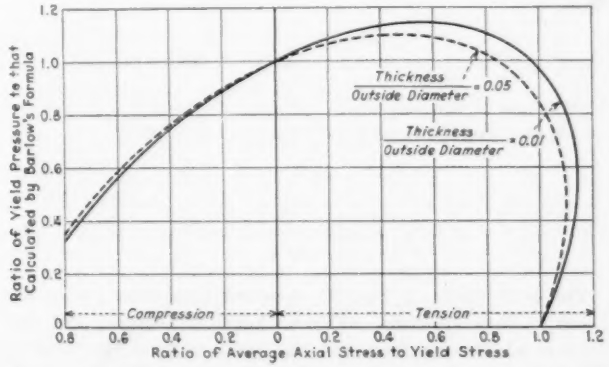


FIG. 31 CHANGE OF YIELD PRESSURE FOR INNER SKIN WITH AXIAL LOADING

the solid line for a thickness $^1/_{100}$ of the external diameter, and the broken line for a thickness $^5/_{100}$ of the external diameter. The curves, or the equation from which they were drawn, may be used to determine the yield pressure under any condition of axial stress if it is known for some other condition of axial stress.

So far, the discussion has been confined to the yield of the inner skin. But the author has pointed out that it is difficult to observe this first yielding, and much easier to observe the yield of the outer skin. The question arises, therefore, as to the pressure which will cause the outer skin to yield. The maximum-energy-of-distortion theory leads to an answer in only the first case—that of no axial stress.

A solution for this case has been given in the author's book "Plasticity," but due to an error in integration the solution there given is not correct. The correct solution is

$$P_1' = \frac{2S_0}{\sqrt{3}} \sin (30° - A) \quad \text{[A-8]}$$

$$a = 1 - 2K = \sqrt{\frac{\sqrt{3}\, e^{\sqrt{3}\,(A - 30°)}}{2 \cos A}} \quad \text{[A-9]}$$

Equation [A-8] may, for convenience, be rewritten as

$$P_1' = 2KS_0(1 + J) \quad \text{[A-10]}$$

in which J is again a correction to Barlow's formula. J is a function of K; the function is plotted in Fig. 32.

By making certain additional assumptions, it is possible to get an approximate solution for the second case—that of no axial plastic strain. This approximate solution is also given in the author's book; in the present notation it is

$$P_2' = \frac{2}{\sqrt{3}} S_0 \log_e \frac{1}{1-2K} \quad \text{[A-11]}$$

This may also be put into a form similar to [A-10]:

$$P_2' = 2KS_0 \frac{2}{\sqrt{3}} (1+J) \quad \text{[A-12]}$$

J as a function of K has also been plotted in Fig. 32.

In the present paper the author gives an Equation [VII] which is said to represent the condition of yielding, and which he has assured the writer in private correspondence is meant to represent the condition for yielding of the outer skin. Equation [VII] leads to

$$P_1'' = 2KS_0 \frac{2(1-K)}{\sqrt{(1-2K)^4 + 3(1-K)^4}} \quad \text{[A-13]}$$

for the case of no axial stress, and to

$$P_2'' = 2KS_0 \frac{2}{\sqrt{3}} \frac{1}{1-K} \quad \text{[A-14]}$$

for the case of the tube with unstayed closed ends. Equations [A-13] and [A-14] can also be written in the form of

$$P_1'' = 2KS_0 (1+J) \quad \text{[A-15]}$$

$$P_2'' = 2KS_0 \frac{2}{\sqrt{3}} (1+J) \quad \text{[A-16]}$$

The solid circles in Fig. 32 represent Equations [A-15] and [A-16]. The agreement between [A-10] and [A-15] and that between [A-12] and [A-16] are very good, especially for values of K not greater than 0.05. This makes it possible to use the author's Equation [VII] as at least a first approximation for the yield of the outer skin under all conditions of axial loading.

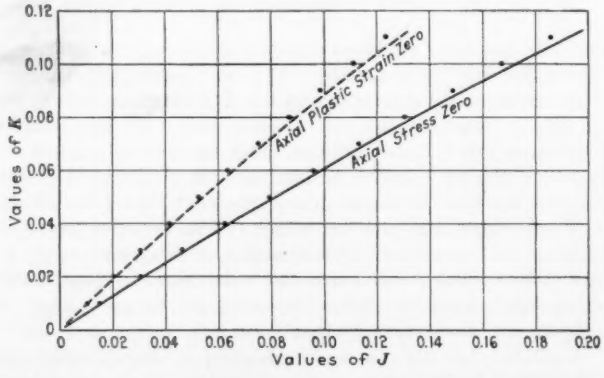


FIG. 32 SOLID LINE: DEVIATION (J) OF YIELD PRESSURE CALCULATED BY BARLOW'S FORMULA FROM PRESSURE CALCULATED FOR YIELD OF OUTER SKIN BY TRIGONOMETRIC FORMULA FOR VARIOUS VALUES OF THE THICKNESS, EXTERNAL-DIAMETER RATIO (K)

Solid Circles Near Solid Line: Deviation (J) of yield pressure calculated by Barlow's formula from pressure calculated for yield of outer skin by Equation [VII] for various values of the thickness, external-diameter ratio (K).

Broken Line: Deviation (J) of $2/\sqrt{(3)}$ times yield pressure calculated by Barlow's formula from pressure calculated for yield of outer skin by logarithmic formula for various values of the thickness, external-diameter ratio (K).

Solid Circles Near Broken Line: Deviation (J) of $2/\sqrt{(3)}$ times yield pressure calculated by Barlow's formula from pressure calculated for yield of outer skin by Equation [VII] for various values of the thickness, external-diameter ratio (K).

In conclusion, the writer would stress again the great practical as well as theoretical importance of this subject, and he suggests that manufacturers as well as users of pipe should continue experiments to test the theory and to determine what modifications must be made because of the differences between the ideal material postulated in the theory and the actual materials which engineers must use.

Equation [VII] in the present notation becomes

$$S_0^2 = 3\left(\frac{P^*(1-K)}{4K}\right)^2 + \left(\frac{(1-2K)^2 P^*}{4K(1-K)} - S_s\right)^2 \quad \text{[A-17]}$$

in which P^* is the pressure when the outer skin reaches the critical condition.

Equation [A-17] differs from [A-1] only by the position of the factor $(1-K)$, which is in the numerator of the first term in [A-17], but is in the denominator of the first term in [A-1]. S_s is the average axial tension in both equations, but it is important to remember that in the case of yield of the outer skin the axial tension is not constant in the cross-section.

To show graphically the change of yield pressure with axial loading, it is convenient to transform [A-17] to

$$\left[\frac{3}{16}(1+a)^4 + a^4\right] n^2 - 2(a^2+a^3)\, qn + (1+a)^2(q^2-1) = 0 \quad \text{[A-18]}$$

in which the symbols have the same meaning as in the similar Equation [A-7]. Equation [A-18] may also be represented by a family of ellipses, two of which are plotted in Fig. 33.

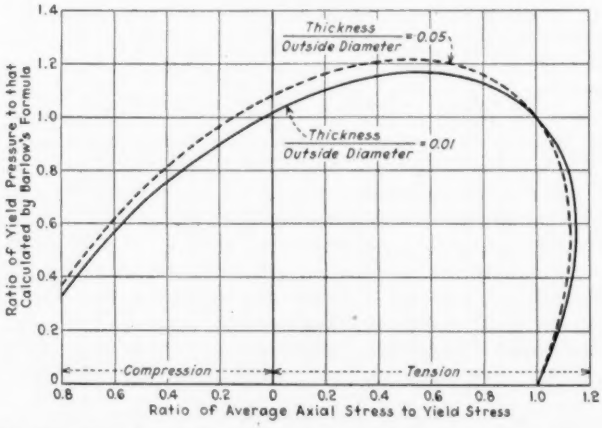


FIG. 33 CHANGE OF YIELD PRESSURE FOR OUTER SKIN WITH AXIAL LOADING

It should be emphasized that Equations [VII], [A-17], and [A-18] are empirical, and must depend for their acceptance on the tests of Lode which are represented in Fig. 18 of the paper. Lode's tests were made on steel tubes having internal diameters of 20 to 30 mm and wall thicknesses of 3 to 3.8 mm. The ratio of thickness to external diameter, the value of K, was in these tests not less than 0.08 and not greater than 0.14; the actual range was possibly less, but cannot be determined from the information given in the paper. On the basis of these tests as shown in Fig. 18 we may provisionally accept Equations [VII], [A-17], and [A-18] for K-values up to about 0.1, as well as Fig. 33.

R. G. STURM.[26] The causes of failure are both fascinating and

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of great importance. Pursuit of them has occupied the minds of many investigators and the equipment of many laboratories. Dr. Nadai has brought together the theories and test results from a number of the most important investigators in such a way that clear comparisons may be made among them.

After carefully studying the various theories and data given, the writer was impressed with the fact that strains have been measured to indicate the beginning of plastic action. Upon the measurement of strains, then, depends the validity of the data used to substantiate the various theories. The strains resulting from stresses are dependent upon the modulus of elasticity and Poisson's ratio for the material in the test specimen. As long as these values remain constant the stresses and strains will be proportional.

If a material exhibits a tendency for the modulus of elasticity to decrease very gradually and for Poisson's ratio to increase simultaneously, readjustments in the strains will take place which under certain combinations may render the change in the ratio of resultant measured strain to computed stress imperceptible, while under other combinations a similar change would be quite noticeable in the measured strain. Thus not only the state of elastic stress but also the state of elastic strain may enter vitally into the interpretation of measurements to verify the applicability of any theory.

In the case of Taylor and Quinney's tests based on the change of internal volume of a tube subjected to direct stress and torsion, one result will be obtained if Poisson's ratio is taken as 0.30, as for purely elastic action, but if it is taken as approaching 0.50, as for completely plastic action, another result will be obtained. Actually Poisson's ratio may vary continuously after the first immeasurably small plastic strain is produced until the metal is yielding freely. This free yielding does not occur at the yield strength in all materials. Precise measurements of volume changes, then, should be interpreted in the light of this gradual change in Poisson's ratio.

It seems to the writer that changes in Poisson's ratio at stresses near the yield point will affect the strains measured. If this is the case, the measured strains should also be interpreted in the light of the change in Poisson's ratio with stress. Since the verification or nullification of any of the theories of strength depends upon measured strains, the proper interpretation of strain measurements is necessary. Perhaps the author has investigated this phase of the problem and could point out to what extent changes in Poisson's ratio at the start of plastic action would affect measured strains.

Author's Closure

The author appreciates the attention and the criticism for his brief summary of the critical conditions for certain types of failure due to yielding, and he will attempt to reply to the various valuable remarks brought out during the discussion as far as the limited space will allow.

Mr. Bethune points to an interesting effect—namely, to the results which will be observed when taking stress-strain readings while a material yields by a repeated action of stresses in cycles. This is a subject about which probably not quite sufficient information is yet available, particularly when the principal axes of stress corresponding to the principal stresses σ_1, σ_2, σ_3 are changing their orientation with respect to the strained body. The elastic anisotropy in the material due to the preceding plastic deformation was probably not very pronounced, because it needs probably much larger plastic strain (such as could be produced by cold rolling or wire drawing only) to produce a preferred orientation of the crystal grains. The strains to which the specimen was subjected in the tests mentioned were, however, much smaller.

The author is indebted for the remarks of two of the discussers, who have indicated the need and desirability to include in the present considerations also the phenomena of slow creep of metals, which, as stated in the paper, had to be omitted, the subject having been confined to that kind of plasticity of metals in which the speed of deformation has not a greater influence on the yield stress. The relation suggested by Dr. Hencky (in the second place) for the expression of a law of creep cannot quite adequately express the known physical facts, as has been pointed out by the author on a previous occasion. The reason is simply because continuity demands that the function $\sigma = f(v)$ expressing the relation between stress σ and rate of creep $v = d\epsilon/dt$ for quite small velocities v should be an odd one around the point $v = 0$. Since this point has been raised by Dr. Hencky, it may perhaps be remarked that another simple function will at least satisfy three conditions assumed to be of importance for certain creep phenomena observed in metallic materials—namely, (*a*) a logarithmical speed law connecting stress σ and rate of creep $v = d\epsilon/dt$ at comparatively large rates of creep; (*b*) a linear relation for quite small stresses or rates of flow, and finally, (*c*) the condition of continuity according to which σ changes into $-\sigma$ if v changes its sign. A creep law which will satisfy these three conditions is (σ stress, $v = d\epsilon/dt$ creep rate):

$$v = v_1 \sin h \frac{\sigma}{\sigma_1}$$

v_1, σ_1 being constants. The last equation leads, for small rates of creep v, to the law of pure viscous flow

$$\sigma = \frac{\sigma_1}{v_1} \cdot v$$

with the "viscosity coefficient" $\eta = \sigma_1/v_1$, and for large creep or strain velocities v the same expression converges toward the function

$$v = \frac{v_1}{2} e^{\frac{\sigma}{\sigma_1}}$$

or

$$\sigma = \sigma_1 \log \frac{2v}{v_1}$$

to the logarithmical creep law.

An equivalent of the amount of elastic stored-up energy in a cold-worked metal has been determined by various investigators. As far as the author recalls, it has been probably first determined by W. Hort in Goettingen some 25 years ago, who measured the amount of heat produced by permanently stretching steel bars in a calorimeter by tension. He found that less heat was produced than would correspond to the thermal equivalent of the work of deformation. G. I. Taylor and W. S. Farren[27] more recently arrived at the conclusion that for steel about 13 per cent of the total work of stretching does not reappear as heat and becomes latent. The increase of internal energy of the material amounted to 13.5 per cent of the work done in the case of steel, 8.5 per cent for copper, 7.5 per cent for aluminum, and 5 per cent for single crystals of aluminum. One remarkable result of the tests by Taylor and Farren was that the ratio of the increase of internal energy to the work of deforming the bars was about the same during the gradual stretching, so that this increase of energy does not appear to have a relation to the amount of cold work or strain hardening.

Mr. McVetty's valuable remarks deserve the support of those interested in a better knowledge of the general physical laws

[27] Proc. Royal Soc. London, Ser. A, vol. 107, 1925, p. 422.

describing the creep phenomena in metals at elevated temperatures. It can be hoped that on questions of the kind mentioned by Mr. McVetty which have not yet found satisfactory treatment, such as creep under combined stress etc., further work by engineers will be done in the near future on account of the various applications from which these questions originate.

To Mr. Sturm's observations, that the strains (which accompany the beginning of plastic actions) were utilized in the determining of the limiting conditions of stress, it may perhaps be replied that in investigations of the kind quoted in the paper it has been indeed the practice not only to measure the forces alone, but also to register the strains together with the stresses. This has decided advantages. The question which is here of importance, and which has also been raised in some of the remarks made by Mr. B. Miller, is whether the right quantities (stresses) upon whose values the reaching of plastic limits depends have been actually sharply enough determined by the tests or not. When using tubes of the sizes mentioned and observing the pressures or forces when the test pieces were yielding *continuously* through the whole wall thickness, it seems to the author that it should not be too difficult to establish the correct relations between observed force resultants and the specific average yield stresses to determine these latter properly. An exact knowledge of the elastic constants of the material does not seem necessary for this purpose; the stresses, however, have to be corrected with regard to the permanent deformations the tubes will have suffered during the gradual yielding.[28] In connection with this point, perhaps mention should be made that the simple expressions such as Equations [2] and [3] for the stress-strain relations can only be considered as expressions for these as long as the elastic parts of the strains are negligible or small compared to their plastic parts. These relations can certainly not be exact expressions of the stress-strain relations if the plastic parts are of the same order of magnitude as the elastic parts or even smaller than these latter.

Coming now to Mr. B. Miller's remarks in which he notes that the author sees fit to discuss only two of the theories of strength, it was the latter's understanding that among the various rules which have been suggested in the course of time and in former years, there were a few which would not satisfy certain fundamental conditions, such as, for example, the rule according to which a strength theory which will lead to a *closed* surface of yielding $F(\sigma_1, \sigma_2, \sigma_3) = 0$ can be discarded at once. A limiting state of stress at which yielding starts can be represented by a point P having the rectangular coordinates σ_1, σ_2, σ_3 in space, by these letters the principal stresses again being designated. As stated in the paper already, it follows at once that all points P representing possible states of stress at a plastic limit will be found along a certain surface: "the surface of yielding" $F(\sigma_1, \sigma_2, \sigma_3) = 0$ characteristic for a given material. As by three equal pressures $\sigma_1 = \sigma_2 = \sigma_3 = -p$ yielding cannot be produced in a material, a point in the σ space having this property cannot be found along the limiting surface of yielding. This latter must therefore be an open surface around the space diagonal $\sigma_1 = \sigma_2$, $\sigma_2 = \sigma_3$ toward its negative side. The theories of strength which lead to intersection points of their surface $F(\sigma_1, \sigma_2, \sigma_3) = 0$ with the straight line $\sigma_1 = \sigma_2$, $\sigma_2 = \sigma_3$ in space in the negative quadrant have to be rejected. To these belong both the maximum stress or maximum strain theories, while the maximum shear theory and the constant octahedral shearing-stress theory lead to open surfaces.

[28] When the permanent strains are already predominant, but still not too large strains (of an order of magnitude of a few per mille or so), Equation [2] being a satisfactory expression of the facts, this equation can be used for correcting for the permanent strains by which the original dimensions of the tubes have changed.

The author was particularly interested to learn from various conversations he had with Mr. Miller that the economical designing of the long pipe lines which are used for transporting oil in the United States through thousands of miles, makes it desirable to utilize the material to its utmost safe stress limits and that a pipe line designed according to Equation [15], or *the condition of constant octahedral shearing stress* $\tau_n = \sqrt{2} \cdot \sigma_0/3 = 0.47\sigma_0$ (σ_0 being the yield stress for pure tension), will be 16 per cent lighter (and consequently so much cheaper) than if the wall thickness is computed by using the maximum shear criterion.

Mr. Miller emphasizes the need to distinguish carefully between the various cases which might occur when thick-walled tubes are subjected to high internal pressure, particularly with regard to the conditions imposed upon the tube at the ends. He also calls attention to the considerable difference which will be found between the pressures in such tubes (the wall thickness of which is perhaps of the order of a tenth of the mean diameter) for first yielding at the inner surface of the tubes and the pressures for a complete yielding of the tubes. In tubes having thicknesses similar to those tested by W. Lode, the pressures producing *first yielding* will be considerably lower than those producing *complete yielding*. As the instant at which a tube first starts to yield in an infinitely thin layer at the inner surface cannot be detected by any method, extensometers fastened to the tube can be utilized only to make visible the instants at which yielding has reached the outer surface of the tube or proceeds through the entire tube. W. Lode has compared the pressures and axial forces only in these latter cases, which follows clearly from a comparison of the stress-strain curves he has observed while his tubes were stretching.

It may appear paradoxical that an equation such as Equation [VII] which has been derived by using the formulas [I]–[III] based on the theory of elasticity could furnish the correct values of the *average* stresses when the tubes are yielding through their whole thickness. This is, however, the case. In fact, for the tubes of the sizes quoted, the curves representing the radial, the tangential, and the axial stresses σ_r, σ_t, σ_z as functions of the variable radii r can all be replaced by straight lines, and it is perhaps noteworthy that this is true in both cases—namely, when the strains are purely elastic and also in the case of complete yielding. If the wall thickness δ is of the order of a tenth of the mean diameter $2r_0 = r_1 + r_2$ or smaller (r_1 inner, r_2 outer, r_0 mean radius of tube), it is convenient to express the stresses σ_r, σ_t, σ_z not as functions of the variable distance r of a point from the axis of the tube, but as functions of another variable

$$x = r - r_0 \qquad \text{[B1]}$$

x being the distance of a point from the center of the wall thickness δ, and to develop all expressions into powers of the small quantity x, retaining only first-order terms. In other words, for tubes with $\delta \leqq 1/10$, the stresses will, with sufficient accuracy, become expressible by linear functions of x. In fact, an elastic stress distribution in a thick-walled tube as given by Equations [I]–[III] might be rewritten in the form:

$$\left.\begin{aligned} \sigma_r &= -\frac{p}{2}\left(1 - \frac{2x}{\delta}\right) \\ \sigma_t &= -\frac{p}{2}\left(1 + \frac{2x}{\delta}\right) + \frac{pr_0}{\delta} \\ \sigma_z &= -\frac{p}{2} + \frac{pr_0}{2\delta} + \frac{P}{2\pi r_0 \delta} \end{aligned}\right\} \qquad \text{[B2]}$$

Taking now, for example, the case of a tube closed at both ends subjected to internal pressure p only, the plastic stress distribu-

tion for complete yielding, using the theory of constant octahedral shearing stress, may also be established for a comparison. It is given by

$$\sigma_r = -p - R, \quad \sigma_t = -p - R + \frac{2\sigma_0}{\sqrt{3}}$$

$$\sigma_z = -p - R + \frac{\sigma_0}{\sqrt{3}} \ldots\ldots [B3]$$

where R stands for

$$R = \frac{2\sigma_0}{\sqrt{3}} \log \frac{r_1}{r} \ldots\ldots [B3a]$$

The pressure p producing complete yielding is found to be equal to

$$p = \frac{2\sigma_0}{\sqrt{3}} \log \frac{r_2}{r_1} \ldots\ldots [B4]$$

Expanding again into powers of x:

$$\log \frac{r_1}{r} = \log \frac{1 - \frac{\delta}{2r_0}}{1 + \frac{x}{r_0}} = -\frac{1}{r_0} \cdot \left(\frac{\delta}{2} + x\right)$$

the expressions [B3] become

$$\sigma_r = -\frac{p}{2}\left(1 - \frac{2x}{\delta}\right), \quad \sigma_t = -\frac{p}{2}\left(1 - \frac{2x}{\delta}\right) + \frac{pr_0}{\delta},$$

$$\sigma_z = -\frac{p}{2}\left(1 - \frac{2x}{\delta}\right) + \frac{pr_0}{2\delta} \ldots\ldots [B5]$$

and the pressure p for complete yielding can now be computed also from the equivalent formula to [B4]:

$$p = \frac{2\sigma_0}{\sqrt{3}}\left(\frac{\delta}{r_0} + \frac{\delta^3}{12r_0^3}\right) \ldots\ldots [B6]$$

From [B5] the average stresses, or what is the same as the stresses for $x = 0$, are found:

$$\sigma_r = -\frac{p}{2}, \quad \sigma_t = -\frac{p}{2} + \frac{pr_0}{\delta}, \quad \sigma_z = -\frac{p}{2} + \frac{pr_0}{2\delta} \ldots [B7]$$

These are exactly the same expressions as would have been obtained should the stress expressions (Equation [B2]) derived for a purely *elastic* state be used with the pressure p chosen according to [B4] or [B6]. In other words, if the *average* stresses of an elastic stress distribution (as given by the formulas [VI]) in a thick-walled tube are adjusted so that in addition they satisfy the plasticity condition (Equation [23]), the corresponding pressure will be about the same as the one at which yielding in the tube would have developed to the (outer) surface.[29]

Comparing Mr. Miller's computations in Fig. 32, it appears from the present point of view that an expansion into powers of x or δ up to the first order is completely sufficient for computations of the stresses in thick-walled tubes, both in the elastic as in the plastic state.

The author could not verify the error found by Mr. Miller in an expression published by the author on a former occasion, as details are not completely given; to his justification he may only add that a few simple and important special cases which are included in the former results obtained by a supposedly erroneous integration checked perfectly, which does not make it to him so probable that the integration would have contained errors.

The author is particularly indebted to Mr. Miller for having shown in detail that, using his own methods when constructing certain curves (such as indicated in Fig. 32) and comparing them with points which were computed by Mr. Miller using the author's Equation [VII], a very good agreement was also obtained. He was wondering a little, however, that notwithstanding these facts and a few of the other reasons which the author endeavored to bring out in his paper infav or of Equation [VII] and of the physical assumptions on which it was based, Mr. Miller emphasizes in his final remarks the "empirical" nature of this particular condition or equation, while the nature, the content, or form of some of the other formulas expressing in terms of stress the statements contained in the maximum shear or the maximum strain theory did not seem to give him reasons for any remarks. With Messrs. Sturm and Miller and the other discussers of the report, the author agrees completely that further critical discussion of the various points which were brought out during this discussion will probably be useful. For example, the lack of isotropy in actual rolled material and the case in which the plastic strains become comparable to the elastic ones deserve more attention. Some new attempts to test particularly this last case in tubes can perhaps be reported later.

[29] Equations [B2] can, by the way, also be used to compute the pressure at which first yielding starts. If the ratio $\delta/r_0 = 1/10$ according to plasticity condition [23] and using the exact (elastic) theory, first yielding would start at a pressure

$$p_1 = 0.191\sigma_0$$

while the approximate formula [B2] gives a value:

$$p_1 = \frac{\sigma_0}{\sqrt{27}} = 0.192\sigma_0$$

Similarly complete yielding in a tube with $\delta/r_0 = 1/10$ has to be expected, according to the exact formula [B4], at a pressure

$$p = \frac{2\sigma_0}{\sqrt{3}} \log \frac{11}{9} = 0.232\sigma_0$$

While the approximate formula [B6] gives a pressure in good agreement:

$$p = \frac{2\sigma_0}{\sqrt{3}} \cdot \frac{\delta}{r_0} = 0.231\sigma_0$$

APM-55-16

Working Stresses

By C. RICHARD SODERBERG,[1] PHILADELPHIA, PA.

This paper gives a general review of the fundamental aspects of the problem of strength and safety of machine parts. After an introductory discussion of the nature and scope of the problem, it offers suggestions for selecting working stresses for those conditions of applications of materials for which adequate knowledge is available.



THE problem of mechanical strength is one of the most important considerations in the dimensioning of machine parts. The designer arrives at a practical solution of this problem by (1) an appraisal of the conditions of service in so far as stress is concerned, (2) an appraisal of the strength of the materials under these circumstances, and (3) a decision as to the margin he is willing to allow between actual service conditions and failure. This last decision is expressed through a set of conventions and rules for establishing *working stresses*, that is, a condition of service which, with regard to the limitations of the strength of the materials, gives the desired insurance against failure. The object of the present paper is to review the fundamental aspects of this process in the light of our present knowledge of our engineering materials.

In the appraisal of the conditions of service in so far as stress is concerned, an abstraction is made from other, perhaps equally important, aspects of the service. Of these may be mentioned appearance, considerations of standard parts and stock sizes, corrosion and wear, problems of assembly and repair, and so on. In the present discussion all of these extraneous considerations will be neglected, but their importance must not be underestimated. Not infrequently, they justify an apparently nonchalant treatment of the problem of strength. In addition, the present discussion will be restricted to those conditions of service where phenomena of structural instability, such as buckling, do not occur. Finally, it will cover only the general problems encountered in the machine industry. It is fully realized that many problems in other fields, such as, for example, civil engineering, require additional specialized treatment.

In endeavoring to define the strength of a material, it is important to remember that strength can be defined only against the background of a certain arbitrary method of testing. In general, this method of testing discloses some form of "failure," which is then selected by the designer as the contingency against which he wants to guard. In a few rare cases the conditions of test happen to agree rather closely with the conditions of service; this happy state of affairs is a rare exception, however. Most frequently the appraisal of strength for a certain actual condition of service represents a wide extrapolation from the conditions of testing.

Granting that the designer's reasoning is sound up to this point, the next step is an evaluation of the risk that he finds it advisable to assume with regard to a possible occurrence of the predicted form of failure. If he is in the fortunate position of being able to tolerate a small percentage of failures, experience will soon point the way to an economical ultimum. In most cases, however, failure spells disaster, and progress in industry has made the consequences of such disasters ever drier.

The vague nature of the process of selecting working stresses must be kept in mind, however, whenever there are temptations in the direction of theoretical elaborations. It is not unlike the process of appraising the soundness of a monetary investment where there is also a problem of reaching a balance between returns and predictable chances of failure. It will be apparent, from the following discussion, that there are as yet very wide gaps in the field of required knowledge, gaps that at present must be filled in by good intuitive judgment.

NOMENCLATURE

Symbol	Units	Designation
a	in.^{2}	Area of cross-section
E	lb/in.^{2}	Modulus of elasticity ($\partial\sigma/\partial\epsilon_e$): E', a value modified for external flexibility
E_p	lb/in.^{2}	Modulus of strain hardening ($\partial\sigma/\partial\epsilon_p$): E_p', a value modified for external flexibility
k	lb/in.	Rigidity (scale) of spring
l	in.	Length
n	1	Factor of safety
p	lb/in.^{2}	External pressure
u	1	Factor of utilization
t	hr	Time: t^*, time parameter
v	1/hr	Strain rate ($\partial\epsilon_p/\partial_2$): v_0, v_1, material constants
ϵ	1	Strain: ϵ_e, elastic strain; ϵ_p, plastic strain; ϵ_0, initial value of strain
γ	1	Shear strain
σ	lb/in.^{2}	Normal stress: σ_i, general limit; σ_e, endurance limit; σ_y, yield point; σ_u, ultimate strength; σ_w, working stress; σ_0, initial stress or steady stress: σ_1, σ', σ'', material constants; σ^*, σ^{**}, stress parameters
τ	lb/in.^{2}	Shear stress: τ_w, working stress

THE MEASURABLE PHYSICAL PROPERTIES OF MATERIALS

The art of testing the physical properties of engineering materials is an extensive one, but the number of tests which actually determine mechanical "strength" is very small. There are only two tests that can be used directly for predicting failure quantitatively, if the word failure is considered in a somewhat restricted sense. These are the tensile test and the fatigue test. No criticism is implied here of the many other forms of tests,

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[1] Manager, Turbine Apparatus Division, Westinghouse Electric and Manufacturing Company, South Philadelphia Works. After receiving his university education, Mr. Soderberg was graduated as naval architect from Chalmers' Institute of Technology, Göteborg, Sweden, in 1919. He received a fellowship from the American Scandinavian Foundation in 1919 for the study of shipbuilding in the U.S.A., and spent 1919–1920 at M.I.T. and the University of Michigan. He was employed as draftsman by the New York Shipbuilding Corp. in 1921, and in 1922 by the Westinghouse Company, where he worked until 1928 on vibration and other dynamical problems connected with heavy traction railway equipment and large power machinery. After two years with the Swedish General Electric Company in the design of large turbo-generators and turbines, he returned to the Westinghouse Company in East Pittsburgh, and was transferred to his present location in May, 1931.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

such as impact tests, bend tests, hardness tests, etc. The purpose of the latter group of tests, however, is either to establish or confirm results that eventually must be derived from the tensile test, or to measure properties other than actual strength. The conventional bending fatigue test would undoubtedly be replaced to advantage by an alternating tensile test if the mechanical difficulties associated with such a test were not too great. The same is probably true of the impact test. In view of this we may regard the fundamental knowledge of the materials, accessible to the designer, as the results obtained by applying a principal stress along the axis of a cylindrical bar. Practically everything else is derived knowledge. Thus, while in actual practice, and in the theories of elasticity and plasticity, we treat stress or strain in the most general three-dimensional aspect, the measurable properties concern practically only the extremely specialized case of a cylindrical bar in tensile or compressive stress.

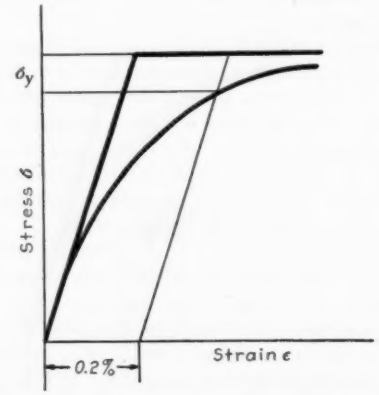


FIG. 1 DEFINITION OF YIELD POINT

The tensile test also furnishes the material for defining the term "failure." When applied to the conventional tensile test at normal temperature, this may mean an actual break of the test piece (ultimate strength), or it may mean a sudden increase in strain without a correspondingly large increase in stress (yield point). It may also mean a departure from the straight-line law between stress and strain (proportionate limit), or it may mean a cessation of the elastic properties (elastic limit).

It is gradually becoming recognized that of all these possibilities, the yield point and the ultimate strength offer the most rational limits of failure. The former applies to ductile materials and the latter to the brittle ones. When, for ductile materials, the stress-strain curve does not exhibit a definite discontinuity, the stress corresponding to a permanent deformation of 0.2 per cent, Fig. 1, has become a widely accepted definition of the yield point.

Even for ductile materials, the results obtained by applying stresses above the yield point are of interest in furnishing a definition of ductility. The general configuration of the stress-strain curve is, in fact, one of the most important sources of information concerning plastic flow. The arbitrary definition of strength and failure just given may sometimes appear to break down in the light of the common effects of strain hardening, after-flow, elastic hysteresis, etc. In such an event, however, it is usually because aspects other than stress assume dominating importance. This is the case encountered with annealed copper at normal temperatures, or of soft steel at elevated temperatures.

When the test piece is subjected to alternating stress, the failure assumes the character of brittle failure under steady stress. The endurance limit is now the criterion of strength, and may be taken as the stress amplitude at which a very large number of cycles ($> 10^7$) are required to produce failure.

When the tensile test is conducted at elevated temperatures, the mechanism of failure changes completely, at least for the ductile materials, which are the only ones for which reliable tests are available. The speed of deformation now assumes a dominating importance, and the test has no meaning unless associated with a strain rate. With this added qualification, the nomenclature established for the tensile test may be retained. Most structures are such, however, that their operativeness depends on the preservation of physical dimensions and initial stresses. Thus, while no actual failure may be involved, the stresses must be limited to values that will preserve the geometrical configuration of the structure within certain established limits. No single-valued quantity of real importance can be extracted from these tests; our only recourse in this case is to extrapolate the results of the tests to the actual conditions and base the stresses on the predicted deformations.

THE NATURE AND SCOPE OF THE PROBLEM OF WORKING STRESSES

This restricted range of the measurable properties of engineering materials has injected a considerable amount of mechanical routine into the procedure of selecting working stresses. When this procedure is viewed from a more general standpoint, however, the following aspects must be considered:

1 An evaluation of the stresses or strains existing in the structure
2 A decision with regard to the methods of test which will reveal the physical properties of the material most relevant to the conditions of service
3 A decision with regard to the type of failure which is likely to occur
4 An extrapolation from the actual tests to the actual conditions of service
5 A choice of margin between the actual conditions of service and those that would produce failure thus defined.

The first aspect, the evaluation of the stresses or strains, is the most obvious and straightforward, and will be almost entirely disregarded in the present paper. This problem offers no difficulty beyond mathematical complexity, provided that the conventional and sometimes questionable premises of the theory of elasticity may be assumed to hold. These are homogeneity of the materials and adherence to Hooke's law. It is true that these are never wholly fulfilled, but the deviations rarely cause real difficulties. At any rate, the uncertainties introduced in this manner are of secondary importance when compared with those encountered in the remainder of the problem.

Due to obvious limitations, it is not always possible to continue through the steps 2 to 5 in the manner suggested. Every problem of strength would become the subject of controversy and receive individualistic treatment. This is undoubtedly desirable from a scientific point of view, but would be derogatory to the normal requirements of business efficiency. The only solution of this difficulty is an orderly classification of the available knowledge on the subject. Such a classification can be accomplished only by somewhat dogmatically settling certain points of controversy. This is true of all branches of applied mechanics, but in most other fields the progress has penetrated beyond the point where alternative interpretations differ seriously.

It is proposed, therefore, to group the available information on the subject of strength in the following tentative classification:

1 Types of Materials
   - *a* Brittle materials
   - *b* Ductile materials

2 Operating Temperature
   - *a* Normal temperatures
   - *b* Elevated temperatures

3 Mode of Stress Application
   - *a* Steady stress
   - *b* Variable (alternating) stress
   - *c* Impact stress

4 Type of Stress
   - *a* Tension and compression
   - *b* Pure shear
   - *c* Combined stress.

Practically every group of this classification can be made the subject of controversy, and none, except the last, can be defined except by arbitrary limits.

Considering first the distinction between brittle and ductile materials, it must be appreciated that both the mode of stress application and the type of stress are involved. Materials which normally fall under the classification of brittle materials may be made to deform plastically under certain conditions, so that the terms "brittle" and "ductile" refer to states rather than types of materials.[2] Many objections of this nature disappear, however, when it is realized that most of our useful engineering materials, which are important from the point of view of strength, fall into one or the other of these classifications. One could, in fact, define brittle materials as "cast iron like" and ductile materials as "mild steel like" and arrive at a fairly satisfactory classification for general practical purposes. Furthermore, practical conditions are such that the brittle materials play a very unimportant rôle in applications where strength is of real importance. In what follows, therefore, very little attention will be given to brittle materials. It is tentatively proposed that the limit be set at 5 per cent elongation in the standard tensile test piece.

The operating temperature is a somewhat arbitrary specialization of a more general heading which might have been denominated "Conditions of Use." It is of profound significance in the machine industry, however, because the higher temperatures inject an entirely new mechanism of failure. The boundary between "normal" and "elevated" temperatures is necessarily vague, but eventually a satisfactory definition will undoubtedly be formulated by reference to the melting point of the material.[3] In the present connection, elevated temperatures are defined as that temperature range at which the permanency of physical dimensions and initial stresses limit the utilization of the material. For most of the ferrous alloys this limit may be placed at about 500 F.

The mode of stress application is likewise somewhat vague. The distinction between impact and other forms of stress offers the least difficulty because it is already established in rigid dynamics. However, the case of impact stress is rather rare, and very little information on the prediction of failure under impact is available. The conventional impact test cannot be used for this purpose except for qualitative comparison of different materials. Just when a stress is steady or variable, on the other hand, may be the subject of much controversy. In actual practice, all stresses undergo many changes during the life of a machine or a structure. The frequency of a fatigue test is not of fundamental importance, within wide limits, so that the real criterion is the total number of reversals to which the structure will be subjected. If this is of the order of 10^7, the stress should be regarded as variable; if it is of the order of 10^3, it may be classified as steady. No more specific line of demarcation can be drawn, except for individual materials. It may be necessary to point out that the case of pulsating stress is to be regarded, in this classification, as a special case of superposition of steady and alternating stress and not as a separate mode of stress application.

The "type of stress" is established by the existing conventions in the theory of elasticity. It is possible to argue that other specialized types of stress, such as hydrostatic pressure, may be included in the classification. Such cases, however, are tacitly included under "combined stress."

As yet this scheme of classification is inadequately fortified with reliable knowledge. Certain combinations, such as the case of impact stress, brittle materials under variable stress or elevated temperatures, and ductile materials under variable stress and elevated temperatures, are practically unknown, and until more knowledge is available, they must be kept out of discussions on working stresses. Practical applications, where necessary, must be made with extreme conservatism. The remaining parts of this paper will be devoted to a brief discussion of those combinations for which fairly adequate knowledge is available at the present time.

MATERIALS AT NORMAL TEMPERATURES

(*a*) Factor of Safety and Utilization of Strength

Let it be assumed that the failure of the element in question depends only on a certain stress σ, and that failure occurs when this stress reaches the value σ_i. The working stress σ_w will then be defined by the relation

$$\sigma_w = u\sigma_i \qquad [1]$$

where u is the factor of utilization. It is evident that $0 < u < 1$. The more common factor of safety is defined by $n = 1/u$. Failure is characterized by $u = n = 1$.

The principal object of the present paper is to suggest rules for defining rationally the quantities σ_i and σ_w. Once this is achieved, the designer will be able to interpret his experience in a rational manner. The actual choice of margin between working stress and failure, the actual factor of safety, must remain the designer's own prerogative. This process, however, involves certain frequently recurring fundamental considerations which it may be profitable to review.

The accuracy by which σ_i and σ_w can be determined is of fundamental importance. It is evident that the more accurately they are known, the more fully can the available strength be utilized with safety. The inaccuracy of σ_i concerns not only the errors of measurements but also those of judgment in definition.

The value of σ_w which is used in [1] often represents an extreme condition of service which is, in itself, an emergency. Common examples of this are the conditions under test pressures for containers and overspeed tests for rotating machinery. Although complete assurance against failure must be obtained in such cases as well, it is not necessary to provide the same degree of safety as in those cases where σ_w represents permanent operating conditions.

σ_w may also represent an assembly stress which will drop to a lower value should yielding start. This is the case of a rotating disk shrunk on its shaft. Obviously, the assembly stress may be raised much above the stress produced directly by the centrifugal force.

These examples suggest a range of values of u corresponding

[2] Nadai and Wahl, "Plasticity," Engineering Societies Monographs, p. 58.

[3] "Homologous temperatures," first introduced by Ludwik, are absolute temperatures of equal proportions of the melting point and may be used for this purpose. See Nadai's "Plasticity," p. 275.

to a uniform degree of economic safety. The question of giving a value of u to each combination is a matter to be decided by the responsible designer. The following may be stated with regard to the extreme limits: In the most favorable case of accurately known stresses and material properties, it should be possible to utilize 75 to 80 per cent of the strength of the materials, that is, $u = 0.7$ to 0.8. In the other extreme condition, such as may occur in cases of variable stress, it should not be necessary to go below $u = 0.25$ to 0.30.

(*b*) Ductile Materials Under Steady Stresses

The yield point in tension, as obtained from the ordinary

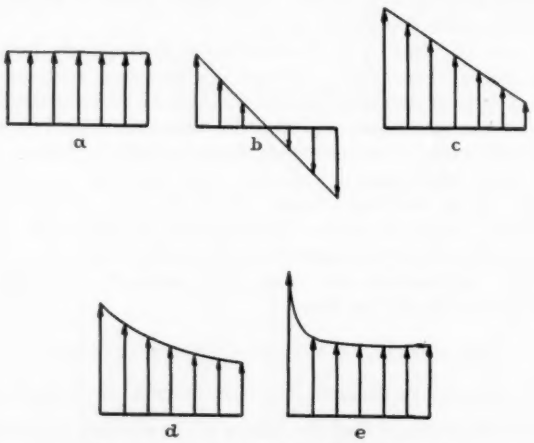


Fig. 2 Types of Stress Distribution

tensile test, furnishes a unique definition of failure. The working stress may thus be expressed as

$$\sigma_w = u\sigma_y \quad [2]$$

This conception enables the designer to effect an economical use of his materials whenever the conditions correspond to those of the tensile test. However, this is seldom the case, so that the above convention must be discussed in the light of the possible deviations from the ideal conditions of the tensile test.

The first deviation of importance occurs when the stress is of the tensile or compressive type but varies in magnitude over the cross-section (Fig. 2). This simple deviation from the test conditions opens an important field of controversy. The general question of combined stresses also enters into the controversy, but it is only in special cases that this aspect is of importance. An example of such a case is the shear stress occurring in the web of an I-beam. This aspect will be discussed in connection with the general case of combined stresses. Apart from this, however, it is by no means self-evident that the various cases shown in Fig. 2 should be covered by the same limiting stress, nor is such a conclusion supported by experiment.

All that can be said with certainty about these cases is that when the stress variation is "sharp," Fig. 2(*e*), only the mean stress has real influence upon failure. This condition is usually referred to as "stress concentration," which may thus be neglected in ductile materials under steady stress. In all the other cases the maximum stress must be considered in predicting failure. The actual importance of the maximum and the mean stress is associated with the external conditions of yielding, which must eventually determine whether it is a question of "sharp" or "moderate" variation of the stress over the section.

Other deviations of importance may be listed under the heading of combined stresses. This presents a more difficult and somewhat controversial extrapolation from the tensile test. The more recent investigations on the subject appear to settle the controversy in favor of the Mises-Hencky interpretation of failure.[4] When considering the purely practical aspects of the problem, however, the simpler maximum-shear theory will cover the facts with sufficient accuracy. Under this interpretation, the failure is attributed to slip in the planes of maximum shear. In the ordinary tensile test the failure at the yield point is actually a shear failure in planes inclined 45 deg to the axis of the specimen. The maximum shear stress is one-half the maximum tensile stress, so that for pure shear Equation [2] is modified into

$$\tau_w = u\frac{\sigma_y}{2} \quad [3]$$

The same reasoning is easily extended into combinations of tension and shear, Fig. 3(*a*), one of the most common of which is the case of bending and torsion. Instead of a single-valued working stress, there is now a case of superposition of one stress upon the other. The maximum-shear theory predicts failure for combinations falling on the ellipse $(\sigma/\sigma_y)^2 + 4(\tau/\sigma_y)^2 = 1$, Fig. 3(*b*). The utilization of the portion u of this strength is represented by another ellipse, contracted in scale by the factor

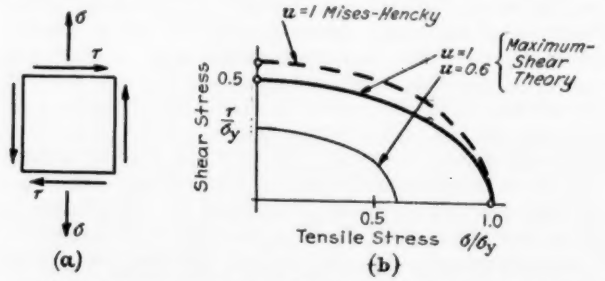


Fig. 3 Superposition of Shear and Tensile Stress

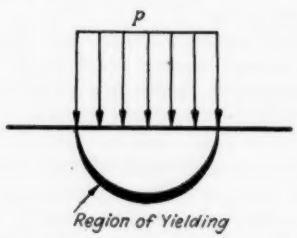


Fig. 4 Failure Under Contact Pressure

u. On this basis the working stresses are determined by the relation

$$\sqrt{\sigma_w^2 + 4\tau_w^2} = u\sigma_y \quad [4]$$

This particular case may be used to advantage to point out the difference between the maximum-shear theory and the more correct Mises-Hencky theory. In Fig. 3(*b*), the latter is represented by another ellipse,[5] the maximum ordinate of which is $1/\sqrt{3} = 0.577$, instead of $1/\sqrt{4} = 0.5$.

The results obtained from the tensile test are generally accepted as holding for compression as well. For practical purposes this is true for most of our ductile materials up to the point where structural instability becomes of importance. There is another form of compression, however, which presents a much more complicated phenomenon and also has real practical importance. This is the problem of contact stresses between sur-

[4] Nadai, "Plasticity," chap. 13.

[5] See paper in present symposium by Dr. Nadai on "Theories of Strength."

faces in machine parts. A great deal of experimental and theoretical work has been done on this subject and the results have practical value. In the common case of uniform pressure along a strip of material, Fig. 4, the predicted yielding will occur along the semicircle shown. The value of the pressure at which yielding starts is $p = \pi/\sqrt{3}\ \sigma_y = 1.81\ \sigma_y$, and is obtained by assuming that there are no friction forces on the contacting surfaces.[6] In reality, this friction is of great importance, since it restricts the lateral contraction. For this reason, actual failure will not take place for pressures of the order of $2\sigma_y$, a value which is thus on the safe side for practical applications.

It may be worth while to point out that contacts of moving surfaces, such as occur in gears or bearings, present an entirely different problem, and one which is beyond the scope of the present paper.

(c) Ductile Materials Under Variable Stress

For conditions similar to those prevailing in the conventional fatigue bending test, the endurance limit σ_e gives a unique definition of failure, so that the working stress may be expressed as

$$\sigma_w = u\sigma_e \qquad [5]$$

The inaccuracies of extrapolation to other circumstances are in this case now much greater than in the previous one.

The maximum stress now plays a much more dominating rôle even in the case of stress concentration. However, there

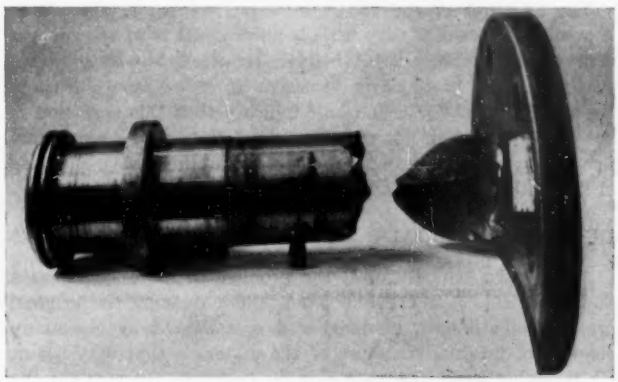


Fig. 5 Failure Under Variable Torsion

is a tendency similar to that described for steady stress in that failure is precipitated by the maximum stress when the variation of stress is "moderate," and by a stress somewhat smaller than the maximum stress when the variation is "sharp." Many results are available with regard to the actual influence of stress concentration,[7] but it is difficult to establish definite general rules. In general, the factors of stress concentration obtained by the theory of elasticity are about double those obtained by fatigue tests on grooved specimens of ductile materials. It is important to remember, nevertheless, that σ_w in [5] represents a stress in excess of the mean stress.

The difference in behavior of the ductile materials under steady and variable stress is associated with the different mechanisms of failure. The failure under variable stress is closer to the brittle type of failure under steady stress, and the two phenomena do indeed display certain similarities. The case of combined stresses is accordingly somewhat complicated.

[6] Nadai, "Plasticity," p. 247.

[7] See paper in present symposium by Peterson on "Stress Concentration Phenomena in Fatigue of Metals."

The knowledge available so far indicates that the configuration of the failure may be interpreted in the light of the maximum-strength theory, but that the numerical value of the endurance limit in torsion comes close to one-half the endurance limit in bending; a relation that would point to the maximum-shear theory. Fig. 5 illustrates a typical failure under variable torsion stress, the 45-deg cracks pointing to failure in planes of maximum stress. It is worth noting that the steel in the shaft shown in Fig. 5 was found to be dirty, and it is conceivable that good material may show different types of failure, although failures of this kind are rare with satisfactory materials. The subject

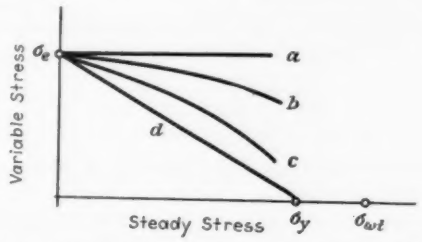
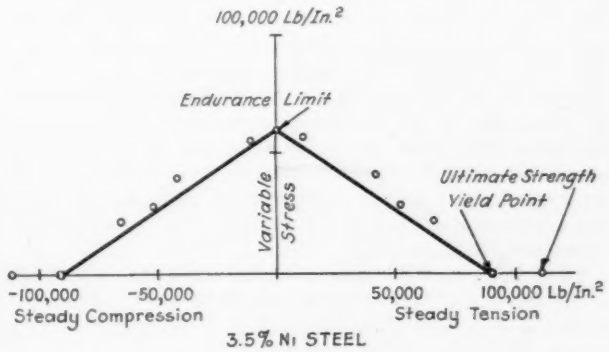


Fig. 6 Superposition of Steady and Variable Stress



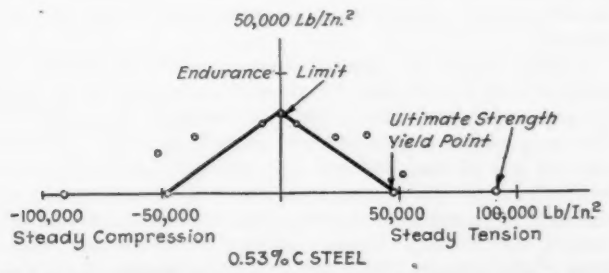


Fig. 7 Examples of Superposition of Steady and Variable Stress

is somewhat too controversial to permit definite conclusions being drawn. Until further information is available, it appears safe to predict failure on the basis of the maximum-shear theory. Whenever possible, an actual endurance test under the combined stress in question should be made.

Another important phase of the subject concerns the problem of the simultaneous existence of steady and variable stresses. This is a case of superposition not unlike that of tensile and shearing stress. Given a material with the yield point σ_y and endurance limit σ_e, subjected to a steady stress σ_0 and a variable stress σ_v, what is the probable combination for failure? The available results indicate the existence of many forms of superposition, from practical lack of interdependence to actual linear

superposition, Fig. 6. The reason for this absence of conclusive data may be ascribed to the fact that the ultimate strength is more closely related to the endurance limit than to the yield point. The linear form of superposition is particularly simple, however, and in general not far from the actual result in most materials, as illustrated by the two examples of Fig. 7.[8] Under this assumption, failure under the simultaneous application of σ_0 and σ_v takes place for the combinations $(\sigma_0/\sigma_y) + (\sigma_v/\sigma_s) = 1$, so that an appropriate rule for the working stresses becomes

$$\frac{\sigma_{0w}}{\sigma_y} + \frac{\sigma_{vw}}{\sigma_s} = u \quad [6]$$

For materials that exhibit the laws of superposition given by the curve b or c in Fig. 6, this law may be changed to

$$\sqrt{\left(\frac{\sigma_{0w}}{\sigma_y}\right)^2 + \left(\frac{\sigma_{vw}}{\sigma_s}\right)^2} = u \quad [6a]$$

but this modification may sometimes fall on the unsafe side of the experimental results.

The case of variable stress is important because of the remarkable influence of factors which do not commonly influence strength. The question of finish, ordinarily not of fundamental importance in strength problems, may assume real importance. Cases where unmachined specimens have shown 50 per cent lower strength than polished specimens of the same material have been recorded. This influence appears to be greater for alloy steels than for carbon steels.[9] The phenomenon known as "corrosion fatigue" covers cases where apparently slight intensities of corrosion affect the endurance limit profoundly, even to the point of raising doubts as to its existence.[10] There is not much doubt that this phenomenon is secondary to stress concentrations produced in the corrosion pits and possibly in intercrystalline cracks.

A less pronounced influence is caused by the surface condition of the specimens. The existence of plating has been found to affect the endurance limit profoundly,[11] possibly due to static stresses introduced by the plating.

These phenomena must be taken care of in individual cases. When more knowledge becomes available, the present concepts in the case of variable stress may have to be changed fundamentally.

Finally, there is another phenomenon which has seemed to warrant more investigation for the case of variable stress than for any other case. This is the phenomenon of "size effect."[12] The results of a fatigue test are affected appreciably by the physical size of the specimen. No adequate explanation has been offered for this phenomenon, but it is fortunately not of sufficient importance to warrant elaborate attention in connection with working stresses. In view of the peculiar conditions of the ordinary fatigue test, it must be considered extraordinary that the size effect is not much larger than the available results indicate. Here the test is made for a calculated bending stress which can only be inferred from the static loading of the specimen.

The case of variable stress represents perhaps the most important and most difficult case in connection with working stress. It is a case, moreover, where standardized rules of convention can do more harm than elsewhere. It is therefore necessary to study one's own special cases particularly closely in this field. One of the most important sources of information arises through the configuration of the cracks. The literature available on this subject is worthy of study.[13]

(*d*) Brittle Materials

The conditions of failure for brittle materials under steady stress are very similar to those for the ductile materials under variable stress, the ultimate strength σ_u being the limiting stress.

The effect of stress distribution is very similar to that for the ductile materials under variable stress, and the rules just discussed for ductile materials under variable stress may be used for brittle materials under steady stress.

The brittle materials frequently have fundamentally different properties for tension and compression; the ultimate strength in compression is often very much greater than for tension. This is particularly the case for cast iron and represents one of the most important reasons for its successful application to many important stress problems. It is necessary, therefore, to deal with two values for the ultimate strength: the tension value σ_{ut} and the compression value σ_{uc}, the latter frequently being four to five times as great as the former.

Under combined stress, the brittle materials may generally be treated by the maximum-stress theory, but in reality the failure problem is much more complicated.[14]

The behavior of the brittle materials under variable stress is a subject on which very little is known. It is to be expected that brittle materials will behave erratically under variable stress and be particularly sensitive to stress concentration. Results obtained for cast iron[15] indicate that this suspicion is not always justified. It must be regarded as good design practice, however, to avoid variable stresses of appreciable magnitude in the brittle materials.

DUCTILE MATERIALS AT ELEVATED TEMPERATURES

At elevated temperatures, the problem of failure assumes a double aspect. On the one hand, there remains the original problem of strength, modified and complicated by the many changes of internal structure of the materials that may occur. On the other hand, there appears the question of preserving the physical dimensions and initial stresses during a certain part of the life of the structure. The former aspect must be treated by the same fundamentals as those given in the preceding sections of this paper, modified for the many possibilities of unorthodox behavior of the materials. The latter aspect will be covered in greater detail in the following discussion.

The information upon which changes in the geometrical and mechanical aspects of a machine are predicted must be obtained from interpretations of long-time creep tests. The nomenclature surrounding the subject of plastic deformation of metals has been enriched rapidly during the past years, and it is now possible to discuss the subject in well-defined terms.[16]

As long as the conditions of service are similar to those obtaining in the long-time creep test, the result of the latter should

[8] For further examples see University of Illinois Bulletin No. 136.

[9] G. A. Hankins and M. L. Becker, "The Fatigue Resistance of Unmachined Forged Steel." Paper presented before the Iron and Steel Institute, Sept. 14, 1932. See also *Engineering*, Sept. 30, 1932, p. 402.

[10] H. J. Gough, "Corrosion Fatigue of Metals." Lecture presented before the Institute of Metals, London, Sept. 12, 1932.

[11] E. C. Lea, "The Strength of Materials as Affected by Discontinuities and Surface Conditions." *Engineering*, Aug. 26 and Sept. 2, 1932.

[12] R. E. Peterson, "Fatigue Tests of Small Specimens With Particular Reference to Size Effects." A.S.T.M. convention, Sept., 1930.

[13] F. Bacon, "Cracking and Fractures in Rotary Bending Tests." *Engineering*, Sept. 23, 1932, p. 372. See also paper in present symposium by R. E. Peterson.

[14] See paper in present symposium by Dr. Nadai on "Theories of Strength."

[15] University of Illinois Bulletin No. 164.

[16] A. Nadai, (1) "Plasticity," chap. 39; (2) "The Creep of Metals." National Applied Mechanics Meeting of the A.S.M.E., 1932. See also papers in present symposium by Robinson, McVetty, and McCullough.

be applicable. This is almost never the case, however, and it is gradually becoming recognized that the conventional high-temperature testing must be complemented by something more fundamental than the long-time creep test. Until this feeling has become translated into concrete results, there is no other choice but to make the best of the available results of this form of testing.

No result on the subject is more important than the law, originally formulated by Ludwik, that if the strain rate is increased by geometric increments, the corresponding stress is increased approximately by arithmetical increments. This law is usually written—

$$\sigma = \sigma' + \sigma'' \log_e \frac{v}{v_0} \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

where σ', σ'', and v_0 are constants of the material and $v = d\epsilon_p/dt$ is the time rate of increase of the plastic strain, or the strain rate. The law can claim only approximate validity within a limited range of strain rate because of the absence of accurate test results. The principal source of information for establishing this law has been the long-time creep test, but there are also available results obtained by somewhat different methods,[17] which give very satisfactory confirmation of the law.

When the results of long-time creep tests are analyzed in this manner, it is found that there is a different stress-strain-rate relation for each value of the plastic strain, the difference usually

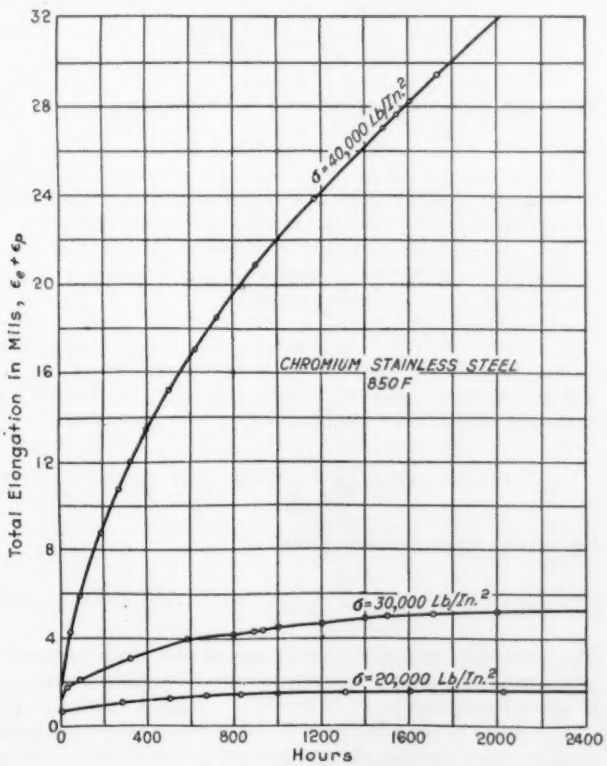


FIG. 8 CREEP CURVES

being a nearly constant addition to the stress for equal increments of the strain. This is due to the phenomenon of strain hardening. When the long-time creep test is continued into large plastic strains at the higher temperatures, an opposing annealing phenomenon sets in, so that the effects of strain hardening may be reversed. It has been customary to record experimental confirmations of Equation [7] for the minimum strain rate.

Notwithstanding the great efforts that have been applied to this problem, very little in the way of a quantitative solution is available. Practically no tests have been made with controlled strain rates, a method of testing that appears to have the greatest

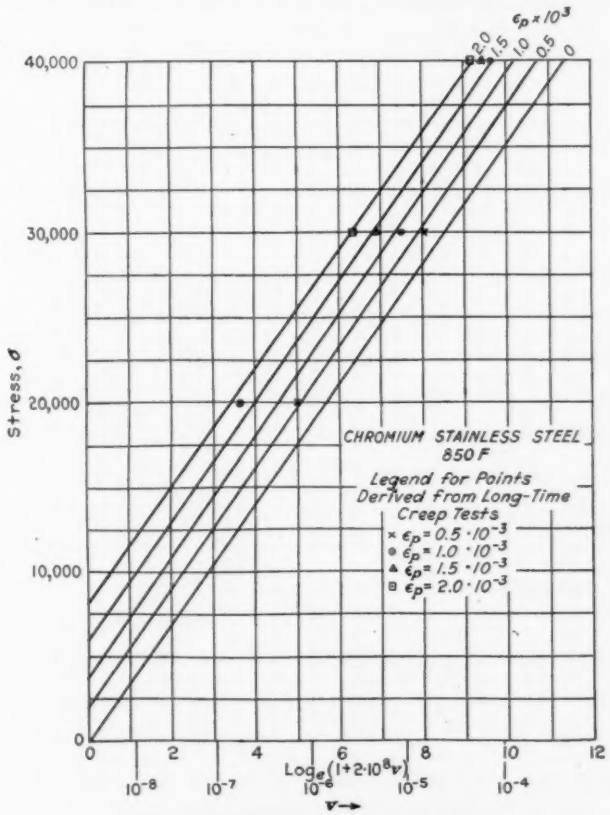


FIG. 9 STRESS-STRAIN-RATE RELATIONS

possibilities of all for short-time tests. It is particularly in the region of very low strain rates that our knowledge of the stress-strain-rate relation is deficient. A limit is soon reached in this direction on account of the limit of time and accuracy of measurements. For example, if the smallest strain that can be measured is 10^{-5}, it will require over ten years to obtain information about a strain rate of 10^{-10} 1/hr. The idea of such a test is nonsensical; laboratory experiments of this type cannot be extended beyond one or two years, and therefore we shall never escape the necessity of extrapolation.

The lack of data for the smaller creep rate suggested a long time ago the existence of an actual limit in stress below which a strain rate of zero can be supported. Succeeding investigations have dispelled this supposition so that the theory of a creep limit is now definitely abandoned. It must not be forgotten, however, that this is an academical argument that probably will never be fully settled.

Equation [7] does not pretend to cover very small strain rates, so that it must be modified by postulating some rational behavior in that range. A reasonable assumption is to make $\sigma = 0$ for $v = 0$. This condition can be satisfied by the law

$$\sigma = \sigma_1 \log_e \left(1 + \frac{v}{v_1}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

where σ_1 and v_1 are now the constants of the material.

[17] H. Deutler, *Physikalische Zeitschrift*, vol. 33 (1932), p. 347.

In order to make a simple correction for strain hardening, it is advisable to introduce the coefficient of work hardening,[18] $E_p = \partial\sigma/\partial\epsilon_p$. The experimental evidence does not indicate an actual constant E_p, but this will be assumed in the following; the error introduced by this simplification is not important in the light of the use that is made of the results. The influence of the plastic deformation upon the stress-strain-rate relation may thus be taken into account by the addition of the term $E_p\epsilon_p$; it then takes the form

$$\sigma = \sigma_1 \log_e \left(1 + \frac{1}{v_1}\frac{d\epsilon_p}{dt}\right) + E_p\epsilon_p \quad [9]$$

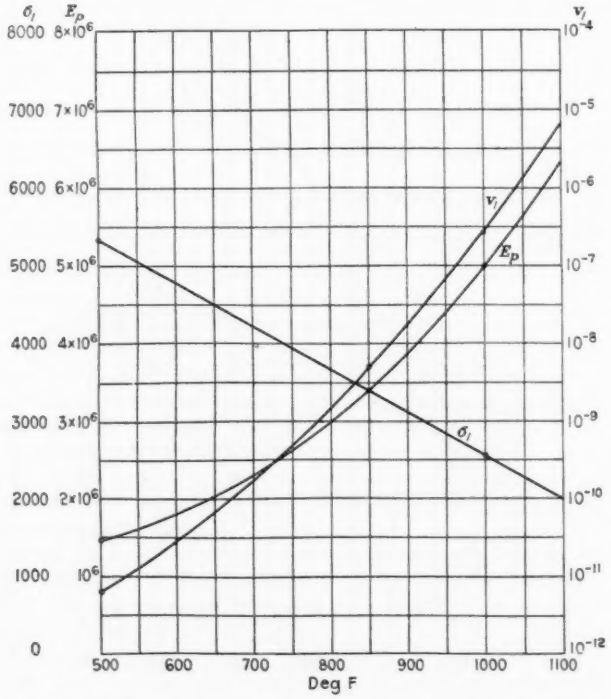


FIG. 10 MATERIAL CONSTANTS OF CHROMIUM STAINLESS STEEL

Fig. 8 shows a typical example of long-time creep curves, obtained from tests on chromium stainless steel at 850F. The points shown in Fig. 9 were obtained by analysis of the material shown in Fig. 8; the procedure consisting in plotting strain rate against strain for the different values of stress. The straight lines of Fig. 9 represent the best fit that can be made of the proposed function [9] and the most reliable of these points. The stress-strain-rate relation for zero strain is obtained by extrapolation and serves as the basis for determining the constants σ_1 and v_1. Fig. 10 gives values of these constants obtained for the same material as functions of the temperature.

The fit obtained in Fig. 9 between the function [9] and the test points is far from perfect, but it is not entirely unsatisfactory in view of the nature of the fundamental material of Fig. 8. It is fair to state, moreover, that the agreement obtained is generally better than that illustrated in Fig. 9.

The curves of Fig. 10 have an unusually simple form and one may be tempted to see some significance in this fact. The basic material is too fragmentary to lend much weight to such speculation, however. In the following discussion, more emphasis is given to the method of dealing with the fundamental experimental material than to the intrinsic nature of the latter.

[18] Nadai, "The Creep of Metals," A.S.M.E. New Haven Meeting, 1932. It may be suggested that this expression be modified into "modulus of strain hardening" to emphasize its relation to the modulus of elasticity.

In extrapolating such results to practical cases, it is instructive and generally sufficient to have access to a simple approximate solution of a relaxation problem of certain general aspects. Consider a tensile-test specimen A loaded in the manner shown in Fig. 11, so that the total elongation is constant. By varying the rigidity of the spring B from zero to infinity, all relaxation combinations of interest may be obtained. A solution of this problem for the stress-strain rate relation [9] is considered of sufficient interest to be included in this discussion.

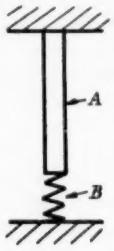


FIG. 11 RELAXATION MODEL

If the area and length of the specimen are denoted by a and l, respectively, the modulus of elasticity of the material by E, and the rigidity of the spring by k, it is possible, by introducing a modified modulus of elasticity

$$E' = E\frac{k}{k + E\frac{a}{l}}$$

to treat the problem in terms of the usual stress and strain relations. The effect of strain hardening is similarly influenced by the presence of the spring; this is taken care of by introducing a modified modulus of strain hardening,

$$E_p' = E_p\frac{k}{k + E_p\frac{a}{l}}$$

The stress-strain relation can now be written as

$$\sigma = \epsilon_e E' + \epsilon_p E_p' \quad [10]$$

where ϵ_e and ϵ_p are the elastic and plastic strains, respectively, also modified for the existence of the spring.

The essential feature about the relaxation phenomenon under discussion is that the total strain will remain constant. Thus, if the initial strain is denoted by ϵ_0,

$$\epsilon_0 = \epsilon_e + \epsilon_p \quad [11]$$

The initial stress may be written $\sigma_0 = \epsilon_0 E'$. Eliminating ϵ_e from [10] and [11] and introducing the stress parameter

$$\sigma^* = \epsilon_e E' = \frac{E'}{E' - E_p'}\sigma - \frac{E_p'}{E' - E_p'}\sigma_0 \quad [12]$$

the plastic strain may be written as

$$\epsilon_p = \frac{\sigma_0 - \sigma^*}{E'} \quad [13]$$

The seemingly arbitrary introduction of the stress parameter serves to eliminate the complications injected by the phenomenon of strain hardening.

The object of the analysis is to derive stress-time and strain-time curves from the stress-strain-rate relation [9]. This is equivalent to a solution of the differential equation

$$\frac{1}{v_1}\frac{d\epsilon_p}{dt} = e^{\frac{E'(\epsilon_0 - \epsilon_p)}{\sigma_1}} - 1$$

for the boundary conditions $t = 0$, $\epsilon_p = 0$, $\epsilon_0 = \sigma_0/E'$. Introducing the time parameter

$$t^* = \frac{1}{v_1}\frac{\sigma_1}{E'} \quad [14]$$

the stress-time relation is found to be

$$t = t^* \log_e \frac{1 - e^{-\sigma_0/\sigma_1}}{1 - e^{-\sigma^*/\sigma_1}} \quad \ldots\ldots\ldots\ldots [15]$$

The corresponding strain-time relation is obtained by introducing the relation [13] between ϵ_p and σ^*. Fig. 12 shows a graph of Equation [15] for values of t between 0 and 0.08 t^*. Fig. 13 gives a more extensive plot on a logarithmic time scale.

When the quantities σ_0/σ_1 and σ^*/σ_1 are large, Equation [15] can be written with sufficient accuracy in the simple form

$$t = t^*[e^{-\sigma^*/\sigma_1} - e^{-\sigma_0/\sigma_1}] \quad \ldots\ldots [16]$$

These results make it possible to predict the time required for the stress to drop from an initial to a final value for an arrangement of the type shown in Fig. 11, provided that the constants σ_1, v_1, and E_p' are known. The influence of strain hardening is usually not of fundamental importance, but it serves to improve the material for each successive reestablishing of the initial stress. The foregoing results require only slight modification to apply to the subsequent relaxations as well.

If the first relaxation proceeds from the initial stress σ_{0a} to a final stress σ_a, there has taken place a permanent plastic strain $(\sigma_{0a} - \sigma_a)/E'$. This strain must be added to the initial strain produced by the assembly stress in the restored state. If the latter is made σ_{0b}, the time required for the stress to drop to the value σ is found to be

$$t = t^* \log_e \frac{1 - e^{-\sigma_{0b}/\sigma_1}}{1 - e^{-\sigma^{**}/\sigma_1}} \quad \ldots\ldots\ldots\ldots [17]$$

where σ^{**} is a new stress parameter determined by the relation

$$\sigma^* = \frac{E'}{E' - E_p'}\sigma - \frac{E_p'}{E' - E_p'}(\sigma_{0b} + \sigma_{0a} - \sigma_a) \ldots [18]$$

and t^* retains its value as determined by [14].

It is evident that σ^{**} differs from σ^* only by the substitution of $\sigma_{0b} + \sigma_{0a} - \sigma_a$ for σ_{0b}. This suggests an obvious rule for extending the results to further periods of relaxation.

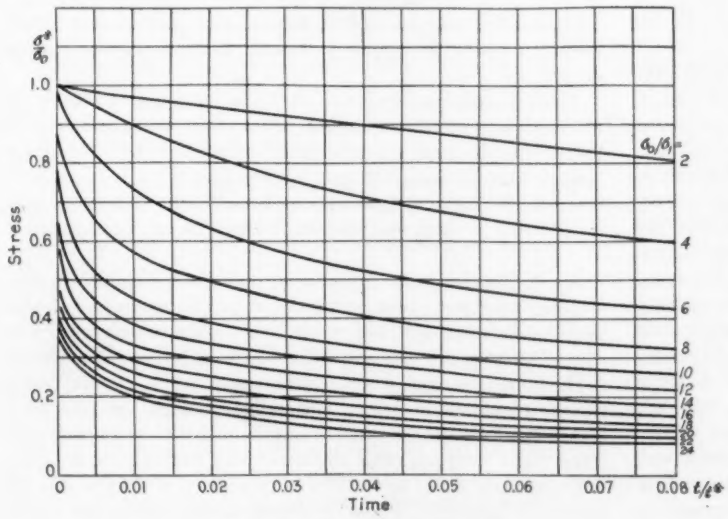


FIG. 12 RELAXATION CURVES

In order to illustrate the significance of the above results, they will be applied to one of the most common applications of materials at high temperatures, namely, a flange bolt. Let the material be chromium stainless steel at 850F, as covered by Fig. 10. The following values are obtained for this material: $\sigma_1 = 3450$ lb per sq in., $v_1 = 5 \times 10^{-9}$ 1/hr, $E' = 25 \times 10^6$ lb per sq in., $E_p' = 3.4\ 10^6$ lb per sq in., and $t^* = 27{,}200$ hr.

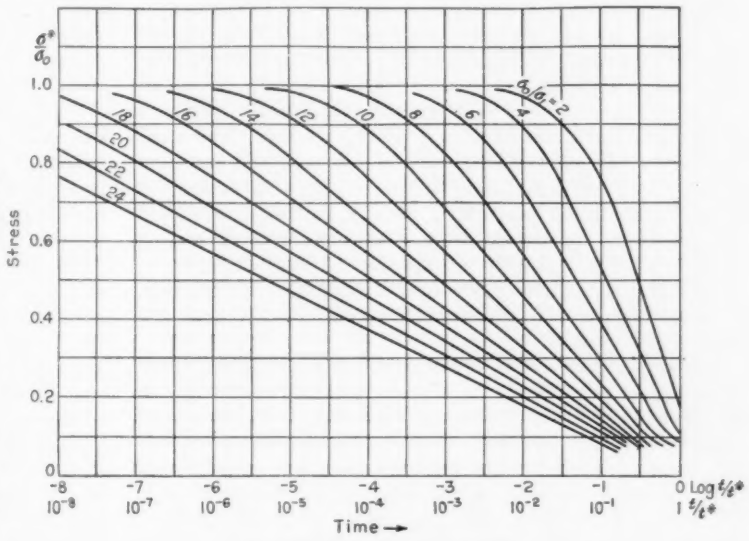


FIG. 13 RELAXATION CURVES

Let the bolt be given an initial stress, permitted to relax to a lower stress, retightened to the same initial stress, and again permitted to relax. Table 1 gives the time of relaxation for both of these periods for several values of the initial and final stress and for two values of the modulus of strain hardening. As a matter of interest, Table 2 gives corresponding values of the relaxation time for the same material under similar conditions, but at 750 F.

If conditions permitted an initial stress of 30,000 lb per sq in. and the retightening of the bolts every 4000 hr, it is evident from these results that the joint must be designed to operate with a stress of about 10,000 lb per sq in. at 850 F, and about 19,200 lb per sq in. at 750 F. The sensitivity of the relaxation phenomena to operating temperatures is worthy of note.

The influence of strain hardening is somewhat overestimated in the above results, since the lower value of E_p is consistent with the test points only for a different set of values for v_1 and σ_1. Carrying out a solution under this new premise, the final result will not be found to differ very materially from the above.

In accordance with Table 1, the stress falls from 30,000 lb per sq in. to 10,000 lb per sq in. in 4000 hr. The strain rate for this material at 30,000 lb per sq in. and zero plastic strain is about 26×10^{-6} 1/hr. If this remained constant, the stress

TABLE 1 RELAXATION OF CHROMIUM STAINLESS STEEL AT 850 F

Initial stress, lb per sq in.	Final stress, lb per sq in.	Time of relaxation, hours: First tightening $E_p' = 3.4 \times 10^6$	First tightening $E_p' = 1.7 \times 10^6$	Second tightening $E_p' = 3.4 \times 10^6$	Second tightening $E_p' = 1.7 \times 10^6$
30,000	15,000	690	484	1,405	585
30,000	10,000	4,020	2,400	11,300	3,780
20,000	15,000	435	408	558	435
20,000	10,000	2,470	1,895	4,020	2,370

TABLE 2 RELAXATION OF CHROMIUM STAINLESS STEEL AT 750 F

Initial stress, lb per sq in.	Final stress, lb per sq in.	Time of relaxation, hours: First tightening $E_p' = 2.6 \times 10^6$	First tightening $E_p' = 1.3 \times 10^6$	Second tightening $E_p' = 2.6 \times 10^6$	Second tightening $E_p' = 1.3 \times 10^6$
30,000	15,000	13,700	10,650	20,900	14,500
30,000	10,000	47,100	42,300	105,100	54,800
20,000	15,000	9,890	9,110	11,400	9,890
20,000	10,000	40,400	37,200	57,100	42,300

would drop to 10,000 lb per sq in. in about 30 hr, or in less than 1 per cent of the actually predicted time. These figures serve to illustrate the importance of considering the relaxation phenomenon correctly.

This example will serve to illustrate the fundamental reasoning to be followed in establishing working stresses. The same method may be used, with greater or less accuracy, for other details. In more complicated structures, such as parts of a turbine cylinder, for example, it may be difficult to establish accurate values for the elasticity of the structure (k), and an accurate analysis is usually not possible. The most important aspects of the analysis cover overall deformations, however, so that it is possible to deal with average stresses and thus simplify the analysis.

Very little can be said as yet about conditions under combined stress or variable stress. It appears probable that creep in plastic metals occurs in all planes and not only, as is often stated, in planes of maximum shear. Until further knowledge is available, however, it does not appear unreasonable to predict creep under combined stresses on the basis of the maximum-shear theory. Irregularities in the distribution of stress are very effectively obliterated at elevated temperatures and can always be neglected, provided that the corresponding external deformations are not objectionable. The case of variable or alternating stress should be treated with extreme caution until more knowledge is available.

The question of margin between actual and predicted performance is somewhat different from the case of failure at normal temperatures. When deformation or relaxation is accompanied by inevitable disaster, it goes without saying that the utilization of the properties of the materials must be fully as cautious as in all other applications. Cases where creep is accompanied by disaster are very rare, however. It may be stated that the fear of creep has in many cases been permitted to dominate designs too extensively. What is frequently defined as safe and comfortable conservatism in this respect may bring far-reaching consequences of a pernicious nature. In permitting the physical dimensions of details such as bolt diameters, wall thicknesses, etc., to swell out to unreasonable proportions, the designer invites other and just as dangerous forms of operating troubles, such as thermal distortions, unreliable materials, and so on. Here, as in many other activities of the engineer, narrow and dogmatic adherence to prematurely established concepts is infinitely more dangerous than practical, intuitive judgment.

There is one aspect of the application of materials that must be treated with extreme conservatism, however, before operating temperatures are increased much beyond the present range. This is the inevitable change of microscopic structure which will become more and more important as the temperatures approach the recrystallization temperatures. Carbide spheroidization[19] in carbon steels is perhaps the most important of these aspects, particularly since it produces a profound change in the plastic properties. The general problem of stability of the cohesion is of utmost importance, because the failures associated with this problem are particularly subtle.

[19] R. W. Bailey and A. M. Roberts, "Testing of Materials for Service in High-Temperature Steam Plant." Proc. of Inst. M.E., vol. 122 (1932), p. 209.

Discussion

A. V. DE FOREST.[20] One interesting point brought out in the paper is the examination of failures which occur in service. If we could accurately analyze the conditions of failure we could then learn amazingly rapidly not to repeat the same mistake. Unfortunately the first reaction to a failure on the part of the engineer is to make the part either heavier or of stronger material without stopping to inquire as to whether failure was caused in the first place by a lack of either of these qualities. A typical example is the well-known habit of changing material when the failure is due to design, and vice versa.

Among the more complicated problems in this field, the author has well brought up the general question of brittle and ductile behavior. All too many metallurgists do not recognize that mere static ductility is frequently of no importance in machine parts, as failure usually occurs before deformation has exceeded the strain available in the most brittle materials.

An even more important point is also raised, and that is the necessity of knowing the working and accidental stresses which are applied to any structure. In an attempt to partially solve this problem, the writer has devised a recording strain gage, sufficiently light and sufficiently inexpensive to apply to many problems involving dynamic stress due to impact loading, fatigue conditions, and stresses due to resonant vibration. This system of measurement consists in a direct record scratched by a diamond on a metal or glass target without a mechanical amplification. The resulting record is measured or photographed under a suitable magnification. In this way a recorder can be built which weighs but a fraction of an ounce, and the only expensive part is the magnification unit which may be used on a large number of individual instruments. This device is commercially made by Baldwin Southwark of Philadelphia, and has been successfully applied to such complicated stresses as occur during airplane maneuvers.

The penultimate paragraph of the paper contains a most important piece of advice, namely, that intuitive judgment is far better than analysis based on undigested concepts. If only designers would intuitively put large fillets at each junction of two straight lines on their drawing board, many failures ascribed to material, or otherwise successful design, would be obviated.

G. B. KARELITZ.[21] Apparently, when we talk of working stresses, we always keep in mind the main stresses in a loaded member, introducing such corrections as are necessary to take care of stress concentration. The writer wishes to call attention to the numerous cases where failure occurs locally, causing nevertheless considerable grief. Generally speaking, in case of a highly localized failure, one has to deal with a three-dimensional-stress state. All problems connected with such a state are quite difficult, mainly due to scarcity of experimental data on the behavior of materials under localized three-dimensional stress. As pointed out in the paper, the present method of treatment consists in extrapolating information obtained from unidirectional simple tests to three-dimensional cases.

That this method is unreliable can be easily demonstrated on, say, ball bearings. The stresses obtained by the application of the Herz formula are of the order of 200,000 to 300,000 lb per sq in., or even higher. The shearing stresses obtained from consideration of the difference between the principal stresses is of the order of 80,000 lb per sq in. The number of stress cycles in the material runs into millions, yet failure does not occur. It would be worth while to undertake an investigation of working

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[21] Westinghouse Elec. & Mfg. Co., Lester, Philadelphia, Pa. Mem. A.S.M.E.

stresses in the case of "surface loading," where a highly localized stress exists under the surface of the metal at the point of application of a concentrated load.

The failure of gear teeth due to pitting is another instance of wide interest. The present theory of pitting explains it by the effect of shearing stresses under the surface of the tooth material, as caused by the contact pressure between the teeth. Citing as an example the case of high-speed reduction gearing, the Navy specifications allow a load of 95 lb per inch face per inch diameter of pinion. This is equivalent to a shearing stress of approximately 11,000 lb per sq in. in the tooth material. This tooth load is considered to be perfectly safe, yet it must be admitted that the stress is fairly high when compared with the fatigue limit for low-carbon steel. There are a number of high-speed reduction gears working successfully at a shearing stress of the order of 18,000 lb per sq in. Apparently, the general criterion of safe working stress is hardly applicable here, and a special treatment is necessary. It would be very profitable and timely to institute a program of research to elucidate the question of allowable working stress for such cases as ball and roller bearings, gear teeth, and similar machine details.

KENNETH G. MERRIAM.[22] It is to be regretted that the very logical notion of "utilization factor," as proposed by the author, has not been in use in airplane design. To become familiar with the present state of affairs in airplane-design procedure, the reader is urged to study the following definitions which are taken from the 1932 issue of "Airworthiness Requirements of Air Commerce Regulations for Aircraft," published by the Aeronautics Branch of the U. S. Dept. of Commerce:

1 *Basic Load.* The basic loads on (or stresses in) an airplane or part when the airplane is at rest or in a condition of unaccelerated flight.

2 *Design Load.* The load for which a member is designed. It is usually obtained by multiplying a basic load by a specified load factor.

3 *Ultimate Load.* The load that caused failure in a test or the load that, according to computations, should cause failure in a member.

4 *Load Factor.* The ratio of some other load to the basic load in which the relative distribution of forces is the same. This other load may be any of the following: The load applied during some special maneuver, the maximum probable load on the airplane or part, the design load, or the ultimate load.

5 *Margin of Safety.* The ratio of the ultimate strength of a member minus the design load to the design load. Usually a member with a positive margin of safety is satisfactory, and one with a negative margin of safety is unsatisfactory.

6 *Factor of Safety.* The ratio of the design load or the ultimate load to the maximum probable load. In airplane design the least factor of safety is usually about 2.

To the writer it would seem desirable to redefine items (2) and (4), and to have item (5) supplanted by the notion of utilization factor, as defined by the author. Realizing that this is a matter requiring very careful study, the writer proposes the following imperfect definitions as a basis for further discussion by airplane designers and others concerned:

2 *Design Load.* The maximum probable load on a member for the condition under investigation, obtained by multiplying the basic load by a specified load factor which is defined below.

4 *Load Factor.* The ratio of the maximum probable load on the member for the condition under investigation to the basic load.

5 *Utilization Factor.* The ratio of the design load to the ultimate load. In airplane design the maximum utilization factor is usually about 0.50.

So far as the writer can see, these new definitions would cause airplane and machine designers to speak a common language when discussing working stresses.

BENJAMIN MILLER.[23] The title of Fig. 1 should be "Definition of Yield Strength for 0.2 Per Cent Set," and the definition of yield point should be "The stress in a material at which there occurs a marked increase in strain without an increase in stress." The paper would then be in line with the A.S.T.M. Tentative Standard E 8-32 T.

In the fifth paragraph following Equation [1], the author states that under favorable circumstances it should be possible to utilize 75 to 80 per cent of the strength of the material. He says further on that although the Mises-Hencky theory is more correct than the maximum-shear theory, the maximum-shear theory is sufficiently accurate for practical purposes. He then points out that the difference between the theories is (for pure shear) the difference between 0.577 and 0.5.

There seems to be a contradiction here. For if a material fails in tension at a stress value of 1000, and it is desired to use the material in shear with utilization of 80 per cent, by the Mises-Hencky theory the working stress can be 462 units, but can only be 400 units by the maximum-shear theory. Therefore if the maximum-shear theory is used in the practical case the factor of utilization will be not 80 per cent but less than 70 per cent, and there will be a 15 per cent waste of material. On the other hand, the shear test may be easier to make than the tension test. Suppose in such a shear test a strength of 577 units is found. Then, if the maximum-shear theory is used, the tensile working stress for 80 per cent utilization will be calculated to be 923 units, so that the factor of utilization on the Mises-Hencky theory would be 92.3 per cent, which might leave an insufficient margin of safety.

Until the law of yielding is established with an uncertainty of much less than 15 per cent, it seems idle to talk of factors of utilization of 75 to 80 per cent.

A. NADAI.[24] With regard to the mathematical form to be suggested to the designer for defining the condition of yielding, it seems to the writer that one of the various reasons which might be brought up in favor of the acceptance of the expression

$$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 = 2\sigma_0^2$$

(σ_0 yield stress for pure tension, σ_1, σ_2, σ_3 principal stresses) is that not only do the results of a larger number of more recently

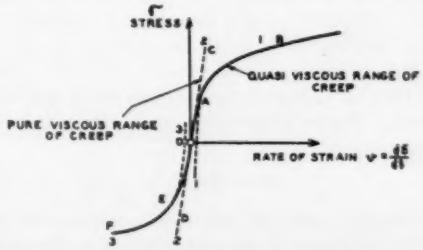


FIG. 14 CONDITIONS OF A VELOCITY-STRESS CURVE AT ORIGIN AT A GIVEN ϵ = CONST.

and carefully made tests under combined stress seem to settle the question in its favor but also mechanical reasons lend further support to this condition and certain important general symmetry properties inherent therein or in the mathematical consequences to which it may lead when further applied to plastic equilibria

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[23] Henry L. Doherty & Co., New York, N. Y.

[24] Westinghouse Research Laboratories, East Pittsburgh, Pa. Mem. A.S.M.E.

or plastic states of stress in bodies. It is not possible to develop these reasons here in detail; suffice it to say that Equation [1] is to be preferred to the expression $\tau_{max} = \text{const}$ (τ_{max} being the maximum shearing stress for the stress distribution σ_1, σ_2, σ_3), both from a mathematical as well as from a physical point of view.

Most timely and valuable contributions to the treatment of actual problems of turbine designers are the author's brief remarks on strength at elevated temperatures and the permanent deformations of metals under prolonged loading. The rational design of certain heavy parts in steam turbines and of all kinds of heavy apparatus subjected to high pressures and elevated temperatures seems to depend to a greater or less extent on two major problems of the creep of metals, namely, the laws of plastic flow under constant load or constant stress and on those of the relaxation from stress or of the processes under which a given stress distribution may gradually disappear and stress will readjust itself with time. Despite a few valuable attempts which have recently been made by mathematicians and by engineers to analyze such cases on an exact basis and although a considerable amount of empirical information has been already gathered on the creep of metals for design purposes, it cannot be said that the theory of this particular kind of plastic deformation has kept pace with the present needs of industry. To these circumstances may perhaps be attributed the surprising fact that a closer agreement in the interpretation of results of the simplest tensile tests among the various investigators has not yet been reached. From this point of view a discussion of some of the assumptions made or of the mathematical expressions suggested may be in order and of practical interest. Only one point will be considered here.

For example, the first attempts by Ludwik and by Cassebaum to express from tensile tests how the yield stress σ at normal temperatures depends on the rate of strain $v = \partial\epsilon/\partial t$ pointed to a logarithmic expression or law such as that given in Equation [7] of the paper. While this law holds for *rapid* speeds of deformation, it evidently fails for small rates of strain converging to zero. The recent tests by Deutler have verified and extended the law from the rates of strain of an ordinary tensile test to the extremely high rates encountered in impact tests (speed ratios: 1 to 1 million) for a few ductile metals tested at normal temperature. The parameters σ', σ'', and v_0 depend on the unit strain ϵ with the understanding that the stresses σ producing the various rates of stretching $v = \partial\epsilon/\partial t$ are to be compared for a given and the same plastic elongation $\epsilon = \text{const}$.

As far as a possible extension of this relation to quite small rates of flow or strain is sought at normal temperatures, certain conditions of uniformity have to be satisfied by the function connecting the stresses σ and velocities v. It must, for example, be assumed that for compression the same speed law holds as for tensile stresses. Hence satisfactory expression of this law in the form of a function

$$\sigma = f(v) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [19]$$

for small values of v, including zero speeds, will have to be an odd function of the variable v in the neighborhood of the point $v = 0$. The simplest function satisfying at this point the condition of uniformity is

$$\sigma = \phi v \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [20]$$

with ϕ as a constant, which is the expression of pure viscous flow. This is one of the reasons which led the writer long ago to assume tacitly that if a speed law like [19] at a constant $\epsilon = \text{const}$. exists in the neighborhood of zero rates of strain, the curve expressing the speed function must have an inflection point and may be approximated for small rates of flow by the straight line [20]. Although the material available from tensile creep tests made with metals at elevated temperatures is not yet conclusive enough to justify acceptance of a law similar to the one (Equation [7]) found to express the facts adequately at much higher rates of strain and at normal temperature, it has been independently suggested in slightly altered form by a few investigators as a possible expression of the creep law of metals at high temperatures and under higher speeds of stretching. Under the expectation that this law is probably valid at elevated temperatures in a limited range of the variables (plastic strain ϵ, plastic rate of strain $v = \partial\epsilon/\partial t$ and stress σ) in the following, and until more conducive information may become available, a slightly modified working hypothesis for predicting creep deformations is proposed. If for constant strains $\epsilon = \text{const}$. in a certain range AB (Fig. 14) of the positive velocities v the logarithmic relation of Equation [7] is found to express the speed law in a satisfactory manner, then the true speed-stress curve may perhaps resemble the curve shown in Fig. 14. This curve is drawn to pass through the origin O, ($v = 0$) where σ follows first a steeply inclined straight line such as OC. At a certain stress it will leave this straight line and will bend and rapidly approach a logarithmic curve such as that shown in Fig. 14 in the branch of a curve AB. The corresponding must be true on the negative side of the speed axis (i.e., for compression). Thus the curve *FEOAB* consisting of portions of two identical logarithmic curves (1) and (3) symmetrically situated with respect to the origin O (*but not passing through this latter point*) and of a straight line (CD) might perhaps represent the speed laws, at least in a certain limited area of the ϵ,v-plane. Whether it is perhaps sufficient to draw the straight line *COD* directly as the common tangent to the two logarithmic curves (1) and (3), or whether the straight line has to be assumed more steeply inclined than the common tangent line, as indicated in Fig. 14, must, on account of lack of proofs and information, be left an open question.

R. E. Peterson.[25] It has occurred to the writer that in connection with certain applications the need for "high-strength" steels has been overemphasized and the additional cost involved perhaps not justified. For instance, in the case of a shaft having a groove and subjected to alternating stress, the strength in terms of different steels would be such as shown in Fig. 1 of the writer's paper in this symposium. Thus, in this particular case, steels with ultimate strengths of 90,000 and 180,000 lb per sq in. (ordinary endurance limits approximately 45,000 and 90,000 lb) actually show but relatively little difference in fatigue of grooved specimens (approx. 30,000 and 37,000 lb respectively). Furthermore, the higher-strength steel would perhaps be brittle, i.e., of lower impact value, so that an occasional overload would be more serious. Consequently, the choice of materials for the case of variable stress together with stress concentration must be carefully considered and not merely based on ultimate strengths or ordinary endurance limits.

The paper states that "it must be regarded as good design practise to avoid variable stresses of appreciable magnitude in brittle materials." Certain heat-treated alloy steels are quite brittle (approaching 5 per cent elongation). Dr. Gillett[26] states: "Despite some individual opinion to the contrary, there is no acceptable evidence that a high static elastic ratio or high ductility, such as may be obtained by suitable alloying and heat treatment, at all improves the endurance ratio. In an endurance test of a small specimen a strong, brittle steel acts just like an equally strong, ductile steel." It would seem that the reason for avoiding brittle materials should not be so much on account of the presence

[25] Westinghouse Research Laboratories, East Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

[26] H. W. Gillett, "Effect of Alloying and Heat Treatment on the Endurance Limit of Steel." Proc. A.S.T.M. (1930), part I, p. 291.

of variable stress but more because of the possibility of impact. In ordinary machine parts, impacts due to accidental overloading or other unforeseen causes may result in fractures or local cracking if the material is brittle.

Heat-treated helical springs having high Brinell hardness and quite low ductility are widely used for variable-stress application. Two reasons may explain why this is possible: absence of pronounced stress concentration, and reduction of impact effect due to large deflection with resulting increased time interval.

With respect to "size effect" in fatigue, the paper states that "it is fortunately not of sufficient importance to warrant elaborate attention in connection with working stresses." This seems to be the case in the reference noted which concerns specimens without stress-concentration effect. However, for geometrically similar specimens with holes or fillets a decided size effect seems to exist in fatigue tests,[27] i.e., endurance strength obtained with large specimens may be only half that obtained with specimens of ordinary size. Much work remains to be done on this question, and while it is too early to make definite recommendations, yet it would also seem that the question should not be dismissed as being of insufficient importance.

In discussing the effect of unmachined surface on fatigue strength, mention is made of cases where 50 per cent lower strength is obtained as compared with polished specimens. It might also be added that tests of leaf springs[28] have shown still greater reductions, as much as 79 per cent. This tremendous decrease is generally attributed to surface decarbonization.

RONALD B. SMITH.[29] The latter part of the paper is of special concern to those who must apply materials to engineering structures at high temperatures. Admitting the controversy existing over the behavior of materials at elevated temperature, the author has approached the problem as an engineer, with a solution that, while not exact, will cope with ordinary design requirements.

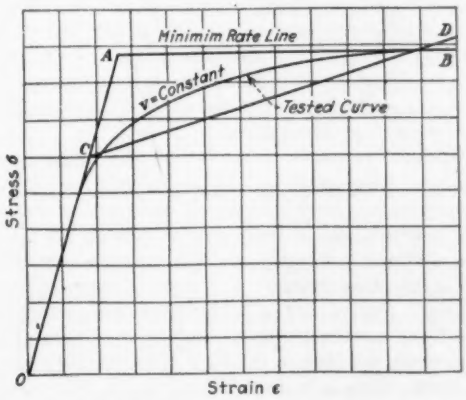


FIG. 15 HIGH-TEMPERATURE STRESS-STRAIN DIAGRAM

Tests at elevated temperature lead to a σ-ϵ diagram, at constant velocity, as shown in Fig. 15. Plotting the usual conception of the Ludwik equation, based on minimum creep rates, is tantamount to constructing the minimum-rate line *OAB;* however by applying the rate equation as a function of ϵ_p, the author approximates the stress curve by *OCD*, where *C* is determined logarithmically.

[27] R. E. Peterson, "Model Testing as Applied to Strength of Materials," A.S.M.E. Bigwin Meeting, June, 1932.

[28] R. G. C. Batson and J. Bradley, "Fatigue Strength of Carbon and Alloy-Steel Plates as Used for Laminated Steel Springs." Proc. Inst. M. E., vol. 120 (1931), p. 301.

[29] Turbine Engineering Dept., Westinghouse Electric & Mfg. Co., South Philadelphia Works. Jun. A.S.M.E.

With the temperatures in vogue in central-station design this is the better approach to the relaxation problem. That the test data are in finer agreement with this analysis than one would gather from Fig. 9 of the paper, is evident from Fig. 16, which is a replot from Ernest L. Robinson's paper "Materials at High Temperature." The tests were made on cast manganese steel at 842 F. As creep continues the stress for all values of plastic

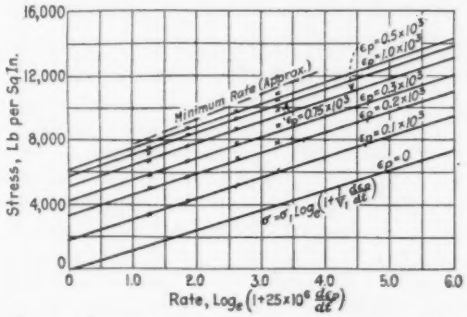


FIG. 16 STRESS-STRAIN-RATE RELATIONS FOR CAST MANGANESE STEEL AT 842 F

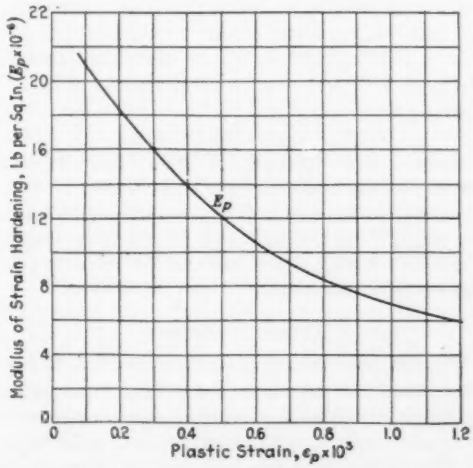


FIG. 17 STRAIN-HARDENING MODULUS FOR CAST MANGANESE STEEL AT 842 F

strain approaches the constant-rate line so prominent as a logarithmic relation in high-temperature literature heretofore.

The evidence from considerable test data indicates that, like Fig. 16, the additional strength resulting from strain hardening does not follow a constant modulus of strain hardening. It is probably some exponential function similar to Fig. 17. Equation 9 can be readily modified for this variation.

$$\sigma = \sigma_1 \log_e \left(1 + \frac{1}{v_1}\frac{d\epsilon_p}{dt}\right) + f(\epsilon_p) \ldots\ldots\ldots [9a]$$

Since $$\sigma^* = \sigma - f(\epsilon_p) = \sigma_0 - E'\epsilon_p$$

the function is easily integrated and Equation [15] becomes

$$t = t^* \log_e \frac{1 - e^{-\sigma_0/\sigma_1}}{1 - e^{-[\sigma - f(\epsilon_p)]/\sigma_1}} \ldots\ldots\ldots [15a]$$

where $$\epsilon_p = \frac{\sigma_0 - \sigma + f(\epsilon_p)}{E'}$$

That the exponential relation is more rational than the assumption of a constant modulus is shown in Fig. 18 where the creep curve for a stress of 8000 lb per sq in. has been calculated by a

graphical integration of [9a] and compared with the test data. If a constant mean value of the modulus is selected the agreement is poor.

Notwithstanding the author's admission that the test results are too meager to justify analysis of Fig. 10, it may not be amiss

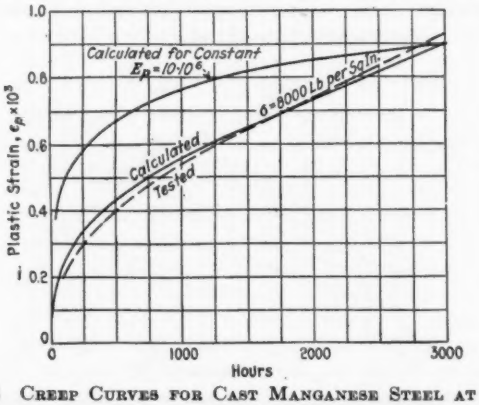


FIG. 18 CREEP CURVES FOR CAST MANGANESE STEEL AT 842 F

to mention the general trend that one would anticipate. Numerous investigators[30] have found that strain hardening disappears at high temperatures. This would conform to a negative slope in the E_p-function, and in case it be represented exponentially, to a flattening of the curve and a reduction in the quantitative value.

Another property of crystalline materials at elevated temperatures is viscosity. Roughly, it is to be expected that stress and rate may be represented by[31] $\sigma = \phi\ (d\epsilon_p/dt)$.

The viscosity ϕ decreases with temperature, and the relation between σ_1 and v_1 should be such that in satisfying Equation [9a] for a constant velocity, they result in an increase in stress as the temperature is reduced.

AUTHOR'S CLOSURE

The importance of recording strain gages, referred to by Dr. De Forest, cannot be overestimated. They have played a very important role in correctly evaluating obscure circumstances of loading in such structures as ships, bridges, railroad rails, and so on. The device mentioned should be of great value on account of its small size, permitting its application to conditions where the usual types of strain gage are not applicable.

The complicated cases of three-dimensional loadings, referred to by Mr. Karelitz, belong to a section of the problem that is particularly difficult. We have to admit that our knowledge of what goes on just under the surface of a ball race or a gear tooth is limited. Another example of this class of phenomenon is the problem of erosion of turbine blading due to impingement of water drops at high speed. It would seem that all of these phenomena have to be set aside for a treatment of an entirely different order than that advanced by this paper. The ambiguous property of materials defined as hardness is probably closest to the actual circumstances for these cases.

The confused terminology in the subject of strength is to be deplored. While the confusion must be eliminated by careful definition of existing terms, rather than by the injection of entirely new terms, it has seemed worth while to the author to continue the use of the term "utilization factor." Wherever working stresses must be defined with precision, the term appears to justify its existence. The modifications proposed by Mr. Merriam in the terms used by the Aeronautics Branch of the U. S. Department of Commerce, injecting this term, would thus appear worth while. It is particularly in connection with superposition of several stresses that the use of the term leads to pronounced simplification.[32]

The proposal of Mr. Miller to change the name yield point to yield stress, to bring the paper into agreement with the A.S.T.M. standards, is a good suggestion. The apparent contradiction in the statement referring to the practicability of using 75 to 80 per cent of the strength of the material may be attributed to the lack of definition of what constitutes "favorable circumstances." This is a subject of great importance that belongs to every discussion of working stress, but which was considered of somewhat too local a nature to be included in the present paper. In applying materials to engineering structures, the possibility of predicting failure accurately varies over a large range. At the one end of this range there is the case of applying ductile materials in the form of a tensile test piece. At the other end there is the case of complicated forms of stress applied to materials of doubtful properties. In the former case the results of the tensile test may be applied directly without the intermediate interpretation of a strength theory. No case can be more favorable than this, and under such circumstances it is not unreasonable to utilize 75 to 80 per cent of the known strength.

Dr. Nadai's comments on the paper are very valuable. It is undoubtedly true that something would be gained by building up rules for working stresses around the von Mises relation for plastic flow. There is a question, however, if the added implication would be justified in view of the general uncertainty of the practical aspects of the subject.

In suggesting the creep law $\sigma = \sigma_1 \log_e (1 + v/v_1)$, the tacit assumption is made that when v is very small the law becomes $\sigma = \frac{\sigma_1}{v_1} v$, where the quantity σ_1/v_1 has the dimension of viscosity and is identical with ϕ. This subject is of practical importance, since very small creep rates cannot be explored experimentally and still play a real rôle in machines intended to operate continuously for decades. It is entirely possible that we shall have to be content with theoretical reasoning in tracing the creep relations for very low creep rates.

Mr. Peterson's reference to the satisfactory fatigue properties of heat-treated alloy steels with low ductility is very interesting. The elongation is a poor measure of ductility for this type of material, however, and it may be that a more correct picture would be obtained if ductility could be measured quantitatively in a more satisfactory manner. It is true, on the other hand, that the designer's fear of brittle materials is associated with the behavior of the materials under unforeseen conditions of loading. If an overstress, whether in the form of an impact or not, may cause a crack, there is a very decided reason for conservatism in the use of the material.

The author wishes to express his appreciation for the work of Mr. Smith in investigating the justification for a constant value of the modulus of strain hardening. It is evident from Fig. 18 that very superior agreement between test and prediction is obtained by an assumption of a variable value for this quantity than that which is obtained by using an average constant value. The difference is not great for the typical case of relaxation; for the case of constant stress, there are usually available actual creep data. While it is thus probable that the use of a constant modulus of strain hardening is accurate enough for practical purposes, the complication of using a variable modulus is not too great to make it objectionable, particularly when the relaxation data are plotted in curves of the type shown in Fig. 13.

[30] For rather extensive tests see M. Polàni and E. Schmid, *Naturwissenschaften*, vol. 17 (1929), p. 301.

[31] A. Nadai, "Creep of Metals," A.S.M.E. New Haven Meeting, 1932.

[32] See author's paper, "Factor of Safety and Working Stress." Trans. A.S.M.E., paper APM-52-2, p. 13.

APM-55-17

Metals at High Temperature—Test Procedure and Analysis of Test Data

By ERNEST L. ROBINSON,[1] SCHENECTADY, N. Y.

Tests on the flow of metals at high temperatures under stresses so small that the resulting distortions are less than one-tenth of 1 per cent show, throughout the duration of tests extending into the second year, continued reductions in rates of flow so that, for such small distortions, it is impossible to associate any particular rate of flow with a corresponding allowable stress. A statement of test results, to be of use, requires in addition to rate of flow and applied stress, certainly a record of total extension and possibly also a record of time elapsed. Furthermore, tests should be made under conditions of stress approximating the conditions of application in actual service.



THE construction of machinery for operation at high temperature requires information on which to base the selection of suitable materials. During the period before actual experience has been had at the higher temperatures this choice must be based on tests of new materials to see if they have properties as good relative to the more severe conditions as the older materials had under the former conditions.

There is one essential consideration, the relative importance of which may shift completely. At low temperatures materials are chosen largely with reference to strength. Allowable safety considerations dictate stresses so small that distortions (which are almost wholly elastic) are of the order of magnitude of 1 mil per inch or less, and are of minor consequence in comparison with strength considerations.

On the other hand, at high temperatures there are certain applications in which plastic distortions of intolerable size would occur long before there could be any danger of failure due to stress. Under these conditions the whole basis of design necessarily shifts from consideration of allowable stress to consideration of allowable distortion, and only such stresses as are accompanied by tolerable distortions can be permitted.

Under such conditions, testing should bear a strict relation to conditions of application.

During the past several years the General Electric Company has been conducting tests on the suitability of materials for use at high temperature.[2]

While there has been substantial agreement with other laboratories as to the flow-rate strength of certain standard materials, there are many cases where the values given by other laboratories appear considerably too high to correspond with the conditions of use represented by the General Electric tests.

It would be of decided advantage to all concerned if test results could be recorded in more complete form in order that corrections to corresponding conditions of application might be possible.

Conditions Insuring Greatest Usefulness of Test Results

It is the purpose of this paper to suggest several considerations that ought to be taken account of in test in order for the results to be of greatest usefulness.

The most difficult element is the time element. High-stress tests completed in a week or two have been found to reverse the relative desirability of materials as compared with tests of longer duration at stresses or extensions which might be tolerable in design. Certainly a one-year test is in no sense a ten-year application, and similarly no one-day test can replace a full-year test until both have been made and the correspondence established. The General Electric Company selected as quickly as possible on the basis of short-time tests of six to eight weeks' duration the most promising materials, and already has tests on the best of these materials proceeding into the second year.

Tests ought to be conducted under conditions approximating the conditions of service. Thus, if a material is to be subjected to steady stress which will not be subject to relief as time goes on, the material should be so tested. This requires the use of several samples at different loads, to define or "bracket" an allowable stress or extension.

Similarly, if a material is to be used for a bolt or assembled with a fit subject to relief as flow occurs, the test should be made in that manner or at least conducted with steps of successively reduced loading approximating a condition of constant elastic plus plastic extension.

A few General Electric tests have been made accidentally with successively increased loadings. These came about because the material proved better than originally thought. Such tests are difficult to interpret, and do not appear to have any corresponding application.

Formula for Correction Purposes

In order to present the importance of testing in the neighborhood of service conditions, a formula is offered which is roughly suitable for correcting from one condition to another. This formula is wholly inadequate for more than correction purposes. In fact, there would be no moral to this story if the formula were good over a long range of values. It represents approximately plottings which prove to be nearly straight lines on log-log paper

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[1] Turbine Engineering Department, General Electric Co. Mem. A.S.M.E. Mr. Robinson was born at Canton, N. Y., on April 11, 1890. He was graduated from the St. Lawrence University in 1911 and from the Harvard Graduate School of Applied Science in 1914 (M.C.E.). For three years he was engaged in construction work and the design of steel and reinforced-concrete structures in New York and in water-power engineering in New England. During the war he served in the Oise-Aisne offensive as a 1st Lieutenant with the 302nd Engineers, U.S.A., and later as Captain and Adjutant of the 2nd Engineer Training Regiment. For the past thirteen years he has been employed by the General Electric Company in its Turbine Engineering Department.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] "Effect of Temperature on Materials Required in Turbine Design," by S. H. Weaver, *G.E. Review*, November, 1930, p. 654. "Flow of Steels at Elevated Temperature," by F. P. Coffin and T. H. Swisher, Trans. A.S.M.E., June 15, 1932, APM-54-6, p. 59.

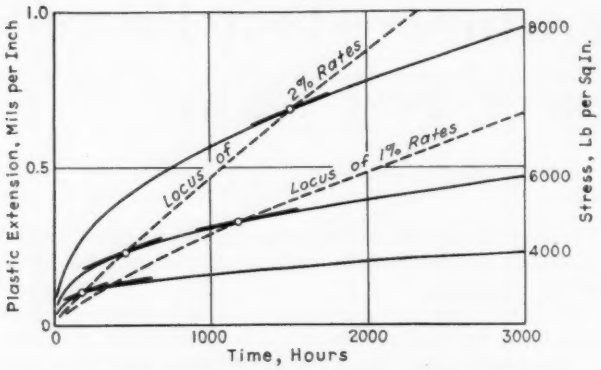


FIG. 1 BEHAVIOR OF TEST BARS UNDER STEADY STRESS
(Material, cast manganese steel; temperature, 842 F.)

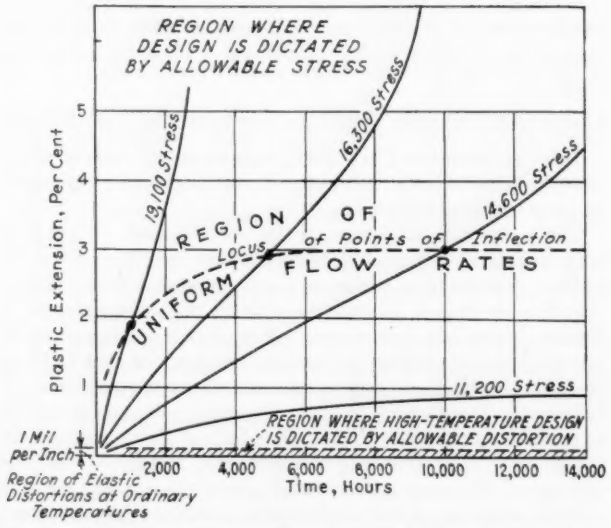


FIG. 2 REGIONS IN WHICH DESIGN IS DICTATED BY DIFFERENT CONSIDERATIONS
(Material, cast manganese steel; temperature, 842 F.)

over a range of one or two squares. Thus a satisfactory correction may be made to a tenfold extent, using a constant exponent, but not over a hundredfold range without using a different exponent.

Let S = applied stress
T = temperature
r = rate of flow
x = total plastic extension

Then

$$\frac{S}{S_0} = \left(\frac{T_0}{T}\right)^p \left(\frac{r}{r_0}\right)^{1/n} \left(\frac{x}{x_0}\right)^{1/k}$$

Expressed in words, the indicated relations are as follows:

1 For a particular previous extension and a specified rate of flow, the strength decreases as the temperature increases. Thus, in the case analyzed below, an increase in temperature of 50 F shows a reduction of one-third in the creep strength. The exponent p is approximately 7 over the range from 700 F to 1000 F.

Where stresses are fixed, and after some definite amount of previous extension, an increase of temperature of 50 F increases the rate of flow some tenfold (more or less). Thus a year's normal distortion may easily be caused by 5 weeks' operation of a machine at 50 F excess temperature.

2 For a particular temperature and a specified previous extension, a 50 per cent increase of stress causes a tenfold increase in the rate of flow. Thus the rate of flow may be reduced to one-tenth its former value by a reduction of stress of approximately one-third. The relation is approximately good for rates ranging from 0.1 per cent per 100,000 hr to 10 per cent per 100,000 hr. n is approximately 6 for the case illustrated hereafter.

3 For a particular temperature and a specified rate of flow, a tenfold accumulation of plastic extension enables the material to carry approximately double the stress at the specified rate. The relation is approximately good for extensions ranging from 0.1 mil per inch to 10 mils per inch, that is, from one-hundredth of 1 per cent to 1 per cent. k is approximately 3.5 for the case illustrated below.

This last characteristic is probably to a certain extent an indication of "strain hardening," the tests all being in the range where strain hardening is apparently incomplete.

THE TESTS CONDUCTED

General Electric tests for the most part cover total plastic extensions of approximately 1 mil per inch, that is, one-tenth of 1 per cent and rarely exceeding two-tenths of 1 per cent.

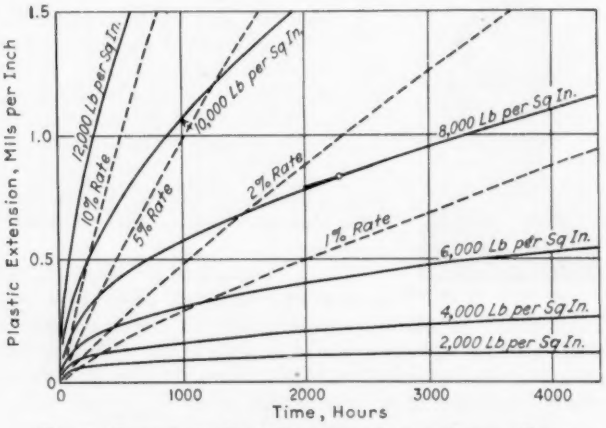


FIG. 3 RECORD OF TEST RESULTS—STEADY-STRESS TESTS
(Material, cast manganese steel; temperature, 842 F.)

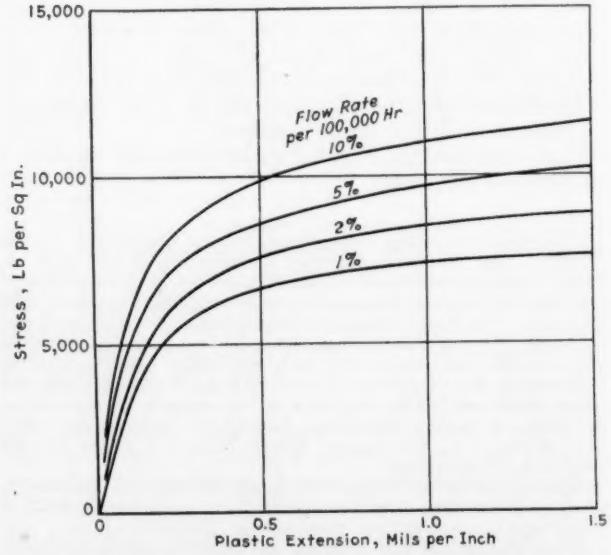


FIG. 4 HIGH-TEMPERATURE FLOW-RATE CHART
(Material, cast manganese steel; temperature, 842 F.)

Throughout this limited range, the materials tried improve in strength with stretching, the flow rate decreasing as the plastic extension increases.

Fig. 1[3] illustrates these relations. Thus with a 6000-lb stress a rate of 2 per cent per 100,000 hr is attained with an extension of 0.23 mil per inch. However, the same test bar after being stretched to 0.33 mil per inch showed a rate of only 1 per cent with the same load. Similarly a bar with 8000 lb stress allowed to stretch 0.68 mil per inch will carry its load at a 2 per cent rate, while a bar stretched only 0.1 mil per inch can carry only 4000 lb load at that rate. In no case has a rate been found steady with total extensions of less than 10 mils per inch or 1 per cent. Thus a test which yields a line, straight within the accuracy of the test data for a period of a month, a year later on will yield another line, equally straight for a month, but at a lesser rate.

Although the tests under consideration were confined to small extensions, it seems necessary to recognize that a material stretched to such amounts as, for instance, 10 per cent or 20 per cent, approaching the limits of ductility, would have to show increasing slopes instead of the decreasing rates described by the formula. In between there is a range of stretch at a steady rate such as that illustrated by Fig. 2.

Many very useful tests have been conducted with reference to this region, which doubtless have utility for applications in which total extensions of several per cent are allowable.

Returning to the region of small distortions, Fig. 3 shows complete curves for 1, 2, 5, and 10 per cent rates per 100,000 hr. Fig. 4 shows the basic information obtained from the series of

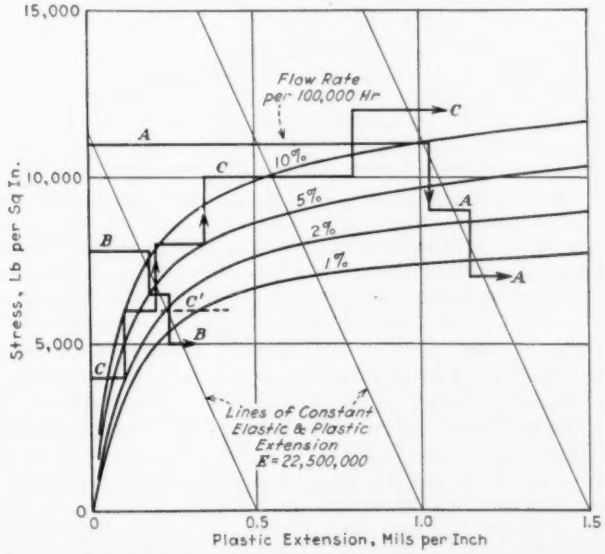


FIG. 5 HYPOTHETICAL FLOW-RATE TEST PATHS UNDER VARIOUS CONDITIONS

(Material, cast manganese steel; temperature, 842 F.)

test bars represented in Fig. 3. The lines of constant stress in Fig. 3 are simply horizontal lines in Fig. 4, crossing the lines of successively smaller rates.

Fig. 5 has been drawn in order to show the significance of the manner of conducting a test. The slant lines represent the conditions of bolts with several degrees of initial tightness, the elastic extension being relieved gradually as the plastic flow occurs.

A and *B* represent hypothetical test paths of single bars with stresses stepped down under conditions approximating constant extension, the results being plotted with the usual log-log graph in Fig. 6. Bar *A* has an initial stress of 11,000 while bar *B* has an initial stress of 7800. The test conducted at the higher stress level would be expected to show a 1 per cent flow-rate strength of

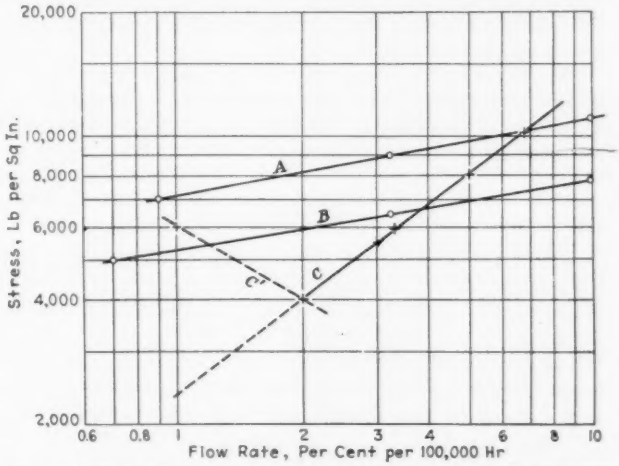


FIG. 6 LOG-LOG PLOT OF FLOW-RATE RESULTS UNDER CONDITIONS SHOWN IN FIG. 5

(Material, cast manganese steel; temperature, 842 F.)

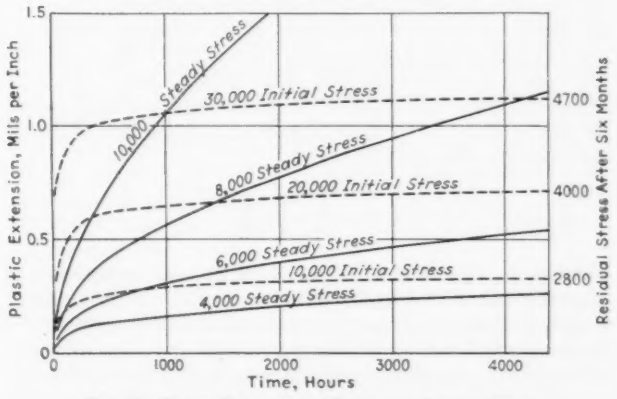


FIG. 7 FLOW WITH TWO TYPES OF APPLICATION

(Material, cast manganese steel; temperature, 842 F
All stresses in pounds per square inch.)

7200, while the lower-level test would show only 5500. In each case the rate is 1 per cent per 100,000 hr for the cited stress.

Path *C* illustrates the difficulty of drawing conclusions from a test in which the initial stress applied was too low. Subsequent increases of stress are likely to lead to unsystematic interpretations. Thus a little more patience at the 6000 stress would show a 1 per cent rate along the broken-line path *C'*, and actually reverse the slope of the log-log plot in Fig. 6.

The type of diagram presented in Fig. 4 gives the relations between three quantities which must be associated in order to define the flow-rate strength of a material at a certain temperature. Doubtless the elapsed time at a high temperature accomplishes chemical effects which may be of importance also. Thus a diagram of rates plotted on a stress-extension base like Fig. 4 can be drawn from a series of step-down tests at several extensions covering only a minor fraction of the time consumed in making the tests illustrated in Fig. 3. But as mentioned before, the definite relation between long- and short-time tests has still to be established. In the meantime it seems preferable to construct the diagram from tests made in imitation of the conditions of application.

[3] This and the succeeding diagrams are based on the material illustrated in Figs. 18 and 19 of the paper by Messrs. Coffin and Swisher mentioned in footnote 2.

Fig. 7 illustrates the two common types of application already referred to for which test information is sought. The solid lines show extension with a steady stress such as occurs in shells or casings with steady internal pressure. The broken lines show the plastic extension which occurs when the initial elastic extension is relieved by the plastic flow as in bolts or shrink fits. In these cases the residual stresses at the end of a six-month period are shown at the end of each curve.

From what has been shown it should be clear that a table of stresses for a nominal rate and a certain temperature needs to be supplemented by a record of extension, and preferably a complete extension-time diagram of the test.

Discussion

A. NADAI.[4] The numerous data on the flow of metals at high temperature which have been published during the last few years in this country, particularly the valuable results of tests to which the present paper makes reference, have recently stimulated further attempts to correlate the considerable information which has become available with time and to express it with the final aim to have it in a more exact or more analytical form available for the purposes of the design of high-temperature equipment.

The demand to express creep data by formulas which would be valid within a limited range of the variables which they contain should be supported. One natural suggestion which might be made in this respect leads to assume that during a tensile test the stress σ (taken per unit of area of the actual cross-sectional area A) might be approximated by a product of a function of the unit strains ϵ and of a function of the rates of strain $v = \partial\epsilon/\partial t$. It may be furthermore assumed that both of these functions would be power functions of their respective independent variables, thus assuming that the stress

$$\sigma = \sigma_0 \left(\frac{\epsilon}{\epsilon_0}\right)^n \left(\frac{v}{v_0}\right)^m \quad \ldots\ldots\ldots\ldots [1]$$

where σ_0, ϵ_0, v_0, n, and m are constants at a given temperature. This is practically the same expression, rewritten here by using a slightly changed notation and assuming that the temperature T is constant, which is the base of a number of conclusions contained in the paper. The author makes it clear that he does not want to have this function used or suggested for other than correction purposes, and that if it may be used, it may be good to represent a limited portion in the field of the variables ϵ or v.

It may perhaps be of some interest for the description of the creep phenomena of metals to try to draw a few simple conclusions, which would be the consequences of the product and the power function law as it was expressed by Equation [1]. It is implicitly assumed when introducing this equation that the elastic part of the strains or of the rates of strains may be of secondary order or be neglected, so that no claims are made to express, for example, any details referring to the elasticity of the bars. Assuming that the stretching in a tensile test at elevated temperature is observed while the load s (this shall be taken for the sake of convenience per unit of area of the *original* cross-sectional area A_0 with which the test was started) is kept constant, on account of the relations:

$$(1 + \epsilon)A = A_0 \quad \ldots\ldots\ldots\ldots [2]$$

$$sA_0 = \sigma A \quad \ldots\ldots\ldots\ldots [3]$$

which must hold during the test, it would follow that the load s would be given by the expression:

$$s = \sigma \frac{A}{A_0} = \frac{\sigma}{1 + \epsilon} = \frac{\sigma_0}{1 + \epsilon} \left(\frac{\epsilon}{\epsilon_0}\right)^n \left(\frac{v}{v_0}\right)^m \quad \ldots\ldots [4]$$

Hence in a tensile test made under a constant load s = const., the speed of strain v corresponding to this load at any strain ϵ is given by solving the equation:

$$\left(\frac{v}{v_0}\right)^m = \frac{s}{\sigma_0}(1 + \epsilon)\left(\frac{\epsilon}{\epsilon_0}\right)^{-n} \quad \ldots\ldots\ldots [5]$$

for the variable v.

Considering in this last expression for the variable v, s as a constant, the strain ϵ_i at which the minimum creep rate $v_{min.}$ will be observed might now be easily found by putting $dv/d\epsilon = 0$ and solving for ϵ. This furnishes an extremely simple result for the strain at which the minimum creep rate is reached; namely, the value

$$\epsilon_i = \frac{n}{1 - n} = \text{const.} \quad \ldots\ldots\ldots\ldots [6]$$

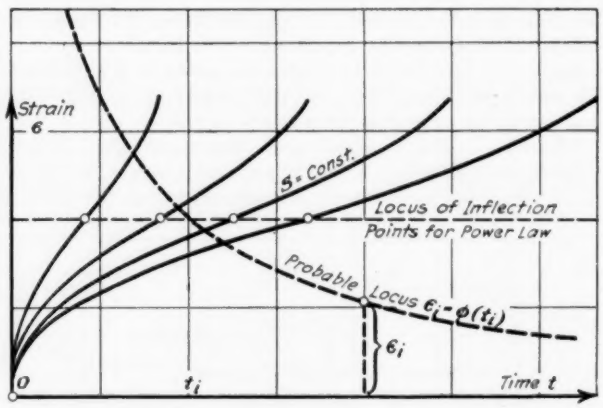


FIG. 8 TIME-ELONGATION CURVES FOR CONSTANT LOADS

Equation [6] indicates that this value depends *only* on the exponent n. The minimum creep rates under various loads s = const. will therefore all be reached at the same unit elongation $\epsilon_i = n/1 - n$, and the creep curves $\epsilon = f(t)$ for various constant loads s = const. will have their inflection points I on the same horizontal line as indicated in the sketch Fig. 8. Thus the locus of the inflection points is a horizontal line. The shape of the time-elongation curves $\epsilon = f(t)$ might also be found by solving [5] for

$$v = \frac{d\epsilon}{dt} = v_0 \left[\frac{s(1 + \epsilon)}{\sigma_0}\left(\frac{\epsilon}{\epsilon_0}\right)^{-n}\right]^{-m} \quad \ldots\ldots [7]$$

and through a subsequent graphical integration of this last equation.

As far as the writer could check the interesting property expressed by [6] just derived from the power function law, Equation [1] is not verified in actual creep-elongation curves.

In the range of loads where these curves in tests have shown inflection points, the locus of these latter in the ϵ, t plot was found to be a curve which is first steeply inclined at the small values of t (at high loads) and which tends to approach rapidly the t = axis for decreasing loads.

It might finally be mentioned that the assuming of a power law as expressed by Equation [1] has also the other consequence that all the creep curves in Fig. 8 for various loads s = const. might be constructed from one of them by multiplying the abscissas (t) of one curve by a constant factor.

The result found by creep tests, that the inflection points hav-

[4] Research Laboratories, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

ing the coordinates ϵ_i, t_i (t_i being the time at which the minimum creep rate $v_{min.}$ is reached, ϵ_i the corresponding strain) of the various time-elongation curves $\epsilon = f(t)$ taken for constant load s = const., seem to lie on a curve $\epsilon_i = \phi(t_i)$, the ordinates ϵ_i of which decrease with increasing t_i, seems not to support the result derived from Equation [1] that all the creep or time-elongation curves $\epsilon = f(t)$ for given loads s = const. could be obtained from each other by stretching their abscissas t by constant amounts in the direction of the time axis.

S. R. PUFFER.[5] The author has shown very clearly the evil effects of slight increase in temperature and stress or in a combination of the two, when a long life with slight deformation at elevated temperatures is required. To insure such a long life requires an accurate knowledge, not only of the materials themselves, but of the maximum conditions of stress and temperature to which they will be subjected.

There are other applications with high stresses and temperatures where a life of 1000 or 2000 hours, or even less, is acceptable, but where the conditions of temperature and stress, or even the rate of plastic deformation, are either unknown or exceedingly hard to determine.

Such conditions are encountered in general in the internal-combustion-engine field, and more particularly in the supercharging of aircraft engines by means of exhaust-gas turbines. Temperatures, instead of being in the neighborhood of 850 F, may range between 1200 F and 1600 F, subject to change without notice. Stresses which are carefully figured for an agreed maximum condition are always uncertain, because these maximum conditions are continually being exceeded, either in an attempt to break some record, by the pilot's maneuvers, increased output of the engine, or other similar reason.

Under such conditions it has been necessary to base the selection of materials and the design features on test bars pulled at the usual testing-machine rate, but at temperatures which cover the operating range. The values obtained are of course not expected to be absolute values for the material, as at these temperatures the effect of creep is very decided. By making such tests on a number of materials, results can be obtained which, although not absolute, are comparable. The only possible choice is to select the material which gives the highest comparative figures, and then start to look for something better.

P. G. MCVETTY.[6] The author's further analysis of some of the test results reported by Messrs. Coffin and Swisher indicates the trend toward interpolation and extrapolation of creep-test data in order to make them more suitable for design use. The curves of Fig. 2 show graphically the extrapolation possible by use of the empirical formula proposed by Mr. Levine and based on four creep tests at 6000, 9000, 12,000, and 15,000 lb per sq in. extended over a total time of 1500 hr. In the published data, the maximum plastic extension was somewhat less than 0.7 per cent, so that the original test curves superimposed upon Fig. 2 would fall within an area less than 1 per cent of that shown. This is mentioned merely to emphasize the extent of extrapolation necessary to bridge the gap between the usual creep-test data and actual service conditions.

The author has shown a locus of points of inflection on the creep curves. In any extrapolation of creep data the approximate location of such inflection points is a matter of vital importance because of the danger of an increasing creep rate within the life of the material in service. From Fig. 2 we might conclude that a plastic extension or creep of at least 3 per cent at any stress less than 14,600 lb per sq in. would be necessary before the creep rate would begin to increase. This appears to give us assurance with an ample margin of safety that inflection points on the creep curves need not be considered in the region where high-temperature design is dictated by allowable distortion. Before accepting this assurance, it seems desirable to review the evidence which supports it.

It is well known that inflections do occur, but they did not appear in the curves upon which Mr. Levine's formula was based. This empirical formula, involving stress, total plastic extension, and time, fits the observed data very well up to 1500 hr, but its use to predict the locus of inflection points cannot be recommended without further experimental evidence.

Little is known about the laws governing the change from a decreasing to an increasing creep rate. It is usually assumed that the plastic extension associated with the early stages of the creep test produces a strain-hardening effect which increases the creep resistance, thereby decreasing the creep rate. During the test the annealing effect of time and temperature is cumulative, leading to softening of the strain-hardened material and an increasing creep rate. When these two opposing forces are balanced we have what appears to be a constant creep rate. At low stresses the change of creep rate is so gradual that it is very difficult to locate the inflection point.

We do have some data on the laws governing the softening of strain-hardened material. It is well known, for example, that the time at a given temperature to produce softening decreases as the amount of plastic extension increases. From this we would expect that an increase in the time at a given temperature would cause a decrease in the plastic extension necessary to start softening. The application of this reasoning to Fig. 2 indicates that the locus of inflection points should start at a high value of stress and plastic extension in a short time and occur at gradually decreasing values of plastic extension as time goes on. Tests at higher stresses appear to support this viewpoint rather than the one shown in Fig. 2.

If we adopt this more conservative prediction of the locus of inflection points, we must admit the possibility of such inflections in the region of low values of stress and plastic extension after very long time. In other words, the creep curve and the locus of inflection points approach each other with increasing time. We are not yet able to say whether or not they meet within the service life of the material, but the possibility of passing the inflection point under service conditions must be kept in mind in predicting the total creep.

The author leaves some uncertainty as to the source of data given in Fig. 3. If these curves represent actual creep tests at six stresses over times up to 4500 hr, they furnish a more reliable basis for the subsequent analysis than that resulting from the application of the Levine formula to the four tests previously mentioned. Another series of tests of this kind would offer a valuable check on the validity of this formula in the region of lower stresses and longer times.

The author has considered an important problem upon which we have very little experimental data. This is the case of constant extension while the initial stress is reduced by the action of creep. This "relaxation" problem is complicated by the flexibility of adjacent members in addition to the assumed change from elastic to plastic deformation in the bolts themselves. Perhaps it should be emphasized that in this problem we are not so much concerned with the instantaneous value of flow rate for a given combination of stress and extension as we are in the stress change. Following a horizontal line in Fig. 3 (constant plastic extension) shows that stress falls rapidly at first and more and more slowly as time goes on. In the actual case, the flexibility

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[6] Research Laboratories, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

and creep of adjacent members cause relaxation to proceed along some line other than that of constant plastic extension or constant sum of elastic and plastic extension. The importance of a quantitative solution of this problem indicates the desirability of further experimental work in this field. For such an investigation particular attention must be paid to the first stage of creep, for it is in this that maximum stress changes may be expected.

The author has made a valuable contribution to our knowledge of the analysis of test data for practical application. He has shown the importance of conducting tests under conditions approximating those of service. He has also shown the importance of keeping in mind the limitations of the original data in any subsequent applications. Our future progress in high-temperature design depends on our skill in combining the knowledge of the metallurgist, the designer, and the testing engineer to establish a reliable basis for choice of materials and working stresses.

Author's Closure

Dr. Nadai points out some very interesting facts about the formula proposed by the author for correction purposes. In making his extrapolation he is kind enough to acknowledge the limitation stated by the author. Adequate comment is made difficult by the lack of numerical scales for strain and time in Fig. 8, contributed by Dr. Nadai. However, Dr. Nadai's Equation [6] gives, for the material under discussion, a locus of inflection points at a strain of 40 per cent, which gives some gage as to the scale of his diagram. Now the author's formula was for limited use in the region covered by Fig. 3. Fig. 3 is contained in the hatched area at the bottom of Fig. 2 and in a region less than half the width of the black boundary line at the bottom of Fig. 8. As stated in the paper, the formula is wholly inadequate for the implied extrapolation.

Mr. McVetty points out the extent of the extrapolation in Fig. 2 of the previously published data. First, it is necessary to bear in mind that Fig. 2 was extended into a region of little interest purely for the purpose of defining how limited is the cross-hatched area of importance. In defense of the diagram it should be noted that subsequent to the publication of the Coffin and Swisher paper the tests were continued up to 5440 hr and 1.84 per cent extension, and at the end of that time the curves were still following Mr. Levine's formula based on the first 1000 hr of test. As a probable estimate of inflection points, the author preferred Mr. Levine's formula.

Dr. Nadai's idea as to the probable shape of the locus of inflection points with very large strains may be entirely correct, and Mr. McVetty's apprehensions as to inflection points with small strains may be warranted at much higher temperatures, but there are no inflection points in Fig. 3, which was prepared by cross-plotting from the actual Coffin and Swisher tests, extending over a longer time and to far higher strains than shown.

In answer to Major Bull's inquiry at the meeting, the composition of the material as given by Coffin and Swisher is repeated here for convenience: C, 0.31; Mn, 1.40; Si, 0.36. Heat treatment: 900 C, 6 hr, furnace cool; 850 C, 4 hr, furnace cool; 700 C, 4 hr, furnace cool.

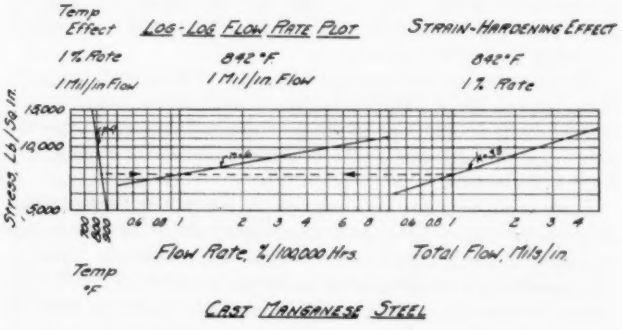


Fig. 9

In conclusion, Fig. 9 represents the author's correction formula and graphically portrays the triple dependence of flow rate on applied stress, temperature, and accumulated creep. In the middle is the usual log-log plot. At the left is given the temperature effect and at the right the strain-hardening effect. The use of such a diagram in preference to the formula will guard against unwarranted extrapolation. When test results are so plotted, deviations from the straight lines present to the eye the limitations of the data. And the diagram serves to bring out the different factors that must be recorded in order to describe the high-temperature performance of a material.

APM-55-18

The New Theory of Plasticity, Strain Hardening, and Creep, and the Testing of the Inelastic Behavior of Metals

By H. HENCKY,[1] CAMBRIDGE, MASS.

The knowledge of the inelastic behavior of metals has experienced considerable growth in the last few years. To draw all the advantages possible from the experiments, frequently very difficult, an effort has been made to bring the entire development under some dominating physical ideas. These ideas are in fact very old, and deal mainly with the proper conception of the hidden elastic energy that is responsible for the statical component of the strain hardening. The analytical treatment of the inelastic behavior gives promise of being valuable not only in the testing of materials, but even for the designer of machines used in the forming of metals.



RECENT research work considering the problem of creep of materials at high temperatures has developed a secure basis on which the further experimental and theoretical study may proceed.[2] However, in the region of small strain velocities the experiment yields only erratic results and a theoretical extrapolation has to be made.[3] The aim of this paper is not to develop new ideas about plasticity, creep, and strain hardening, but to form a coherent picture and to extend the scope of the basic laws verified by experiment in such a manner that future experiments can be made under the guidance of a theory flexible enough to represent all phenomena of practical importance.

I—The Physical Facts Underlying the Behavior of Metals

It has been repeatedly emphasized that the postulates of the mathematical theories concerning the behavior of materials are not fulfilled. The engineer occupied with the testing of materials is especially likely to underestimate the value of theoretical research, which is not to the advantage of the increase of our knowledge. The reason for such prejudice is a misunderstanding of the meaning of mathematical theories.

The value of a mathematical theory is not that it gives a complete description of the behavior of any individual material, but that it furnishes a mental background against which the true behavior of a specimen can be distinguished and measured.

Theories are simply mental extensions of our physical instruments and have nothing to do with matter in itself. Every theory is developed by the mind, and therefore a theory can have practical value, but cannot be true in any absolute sense.

However, leading scientists and engineers have already recognized the valuable suggestions given by applied mechanics and mathematical physics and have tried to extend the classical theory of elasticity in order to reduce our knowledge of materials to physical constants which can be measured and defined free from individual arbitrariness.[4]

The mathematical theory must be based on the physical facts common to all plastic metals. The following facts have to be given consideration:

(1) All metals are polycrystalline substances. Stress and strain are defined only for such elementary particles as contain thousands of crystal grains. The true stresses at the grain boundaries we call microstresses and strains. The individual microstresses and strains cannot be measured at the time being, our instruments giving only statistical averages.

(2) Permanent set in the material can have two causes: The heat movement of the molecules in materials under stress can produce changes in the distance between points having a finite distance (thermal plasticity). The individual crystal grains can undergo finite deformations through sliding along certain planes (athermal plasticity).

(3) Plasticity in polycrystalline metals is always a mixture of thermal and athermal plasticity. However, for iron and steel especially, one can state that plasticity begins at a certain definite value of the intensity of stress. The intensity of stress is given by the following expression, where σ_1, σ_2, and σ_3 are the principal stresses and $\sigma = 1/3\ (\sigma_1 + \sigma_2 + \sigma_3)$ is the hydrostatic part of the stress:

$$\text{Intensity } \sigma_r = \sqrt{(\sigma_1 - \sigma)^2 + (\sigma_2 - \sigma)^2 + (\sigma_3 - \sigma)^2} \ldots [1]$$

The intensity is *essentially* a positive quantity and is independent of the intensity σ of the hydrostatic stress. The condition of plasticity is expressed by the equation:

$$|\sigma_r| = 2P = \text{constant}$$

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[1] Associate Professor of Mechanics, Massachusetts Institute of Technology. Mem. A.S.M.E. Professor Hencky received the certificate of Civil Engineering from the Technical University of Munich (1908) and the degree of Doctor from the University of Darmstadt (1913). Before the war he spent a few years with the States Railways in Alsace-Lothringen in Strasbourg and in Russia. At the outbreak of the war he was taken prisoner in Charcow and was interned in the Ural district. Afterward he was a lecturer for three years in the Universities of Darmstadt and of Dresden and for eight years with the University of Delft, Holland, on applied mechanics. He has been with the Massachusetts Institute of Technology for three years.

[2] Compare A. Nadai, "The Creep of Metals," Trans. A.S.M.E., Applied Mechanics Division, April-June, 1933, vol. 1, no. 2, paper APM-55-10. A fairly complete reference list of all important publications considering creep can be found therein.

[3] Compare C. R. Soderberg, "Working Stresses," Trans. A.S.M.E., Applied Mechanics Division, July-September, 1933, vol. 1, no. 3, paper APM-55-14.

Contributed by the Applied Mechanics Division and presented at the Semi-Annual Meeting, Chicago, Ill., June 26 to July 1, 1933, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] A number of practical problems has been treated by A. Nadai in his paper, "The Creep of Metals" (see footnote 2). Compare also, Gleason MacCullough, "Applications of Creep Tests," Trans. A.S.M.E., Applied Mechanics Division, July-September, 1933, vol. 1, no. 3, paper APM-55-12.

(4) At elevated temperatures the loaded material does not reach a static equilibrium, but shows a permanent creep. For high rates of strain an empirical formula is known, connecting strain velocity and stress. A workable theory of plasticity must embody this formula (as recommended by Nadai and Soderberg, loc. cit.).

(5) Just as we have the conception of active and passive earth pressure in the statics of loose masses, must we distinguish between the elastic-plastic system in the loading phase and the elastic-plastic system in the unloading phase. Practically only in the loading phase of a system can plasticity occur. The condition for loading is, with $\bar{\sigma}_r$ as the intensity of the microstresses:

$$\frac{d\bar{\sigma}_r}{dt} > 0; \quad \frac{d\sigma_r}{dt} = 0$$

The condition for unloading is:

$$\frac{d\sigma_r}{dt} < 0$$

We will see presently that it is possible to embody all these points in a mathematical theory which contains the classical theory as a special case and at the same time solves the problem of creep for small strain velocities without making arbitrary assumptions. Before we can give mathematical formulas we must explain what we should understand by the terms "stresses" and "microstresses."[6]

The macromechanical stress is the force acting on a small unit area consisting of the cross-sections of many hundreds of crystal grains. If we could measure the stress intensity in the individual grain, we would not find any constancy of the stress in passing from one crystal to another. Would it now have any practical meaning to introduce the true stress? Obviously not, because these individual stresses do not interest us very much. But the true elastic energy stored up in the body is of great interest to us because we have to pay for it an equivalent amount of mechanical energy.

The mechanical-energy equivalent connected with the average macromechanical stress is naturally smaller than the true elastic energy. However, we can now imagine a stress system $\bar{\sigma}_i$ ($i = 1, 2, 3$), which added to the system σ_i ($i = 1, 2, 3$) would yield just the amount of the true elastic energy. This system $\bar{\sigma}_i$ we will call the micro-stress system.

To help somewhat our imagination in handling these two systems, we assume each system to be connected with an elastic structure of its own and both structures as being forced to occupy the same space. Let us call these two structures the $\bar{\sigma}_i$ system and the $\bar{\sigma}_i$ system.

Assuming now for a moment that the $\bar{\sigma}_i$ system does not exist, we arrive at the classical theory of the ideally plastic body.

Denoting the principal strains by $\epsilon_1, \epsilon_2, \epsilon_3$, we can separate the change in volume by subtracting $\epsilon = 1/3(\epsilon_1 + \epsilon_2 + \epsilon_3)$, and get the pure deformation $\epsilon_i - \epsilon$ ($i = 1, 2, 3$).

In the elastic region, with $\sigma_r \leqq 2P$, we will have the law of Hooke in the form:

$$\sigma_i - \sigma = 2G(\epsilon_i - \epsilon) \ (i = 1, 2, 3) \ldots\ldots\ldots\ldots [2]$$

The strain is measured by the logarithm of the length after divided by the length before the deformation took place. The elastic law for changes of volume

$$\sigma = 3K\epsilon \ldots\ldots\ldots\ldots\ldots\ldots [2a]$$

remains the same in the elastic and in the plastic region. If $\sigma_r = 2P$, and we are loading the system further, we have to introduce the conception of intensity of strain:

$$\epsilon_r - \epsilon = \sqrt{(\epsilon_1 - \epsilon)^2 + (\epsilon_2 - \epsilon)^2 + (\epsilon_3 - \epsilon)^2} \ldots\ldots [2b]$$

giving the relation

$$\sigma_i - \sigma = 2P \frac{\epsilon_i - \epsilon}{\epsilon_r - \epsilon} \ (i = 1, 2, 3) \ldots\ldots\ldots\ldots [2c]$$

and satisfying the condition of plasticity $\sigma_r = 2P$.

The unloading of such a system proceeds always without further permanent set if the remaining stresses after unloading do not exceed the allowed stress intensity.

The difficulty with this ideally plastic body is that in loading a test piece the stress would be the same no matter with which velocity of strain we are performing the experiment above the elastic limit. Now the work done in straining the body must be transformed into heat vibrations, and this transformation takes time. Moreover, if we shorten that time, we must overcome the mass inertia of the molecules, which manifests itself in an entirely independent manner from the macromechanical mass accelerations taken care of in the fundamental equation of dynamics.

The ideally plastic body is therefore a statical possibility, but a dynamical impossibility. Moreover, as the metals have a grain structure, the true stress is discontinuous at the grain boundaries and a statical extra-resistance has to be accounted for. On account of the mathematical difficulties, we assume the simplest law possible for this hardening effect—namely:

$$\bar{\sigma}_i - \bar{\sigma} = 2H \ (\epsilon_i - \epsilon) \ldots\ldots\ldots\ldots\ldots [3a]$$

Accordingly we put $\bar{\sigma} = 0$. The total stress will be:

$$\sigma_i + \bar{\sigma}_i - \sigma = 2(G + H)(\epsilon_i - \epsilon) \ldots\ldots\ldots [3b]$$

It must be understood that the hardening effect exists *also in the elastic region*, but cannot be noticed there because the strain-hardening coefficient H is so small compared with G that even the fluctuations in the value of G are greater. We have now a two-phase system, or two elastic structures forced to the same deformations, only that one system ($\bar{\sigma}_i$) has, compared with the other (σ_i), a very small modulus of shear ($H \cong 10^{-3} G$). The σ_i system has the peculiarity that the stress intensity never can exceed a constant value $2P$. The stress intensity $\bar{\sigma}_r$ of the $\bar{\sigma}_i$ system is surely limited also, but at present we do not know much about those limitations. It is certain only that with increase of temperature a marked relaxation sets in. It is quite possible that a certain intensity of the microstress must be reached before relaxation begins. We shall come back to this point farther on.

Here we shall assume that relaxation immediately sets in, but only in the $\bar{\sigma}_i$ system. The σ_i system is not subject to any relaxation.

If we remember that the $\bar{\sigma}_i$ system is simply the representative of the elastic energy due to the discontinuities of the stress at the grain boundaries, we can understand that a mechanical energy concentrated on so small a space is likely to disintegrate into heat movement of the molecules and that this must happen so much more easily if the energy of the microstresses is bigger.

It is obvious that the Equation [3a] has to be replaced by another one introducing the disintegration of energy according to Maxwell:

$$\frac{d\bar{\sigma}_i}{dt} = 2H \cdot \frac{d(\epsilon_i - \epsilon)}{dt} - \frac{\bar{\sigma}_i}{T} \ldots\ldots\ldots\ldots [4]$$

Now if we want to embody the results of Deutler, Nadai, and Soderberg in our theory, we have only one choice for the relaxa-

[6] Compare, also, A. Nadai, "Plasticity," chaps. 25 and 38.

tion time T. It must depend on the intensity of stress in the $\bar{\sigma}_i$ body. We must have:

$$T = T_0 \frac{\omega|\bar{\sigma}_r|}{e^{\omega|\bar{\sigma}_r|} - 1} \quad [4a]$$

T_0 is the relaxation time for small values of ω, $1/\omega$ has the dimension of a stress and is a new important constant.

A simple model of such an elastic-plastic system would be a common iron test piece with a rubber skin glued to the surface. Plastic deformation of the inner test piece would be accompanied by elastic extension of the rubber skin.

On account of the big difference in the elastic moduli, the unloading would release only a small percentage of the stress in the rubber. If now we were using raw rubber, relaxation would come into play, and after some days the stress would have disappeared in the rubber also.

We see that the Equations [1] to [4] form a suitable and usable scheme for the study of the mechanics in the inelastic states of matter. Every new coefficient introduced has a clear mechanical meaning.

Equation [1] applied to different stress systems yields different numerical coefficients.

A number of interesting problems concerning power-press operations can be studied and the experimental work can be simplified considerably through the use of Equation [1].

II—Some Applications of the Equations of Plasticity and Creep

The material is considered incompressible. We are leaving out the indices for the stresses which must not lead to confusion with $\sigma = \frac{1}{3}(\sigma_1 + \sigma_2 + \sigma_3)$, the symbol of the hydrostatic part of the stresses.

(1) *The amount of mechanical energy needed to change the form of a test piece.*

Let us consider first the simple tensile and compression test. For the uniaxial strain we use the symbol λ. σ = stress. $\sigma_0 = \sqrt{3/2}\,P$ = the uniaxial limit of elastic stress.

Our equations simplify to $\sigma = 3G \cdot \lambda$, in the elastic region; $\sigma = \sigma_0 + \bar{\sigma}$, in the plastic region.

$$\frac{d\bar{\sigma}}{dt} = 3H \cdot \frac{d\lambda}{dt} - \frac{\bar{\sigma}}{T} \quad [1]$$

$$T = T_0 \frac{\omega|\bar{\sigma}|\sqrt{2/3}}{e^{\omega|\bar{\sigma}|\sqrt{2/3}} - 1} \quad [1a]$$

For the sake of simplicity we write simply ω instead of $\omega \cdot \sqrt{2/3}$, and obtain:

$$T = T_0 \frac{\omega|\bar{\sigma}|}{e^{\omega|\bar{\sigma}|} - 1} \quad [1b]$$

If one end of the test piece is fixed and the other moves with the speed v, this velocity cannot be constant if we wish to have stationary flow.

For stationary flow we have $d\bar{\sigma}/dt = 0$ and $d\lambda/dt$ = constant. With the length l_0 at time $t = 0$ and speed $v = v_0$, l at time t, and $v = dl/dt$, we get:

$$\frac{d\lambda}{dt} = \frac{d}{dt}(ln\,(l/l_0)) = \frac{dl}{dt} \cdot \frac{1}{l} = \frac{v}{l}$$

whereas $v_0/l_0 = v/l$

as a condition of stationary flow.

Combining [1] and [1b] and remembering that in formula [1b] the absolute value has to be introduced, we get:

$$\bar{\sigma} = \pm 1/\omega \cdot ln\,\{1 + 3H \cdot \omega \cdot v_0 \cdot T_0/h_0\} \quad [2]$$

where + = tension and — = compression.

The behavior of the material according to the assumed law of creep is perfectly symmetrical with respect to tension or compression, but the end of the test piece has to be accelerated in the case of tension and retarded in the case of compression in order to get v/l constant.

If F is the instantaneous cross-section, the work will be $W \cdot V = \int (\sigma_0 + \bar{\sigma}) \cdot F \cdot \delta l$, and with $F \cdot l = V$ = constant:

$$W = (\sigma_0 + 1/\omega\, ln\,[1 + 3H\omega v_0 T_0/h_0]) \cdot ln(l/l_0) \quad [3]$$

This equation can be used to define a measure of ductility. We take a constant amount of W and compare the corresponding strains. Applying the energy W in form of kinetic energy and solving Equation [1] for the case of impact, we get a measure of dynamic ductility or malleability.

In general we must emphasize that the use of any proportionality between stress and strain velocity is very objectionable. To prove that, we assume the simpler case of small ω. Maxwell's law is then:

$$\frac{d\bar{\sigma}}{dt} = 3H \cdot \frac{d\lambda}{dt} - \frac{\bar{\sigma}}{T_0} \quad [4]$$

Let us assume a velocity v is rapidly decreasing, so that $d\lambda/dt = v_0/l_0 \cdot e^{-t/T_0}$. Introducing this into [4], we have the equation to solve:

$$\frac{d\bar{\sigma}}{dt} + \frac{\bar{\sigma}}{T_0} = \frac{3H \cdot v_0}{l_0} e^{-t/T_0} \quad [4a]$$

and the solution is:

$$\bar{\sigma} = e^{-t/T_0}\left(\bar{\sigma}_0 + \frac{3H \cdot v_0}{l_0} \cdot t\right)$$

with $\bar{\sigma} = \bar{\sigma}_0$ for $t = 0$; or using the value of $d\lambda/dt$:

$$\bar{\sigma} = \left(\frac{\bar{\sigma}_0 v_0}{l_0} + 3Ht\right)\frac{d\lambda}{dt} \quad [4b]$$

In this formula we have the so-called coefficient of friction as a function of the time, showing clearly that in general a friction analogous to viscosity in liquids does not exist, if we assume relaxation.

To come back to our tensile and compression test in the plastic region, we would like to know if the law of friction as it is given for stationary flow in Equation [2] can be approximately applied even in case the speed does not satisfy the condition $v_0/l_0 = v/l$. Most of our testing machines are built so that the end of the test piece has a constant speed. In this case $l = l_0 + v_0 \cdot t$; $d\lambda/dt = v_0/l$; and $dl = v_0 \cdot dt$.

Introducing $x = l/v_0T_0$ as a new variable instead of t and $\bar{\sigma} = 3H\Sigma$ to extract the dimensional quantities, Equation [4] becomes:

$$\frac{d\Sigma}{dx} + \Sigma = \frac{1}{x} \quad [5]$$

The solution of this equation leads to a new function, the integral logarithm.[6] The symbol of this function is:

[6] Compare Jahnke-Emde, "Funktionentafeln," pp. 19–23, 1909.

$$Ei(x) = \int_{\infty}^{x} \frac{e^x}{x}\, dx$$

The solution of [5] gives:

$$\Sigma = e^{-x}\left[+ \frac{e^{+x_0}}{x_0} + Ei(x) - Ei(x_0)\right] \ldots\ldots [6a]$$

for tension, and:

$$\Sigma = e^{+x}\left[- \frac{e^{-x_0}}{x_0} + Ei(-x) - Ei(-x_0)\right] \ldots\ldots [6b]$$

for compression. Equation [2] would yield, with $\omega = 0$, $\sigma = 3H \cdot v_0 T_0/l$.

In the new notations therefore:

$$\Sigma = 1/x \ldots\ldots [6c]$$

For the sake of comparison we give the values of Σ corresponding to [6a], [6b], and [6c]; [6c] corresponds to the law of friction used in hydrodynamics:

Tension			Compression		
x	Σ [6a]	Σ [6c]	x	$-\Sigma$ [6b]	$-\Sigma$ [6c]
1.0	1.0	1.0	2.0	0.5	0.5
1.2	0.99	0.84	1.8	0.51	0.55
1.4	0.99	0.72	1.6	0.52	0.63
1.6	0.97	0.63	1.4	0.55	0.72
1.8	0.94	0.55	1.2	0.59	0.84
2.0	0.91	0.50	1.0	0.65	1.00

The comparison shows the deviations from the fluid viscosity very strongly. Naturally for higher values of x, [6a] and [6b] merge asymptotically into $1/x$. But only for values $x \geqq 17$ equality is reached approximately.

The use of $1/x$ would lead to calculations of the amount of work which would be too small for tension and too large for compression, whereas the use of our equations would yield results comparable with the experiment and therefore also suggesting new experiments.

As the relaxation time depends on the intensity of stress, the stress must be applied in as concentrated a form as possible so as to get the greatest effect with the smallest amount of energy. This is also the reason why in rolling mills of more recent construction small cylinders are used in direct contact with the plastic material. To hinder any bending, these rollers are backed up by bigger ones. It is true that the rolling-mill experts have found out these things without the direct aid of the mathematical theory, but this shows only that a number of improvements could be made even in the manufacturing processes by determined application of the laws of plasticity and creep.

(2) *The tensile test with metal wires.*

Developing a mathematical theory of plastic deformation means the amassing of a great amount of information, considering even such a trivial thing as a simple tensile test.

We want now to find out what happens if we load a wire so that the plasticity limit is surely passed by the stress when we then leave the wire to itself, with the load hanging on.

There is a dynamical problem involved also, because of the mass acceleration of the load, but on account of the slow movement we can neglect this influence.

We have in the elastic region, as before

$$\sigma = 3G \cdot \lambda \ldots\ldots [1]$$

and at the same time the hardening stresses

$$\bar{\sigma} = 3H \cdot \lambda \ldots\ldots [2]$$

and the total stress $\sigma + \bar{\sigma} = 3(G + H) \cdot \lambda$ in the elastic region. After having passed the plasticity limit, we have the stress $\sigma_0 + \bar{\sigma} = \sigma_0 + 3H \cdot \lambda$ up to a certain critical intensity of the hardening stress $\bar{\sigma}'$. If that stress is passed also, relaxation sets in.

With $\bar{\sigma}'$ as this limit, the law of relaxation will be expressed by

$$\frac{d\bar{\sigma}}{dt} = 3H \cdot \frac{d\lambda}{dt} - \frac{\bar{\sigma} - \bar{\sigma}'}{T} \ldots\ldots [3]$$

$$T = T_0 \frac{\omega|\bar{\sigma}|}{e^{\omega|\bar{\sigma}|} - 1} \ldots\ldots [3a]$$

The introduction of a limiting intensity of the hardening stresses is hypothetical and has to be justified. But this can be done easily. All elastic constants are functions of the temperature, and we know that temperature is only an indirect manner of measuring the vibrational energy of the molecules. But we have seen that the stress intensity of the hardening stresses is likewise only an equivalent to the elastic energy of these microstresses. The mechanical energy on its way to disintegrate into heat can assume an intermediate level in the form of elastic microenergy. In relation to macromechanical energy, this elastic energy of the microstresses in the grain boundaries comes into account only statistically, and *the intensity of these stresses* as defined by us *is a scalar quantity of the same character as the temperature.* Physically, our stress intensity is a sort of "cold" temperature. As quite a considerable amount of energy is, in solid bodies, frozen in in the form of elastic energy, we cannot build up the mechanics of the inelastic state without these two parameters—the temperature and the stress intensity.

But let us come back to our example. The problem is properly a two-dimensional one, because λ is a function of the place as well as of the time; but in stretching the wire under a constant load, we have with F as instantaneous cross-section $F(\sigma_0 + \bar{\sigma}) =$ constant; therefore, differentiating, we get:

$$(\sigma_0 + \bar{\sigma})\ \delta F + F \cdot \delta\bar{\sigma} = 0$$

But because of the incompressibility of the material assumed

$$\delta(F \cdot l) = F \cdot \delta l + l \cdot \delta F = 0$$

Combining these two equations gives

$$\delta\bar{\sigma} = (\sigma_0 + \bar{\sigma}) \cdot \delta l/l = (\sigma_0 + \bar{\sigma}) \cdot \delta\lambda \ldots\ldots [4]$$

and after integration

$$\sigma_0 + \bar{\sigma} = (\sigma_0 + \bar{\sigma}_0)e^{\lambda - \lambda_0} \ldots\ldots [4a]$$

where

$$\lambda_0 = \frac{\sigma_0 + \bar{\sigma}_0}{3(G + H)} \cong \frac{\sigma_0}{3G} \ldots\ldots [4b]$$

Introducing [4] into Equation [3], we get an important relation which allows us to calculate a limiting strain. If this strain limit is reached, the wire will no longer elongate uniformly, but will show a sudden reduction in cross-section somewhere, and will break.

The combination of [4] and [3] yields

$$\frac{d\bar{\sigma}}{dt} = \frac{(\bar{\sigma} - \bar{\sigma}')(\sigma_0 + \bar{\sigma})}{T \cdot [3H - (\sigma_0 + \bar{\sigma})]} \ldots\ldots [5]$$

This equation is meaningless for $\bar{\sigma} < \bar{\sigma}'$, but if this limit is passed, the increase in stress becomes infinite for

$$3H = \sigma_0 + \bar{\sigma} = (\sigma_0 + \bar{\sigma}_0)e^{\lambda - \lambda_0} \text{ (using [4a])} \ldots\ldots [5a]$$

or

$$e^{\lambda} = e^{\lambda_0} \cdot \frac{3H}{\sigma_0 + \bar{\sigma}_0} \cong 3H/\sigma_0 \ldots\ldots\ldots\ldots [5b]$$

The critical strain is therefore independent of the speed $\lambda_{critical} = ln(3H/\sigma_0)$.

The strain reached until reduction of the cross-section sets in allows us to calculate H if σ_0 is known. We see that under all circumstances $3H > \sigma_0$.

The integral of Equation [3] can be expressed by a series of integral logarithms, but it is simpler to use numerical integration.

The critical stress $\bar{\sigma}'$ can be found by precise measurements of the creep rate, which has to be zero at this limit.

The only hypothetical part in our theory is the assumption of the same law of elasticity for the microstresses as for the elastic stresses. But even if we shall find here any deviation, we shall nevertheless continue to use the linear law. Even so, the mathematical difficulties of space problems are enough.

(3) *The moving of a cylinder in a plastic mass and the depth to which the zone of plastic deformation can penetrate.*

The depth to which the zone of plasticity can penetrate is important for the design of manufacturing machines for metal forming. The limiting surface between the plastic and the elastic region is weaker than the elastically strained part, and the forming of such surfaces of separation must be restricted as much as possible. In the extruding of metals we would have dead corners for the plastic movement, mostly there where we would have eddies in the case of a fluid.

A great aid for the solution of plastic-flow problems is that the flow lines can be calculated as if we had a viscous liquid. Only for the relations between stress strain and strain velocity the laws of viscous fluids are not applicable. In general, one cannot treat in Euler coordinates the movement of a plastic-elastic body, because the elastic energy is connected with the moving mass and not with the space element at which it happens to be for a moment.

However, the case of a cylinder moving in a plastic mass is an exception and can be treated by classical methods, because the flow lines are circles and the deformation is constant along such a circle.

Let us assume $-\tau_0$ as the limiting shearing force of the plastic material and $\bar{\tau}$ as the strain-hardening influence. The equation of equilibrium alone determines the stress distribution. It becomes, using polar-coordinates:

$$\frac{d(-\tau_0 + \bar{\tau})}{dt} - \tau_0 + \bar{\tau} = 0 \ldots\ldots\ldots\ldots [1]$$

We shall denote with
a = radius of cylinder
r = radius of an element
r_0 = radius of the boundary between the elastic and the plastic part.

Putting $r = a \cdot \rho$ to concentrate the dimensional quantities, we get the solution:

$$\bar{\tau} = -\tau_0 [(\rho_0/\rho)^2 - 1] \ldots\ldots\ldots\ldots [2]$$

ρ_0 is unknown and can be found only when the formula for the velocity is established. The only velocity which a point can have is perpendicular to the radius. With v as velocity at r and v_a at $r = a$, we get the strain velocity

$$\gamma = \frac{dv}{dr} - \frac{v}{r} \ldots\ldots\ldots\ldots [3]$$

and for stationary flow from $\frac{d\bar{\tau}}{dt} = G \cdot \gamma - \frac{\bar{\tau}}{T}$, with $\frac{d\bar{\tau}}{dt} = 0$:

$$\bar{\tau} = GT \cdot \gamma \ldots\ldots\ldots\ldots [4]$$

where

$$T = T_0 \frac{\omega|\bar{\tau}|\sqrt{2}}{e^{\omega|\bar{\tau}|\sqrt{2}} - 1} \ldots\ldots\ldots\ldots [4a]$$

Combining [2], [3], [4], and [4a], we get the differential equation:

$$\frac{dv}{d\rho} - \frac{v}{\rho} = -\frac{a}{G\omega T_0\sqrt{2}} \left(e^{\tau_0\omega\sqrt{2}\left[\left(\frac{\rho_0}{\rho}\right)^2 - 1\right]} - 1 \right) \ldots [5]$$

The solution is given by

$$v = \rho \left[v_a - \frac{a}{G\omega \cdot T_0 \cdot \sqrt{2}} \int_1^{\rho} (e^{\omega\tau_0\sqrt{2}((\rho_0/\rho)^2 - 1)} - 1) \frac{d\rho}{\rho} \right]. [5a]$$

We see immediately that the condition $v = v_a$ is fulfilled for $\rho = 1$. The integral can easily be reduced to the integral logarithm.

Denoting again with $Ei(y)$ the integral $\int_{\infty}^{y} e^y \frac{dy}{y}$, we get the solution of [5a] in the form:

$$v = a\rho \left(\frac{v_a}{a} + \frac{ln\,\rho}{G \cdot \omega \cdot T_0\sqrt{2}} - \frac{e^{-\omega\tau_0\sqrt{2}}}{2G\omega T_0\sqrt{2}} \left[Ei(\beta) - Ei\left(\frac{\beta}{\rho^2}\right) \right] \right)$$

$$\beta = \omega\tau_0\sqrt{2}\rho_0^2 \ldots\ldots [6]$$

For $\rho = \rho_0$, we get an equation in which ρ_0 is the only unknown quantity:

$$\frac{Ei(\omega\tau_0\sqrt{2}\rho_0^2) - Ei(\omega\tau_0\sqrt{2})}{2e^{\omega\tau_0\sqrt{2}}} - ln\,\rho_0 = \frac{v_a}{a} \cdot T_0\omega G \cdot \sqrt{2} \ldots [7]$$

For large values of $\omega\tau_0\sqrt{2}$, it is comparatively easy to solve this transcendental equation numerically. For instance, for $\omega\tau_0\sqrt{2} = 20$, $\frac{v_a}{a} T_0\omega G\sqrt{2} = 1000$, we get $\rho_0 = 1.22$.

Amplitudes of Non-Harmonic Vibrations[1]

By J. P. DEN HARTOG,[2] CAMBRIDGE, MASS.

IN THE usual theory of the vibration of an undamped single-mass system, the spring is assumed to have a linear characteristic, i.e., the force-deflection diagram is represented by a straight line through the origin. It is known that the free as well as the forced vibrations of such a system can be completely calculated in all detail. This is entirely different when the spring characteristic ceases to be linear. Then only the *free* vibrations of the system can be determined for a given starting condition by a process of graphical or numerical calculation. Though this is rather complicated and generally requires a considerable amount of work, at least the solution can be obtained with any desired degree of accuracy, according to the simple rule "the more work, the greater accuracy."

The results thus found are by no means as general as those for the case of linear springs, because the natural frequency comes out to be dependent on the amplitude. Moreover, the principle of superposition ceases to be valid, so that each particular case has to be solved by itself.

When steady-state *forced* vibrations are considered, the situation becomes even worse. Here not only is there no superposition, but even the golden rule of "the more work, the greater accuracy" is no longer true. An approximate solution exists, originated by Martienssen, on the analogous electrical problem of the alternating current in a circuit containing a condenser and an iron-cored inductance in series. This solution was later applied to mechanical problems by Duffing and by Rüdenberg. For the special cases of spring characteristics represented by an ordinary or by a cubic parabola, Duffing in his book gives a method whereby this result can be improved by successive approximations. An exact solution was obtained by S. J. Mikina and the author for spring characteristics made up of three sections of straight lines. For that case (the only one for which an exact solution exists so far) a comparison was made with the approximation of Martienssen, showing that under unfavorable circumstances this approximation may be 75 per cent off or more.

In this paper it is pointed out that Martienssen's original graphical solution follows a somewhat arbitrary procedure in that he satisfies only the condition of equilibrium between the various forces at the end of the stroke and thus necessarily violates other conditions. The solution can be modified by satisfying some other condition—for instance, the energy relation during a quarter cycle of the motion. When this is done the condition of equilibrium at the end of the stroke is violated. This "quarter cycle energy method" gives results which are no better than those of the Martienssen solution.

Another graphical solution has been worked out whereby two conditions are satisfied simultaneously. These two were chosen to be the equilibrium of the forces at the end of the stroke ($\omega t = 90°$) and also at another point somewhere between the neutral middle position and the end of the stroke ($\omega t = 60°$). This method naturally gives results which are considerably more accurate than those of Martienssen. They can be obtained by a rather simple graphical construction described in the paper.

The numerical results of the various methods are compared with each other for the case of the broken-straight-line characteristic for which the exact solution is available.

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[1] Abstract of paper presented at the Semi-Annual Meeting, Chicago, Ill., June 26 to July 1, 1933, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS; for full text see *Journal Franklin Institute*, October, 1933, pp. 459–473.

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APM-55-19

Stress-Concentration Phenomena in Fatigue of Metals

By R. E. PETERSON,* EAST PITTSBURGH, PA.

This paper consists of two parts: (I) General Considerations, which is a brief review of available data on the effect of holes, fillets, grooves, etc., on endurance strength; and (II) Some Observations Concerning Fatigue Fractures, wherein notes on crack propagation as related to stress-concentration phenomena are presented.



THE effect of stress concentration[1] on the endurance of metals subjected to alternating stresses is of importance to engineers, since stress concentrations are invariably present in machines owing to fillets, grooves, holes, threads, keyways, etc., in their parts. It is proposed in this paper, first, to review briefly some of the significant results of value to the engineer and, second, to present certain observations concerning fatigue fractures. The paper deals primarily with the rotating-beam fatigue test (alternating bending), but one section is devoted to alternating torsion tests.

I—GENERAL CONSIDERATIONS

1—Terminology[2]

In mathematical analyses of stress concentration based on the theory of elasticity, the results are usually stated in terms of a "theoretical stress-concentration factor"

$$K = \frac{\text{maximum stress}}{\text{average nominal stress}}$$

(The average nominal stress $S_n = P/a$ = load/net area, or in bending, $S_n = M/(I/c)$ = moment/net section modulus.)

Similarly, the results of fatigue tests of fillets, holes, grooves, etc., are stated in terms of a "fatigue stress-concentration factor"

$$k = \frac{S_e}{S_e'} = \frac{\text{ordinary endurance limit (without stress concentration)}}{\text{endurance limit with stress-concentration effect (fillet, hole, etc.)}}$$

Sometimes it is of value to express the percent decrease (c) of endurance strength due to stress concentration.

$$c = \frac{S_e - S_e'}{S_e} \times 100$$

It is apparent that $c = (1 - 1/k)100$. As a measure of fatigue stress-concentration effect in comparison with the corresponding theoretical value, a "stress-concentration index" may be used:

$$q = \frac{k - 1}{K - 1}$$

The value of q ranges from zero (when $k = 1.0$, as for certain tests of cast iron) to unity (should the fatigue stress-concentration factor k attain the theoretical value K; up to the present, however, no cases of this seem to be on record).

2—Material as a Variable

While theoretical stress-concentration factors are independent of the material (in so far as it conforms to Hooke's law), test data show that fatigue stress-concentration factors (k) are *not*. Although insufficient data are available at present, it seems that the variation of k for similar test pieces of different materials cannot be correlated completely with any of the ordinary properties of materials, such as ductility,[3] hardness, etc. For steels a possible relation exists between fatigue stress-concentration factors and ultimate strength. Fig. 1 shows the variation of k for a variety of steels of different ultimate strengths, all specimens having a 60-deg sharp groove 0.1 mm deep.[4] A similar relation is indicated by Fig. 2. Tests of cast iron have shown little or no fatigue stress-concentration effect,[5] the reason apparently being that the flakes and pores inherent in cast iron produce considerable stress concentration so that the effect of an imposed discontinuity is largely masked. It has been suggested that materials having small stress-concentration effect in fatigue are also materials having large internal damping,[6] and vice versa.

3—Effect of Size of Discontinuity

If the material and size of a specimen are kept constant and the size of a discontinuity (such as a hole) is varied, theoretical stress-

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[1] For general discussions of stress concentration see Timoshenko, "Strength of Materials," vol. ii, chap. vi (Van Nostrand); also Seely, "Advanced Mechanics of Materials," part iii (McGraw-Hill).

[2] The German investigators use expressions corresponding to those mentioned in this section. *Formziffer* (α_k) is the same as "theoretical stress-concentration factor" (K); *Kerbwirkungszahl* (β_k) corresponds to "fatigue stress-concentration factor" (k); and *Kerbempfindlichkeit* is equivalent to "stress-concentration index" (q). See, for instance, Thum and Buchmann, "Dauerfestigkeit und Konstruktion" V.D.I. Verlag (1932).

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] Moore, R. R., "Effect of Grooves, Threads, and Corrosion on the Fatigue of Metals," Proc. Am. Soc. Test. Matls., vol. 26 (1926), part ii, p. 255.

[4] Mailänder, R., "Dauerbrüche und Dauerfestigkeit," *Kruppsche Monatshefte*, vol. 13 (1932), p. 65.

[5] Thum, A., and Ude, H., "Die mechanischen Eigenschaften des Gusseisens," *Z.V.D.I.*, vol. 74 (1930), p. 257.

[6] Heydekampf, G. S., "Damping Capacity of Materials," Proc. Am. Soc. Test. Matls., vol. 31 (1931), part ii, p. 157.

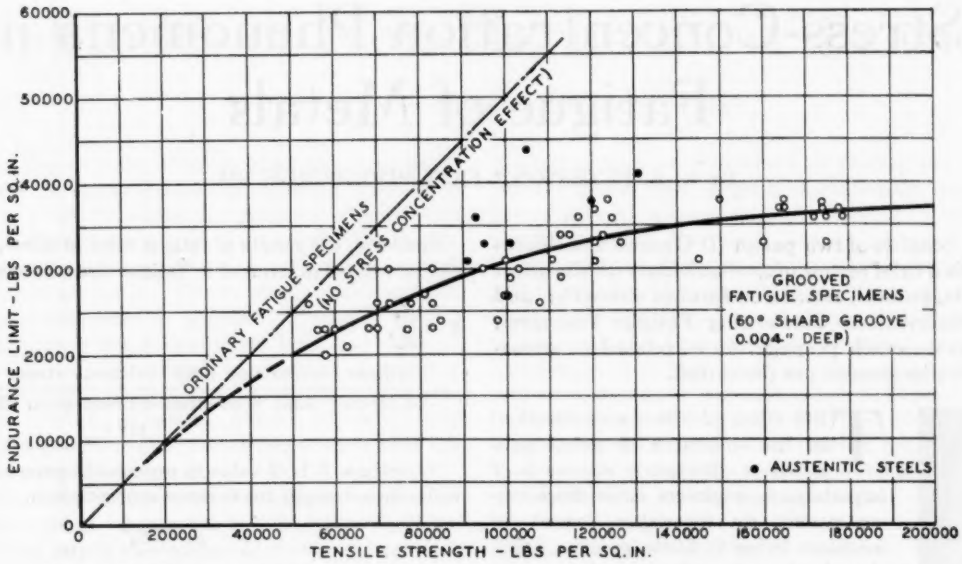


FIG. 1 ENDURANCE LIMITS FOR STEEL SPECIMENS (WITH GROOVES) AS DEPENDING ON MATERIAL

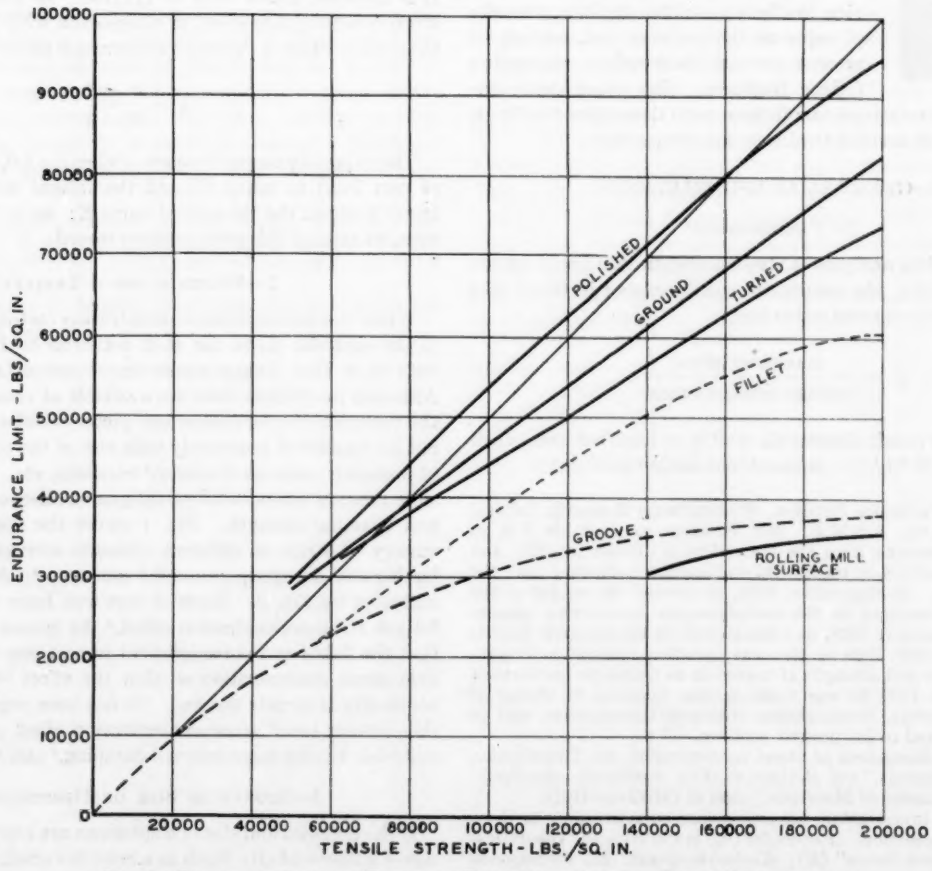


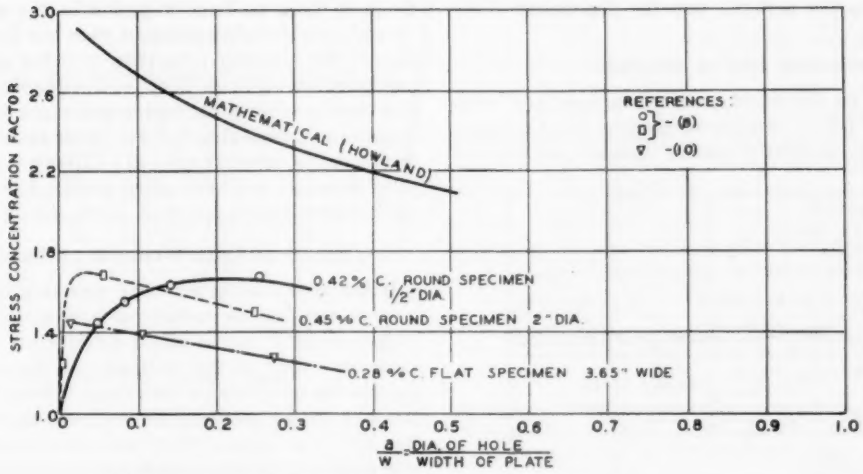
FIG. 2 EFFECT OF SURFACE FINISH ON ENDURANCE LIMIT OF STEEL SPECIMENS

concentration factors[7] increase as the size decreases. [Fig. 3(*a*).] Fatigue stress-concentration factors[8,9] show a similar tendency except for a marked decrease for very small discontinuities (but not necessarily small values of a/w). This is shown by plotting the data of Fig. 3(*a*) in terms of size of hole, as in Fig. 3(*b*). The upper curves of Fig. 3(*b*), representing data obtained with rotat-

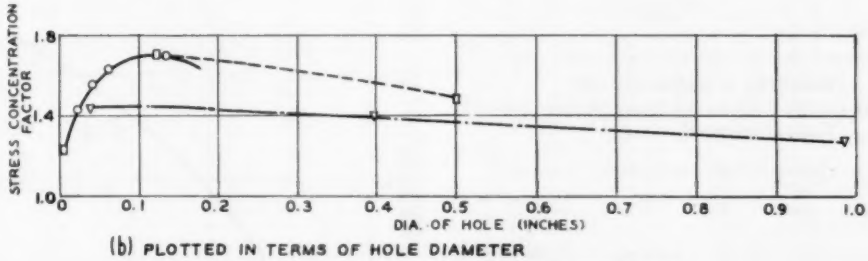
[7] Howland, R. C. J., "On the Stresses in the Neighborhood of a Circular Hole in a Strip Under Tension," Phil. Trans. Roy. Soc. Lond., vol. 229 A (1930), pp. 49–86.

[8] Data from Westinghouse Research Laboratories.

[9] Liljeblad, R., "An Investigation of the Fatigue of Metals Due to Locally Concentrated Stresses," no. 47, Ingeniörs Vetenskaps Akadamien. Published in English by Svenska Bokhandelscentralen (Stockholm).



(a) PLOTTED IN TERMS OF DIMENSIONLESS ABSCISSAS



(b) PLOTTED IN TERMS OF HOLE DIAMETER

FIG. 3 STRESS-CONCENTRATION FACTORS FOR HOLES

ing cantilever specimens of about the same material, indicate that for very small holes the fatigue stress-concentration factor may depend on the hole diameter regardless of the specimen size (except for very small specimens, since a/w must also be small).

Curves of fatigue stress-concentration factors for specimens having grooves tend also to have the same general form as the curve of theoretical factors.[10] However, as in the case of hole tests, the fatigue stress-concentration factors tend to approach unity for small grooves, the limiting case being fine scratches, which do not alter the endurance limit nearly as much as do deeper grooves.[11] (See also Fig. 1.) Different degrees of cold work due to the machining of different-sized discontinuities may be an important variable affecting the results, particularly in the case of small discontinuities.

Corresponding data for fillets are shown in Fig. 4. Here again the fatigue stress-concentration factors[8,12,13,14] follow the same tendency as the theoretical (photoelastic) stress-concentration factors,[15] although the former are lower than the latter. In general, it can be stated that fatigue stress-concentration factors are lower than theoretical stress-concentration factors, the devia-

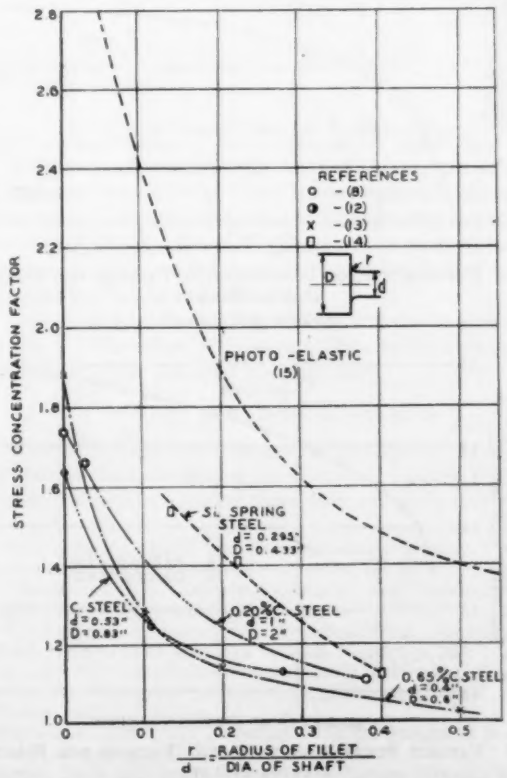


FIG. 4 STRESS-CONCENTRATION FACTORS FOR FILLETS

[10] See also Report of Research Committee on Fatigue of Metals (Appendix) "Present-Day Knowledge of Fatigue," Proc. Am. Soc. Test. Matls., vol. 30 (1930), part i, p. 299. *Note:* In this reference, data from reference no. 9 are incorrectly represented.

[11] Thomas, W. N., "The Effect of Scratches and of Various Workshop Finishes on the Fatigue Strength of Steel," British Aeronautical Research Committee Report no. 860.

[12] Stodola, A., and Schüle, A., *Schweiz. Bauzeitung*, vol. 71 (1918), p. 145.

[13] Lundgren, A., *Jernkontorets Annaler* (Swedish), vol. 115 (1931), p. 1, Nordiska Bokhandeln (Stockholm).

[14] Schneider, W., "Beitrage zur Frage der Schwingungsfestigkeit," *Stahl u. Eisen*, vol. 51 (1931), p. 288.

[15] Timoshenko, S., and Dietz, W., "Stress Concentration Produced by Holes and Fillets," Trans. A.S.M.E., vol. 47 (1925), p. 199.

tion depending upon the material and the dimensions of the specimen.

4—Effect of Size of Specimen

Fatigue-test data on size effect are very meager, and much remains to be done before definite conclusions can be drawn. In a few tests of geometrically similar fatigue specimens of different sizes, no large variation in the endurance limit was found[16] provided the specimens were free from stress concentration at the breaking zone (Fig. 5). For geometrically similar specimens having holes, fillets, and artificial cracks, it was found[17] that small specimens had higher endurance limits (lower values of k) than large ones (Figs. 6, 7, 8). Here again the cold work of the machining process may enter as a variable. A discussion of other possible causes has been given previously.[17] As stated before, further investigation is needed on the size-effect problem.

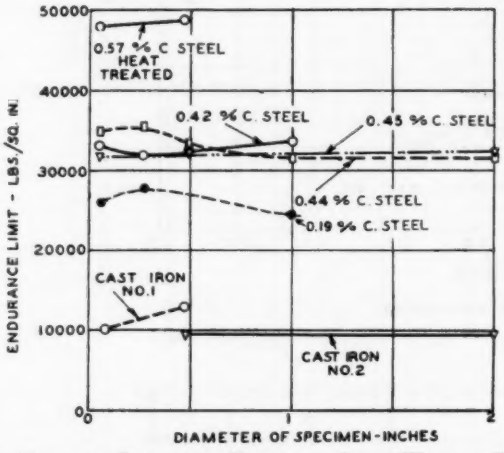


Fig. 5 Effect of Size on the Endurance Limit (Without Stress Concentration)

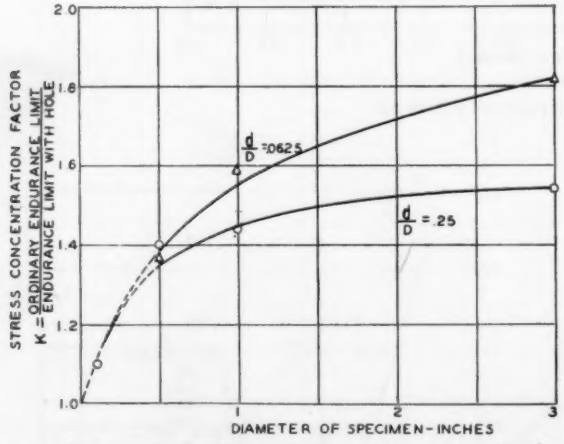


Fig. 6 Fatigue Stress-Concentration Factors for Specimens Having Holes
(0.45 per cent C steel.)

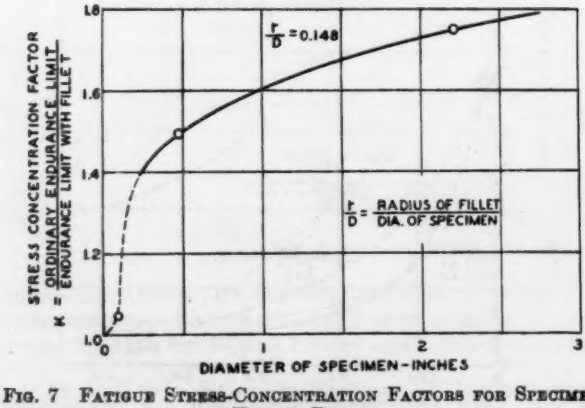


Fig. 7 Fatigue Stress-Concentration Factors for Specimens Having Fillets
(0.57 per cent C steel, heat treated.)

5—Effect of Cold Working on Fatigue Properties

It has been demonstrated by several investigators[18,19,20] that by cold working the surfaces of fatigue specimens the ordinary endurance limit can be raised considerably (as much as 30 per cent, depending on the material and degree of working). An indentation made with a center punch (conical end) into the surface of a fatigue specimen did not lower the endurance limit, the fracture occurring away from the indentation.[20] A hole of similar size made with a countersink reamer (conical end) reduced the endurance limit considerably (21 to 35 per cent, depending on the material). In the former case considerable plastic deformation took place. Apparently the beneficial effect of cold working is also operative in cases where stress concentration is present.

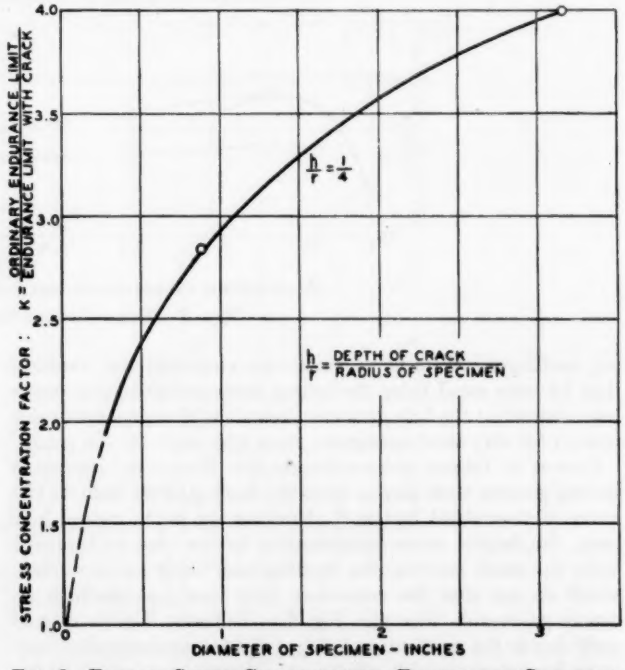


Fig. 8 Fatigue Stress-Concentration Factors for Specimens Having Artificial Cracks
(0.44 per cent C steel.)

6—Fatigue Strength of Machine Elements

A higher fatigue strength has been found for specimens having

[16] Peterson, R. E., "Fatigue Tests of Small Specimens With Particular Reference to Size Effect," Proc. Am. Soc. Steel Trtg., vol. 18 (1930), p. 1041.

[17] Peterson, R. E., "Model Testing as Applied to Strength of Materials," Trans. A.S.M.E., 1933, Applied Mechanics, vol. 1, no. 2, p. 79.

[18] Föppl, O., "Das Drücken der Oberfläche von Bauteilen aus Stahl," *Stahl u. Eisen*, vol. 49 (1929), p. 575.

[19] Behrens, P., "Das Oberflächendrücken zur Erhöhung der Drehschwingungsfestigkeit," N.E.M. Verlag, Berlin (1930).

[20] Armbruster, E., "Einfluss der Oberflächenbeschaffenheit auf den Spannungsverlauf und die Schwingungsfestigkeit," V.D.I. Verlag (1931).

TABLE 1 EFFECT OF SURFACE FINISH ON ENDURANCE LIMIT

(See also Fig. 2)

k = fatigue stress-concentration factor
c = decrease of ordinary endurance limit (polished specimen) } See Sec. 1 "Terminology"
(R) = rough; (S) = smooth

Material	Reference*	Rolling mill surface k	Rolling mill surface c	Turned k	Turned c	Filed k	Filed c	Ground k	Ground c	Emery k	Emery c
0.28% C steel	20	..	..	1.12	11	..	..	1.01	1	..	..
0.33% C steel	11	..	..	1.14	12	1.23 (R)	18	1.04 (S)	4	1.06 (R)	6
		..	..	..	..	1.09 (S)	8	..	..	1.02 (S)	2
0.49% C steel	24	..	..	1.21 (R)	17	..	..	1.11	10	1.06	6
		..	..	1.19 (S)	16	..	..	..	..	..	..
Chrome-nickel steel; 0.31% C; 1.42% Cr; 3.07% Ni.	20	..	..	1.15	13	1.09	8	1.03	3	..	..
0.35% C steel	25	1.25	20	..	..	..	..	..	..	..	..
Spring steel: 0.45% C, 2.0% Mn	25	2.18	54	..	..	..	..	..	..	..	..
Spring steel: 0.45% C, 1.0% Cr, 1.0% Si	25	2.38	58	..	..	..	..	..	..	..	..

* Numbers in column refer to footnotes in body of paper.

screw threads as compared with specimens having a single groove of screw-thread form.[3]

Rotating-beam fatigue tests of specimens with keyways[21] have shown higher strength for sled-runner keyways (cut with an ordinary milling cutter of same width as keyway) than for profiled keyways (cut with an end mill of same diameter as width of keyway). In these tests failure occurred at the end of the keyway.

Data are needed on the fatigue occurring at the edges of fitted members, such as at gear and pulley hubs. A general investigation of fatigue phenomena occurring in press fits would be of considerable value.

TABLE 2 FATIGUE STRESS-CONCENTRATION FACTORS FOR SPECIMEN HAVING TRANSVERSE HOLES

(See also Figs. 3 and 6)

Material	Reference*	Width (or diam.) w inch	Hole diam. ÷ width a/w	Fatigue stress-conc. factor k	% Red. end. lim. c	Theor. stress: conc. factor K	Stress-conc. index q
Armco iron (0.02% C)	26	0.30	0.183	1.63†	39	2.53‡	0.41
0.49% steel (H. T.)	26	0.30	0.183	1.51	34	2.53	0.33
0.52% C steel (H. T.)	26	0.30	0.183	1.58	37	2.53	0.38
Cyclops metal (annealed) (0.3% C, 8.3% Cr, 19.7% Ni)	26	0.30	0.183	1.25	20	2.53	0.16
Cast iron	27	0.35	0.157	1.15	13	2.57	0.10

* Numbers in column refer to footnotes in body of paper.
† Based on net section.
‡ See footnote no. 7.
H. T. = heat treated.

TABLE 3 FATIGUE STRESS-CONCENTRATION FACTORS FOR SPECIMENS HAVING GROOVES

(See also Fig. 1)

Material	Reference*	Depth t inch	Spec. diam. d inch	Rad. at bottom of groove r, inch	Fatigue stress-conc. factor k	% Red. end. lim. c	Theor. stress-conc. factor K†	Stress conc. index q
0.10% C steel	28	0.008	0.6	0.002	1.07	6.5	5	0.02
Do	25	0.008	0.3	0.002	1.21	17	5	0.05
0.14% C steel	..	0.025	..	0.0078	1.52	34	4.30	0.16
0.29% C steel (H. T.)	3	0.038	0.480	0.01	2.62	62	4.92	0.41
Do	3	0.076	0.480	0.015	2.50	60	5.3	0.35
Do	3	0.094	0.591	0.063	1.53	35	..	..
Do	3	0.156	0.716	0.125	1.31	24	..	..
Do	3	0.281	0.966	0.250	1.22	18	..	..
0.36% C steel	30	0.012	0.34	0.006	1.75	43	3.83	0.26
0.42% C steel	28	0.008	0.6	0.002	1.43	30	5	0.11
0.62% C steel	30	0.012	0.34	0.006	1.78	44	3.83	0.27
0.85% C steel (H. T.)	29	0.0035	..	0.0078	1.60	37	2.55	0.39
Do	29	0.025	..	0.0078	2.65	62	4.30	0.50
Do	29	0.050	..	0.0078	2.65	62	5.96	0.33
Spring steel: (0.56% C, 2.3% Si) rolled	25	...	..	....	1.33	25	5	0.08
Do	25	...	..	....	2.45	59	5	0.36
Do	25	...	..	....	2.57	61	5	0.39
Chrome-nickel steel (0.35% C; 6.2% Cr + Ni) annealed	30	0.012	0.34	0.006	1.95	49	3.83	0.34
Do., H. T.	30	0.012	0.34	0.006	2.2	55	3.83	0.42
Chrome-nickel steel (3.5% Ni, 0.8% Cr, H. T.)	28	0.008	0.6	0.002	2.00	50	5	0.25
Cast iron	28	0.008	0.6	0.002	1.00	0	5	..
Do	27	0.2	0.74	0.078	1.00	0	..	..
Do	27	0.078	0.5	0.078	1.09	8	3	0.05
Cast steel (0.2% C) annealed	25	0.008	0.3	0.002	1.31	24	5	0.08
Elect. copper (ann.)	28	0.008	0.6	0.002	1.29	22	5	0.07
Duralumin (ann.)	28	0.008	0.6	0.002	1.50	33	5	0.13
Do	25	...	..	....	1.2	17	5	0.05

* Numbers in column refer to footnotes in body of paper.
† $K = 1 + 2\sqrt{(t/r)}$ for small notches (footnote no. 31).
H. T. = heat treated.

[21] Peterson, R. E., "Fatigue of Shafts Having Keyways," Proc. Am. Soc. Test. Matls., vol. 32 (part II), p. 413.

TABLE 4 FATIGUE STRESS-CONCENTRATION FACTORS FOR SPECIMENS HAVING FILLETS

(See also Figs. 4 and 7)

Material	Reference*	d inch	D inch	r/d	Fatigue stress-conc. factor k	% Red. end. lim. c	Theor. stress-conc. factor K†	Stress-conc. index q
Armco (0.02% C)	24	0.275	0.40	Sharp corner	1.63	39	..	..
0.30% C steel	28	0.37	..	0.21	1.13	11	1.85	0.15
Do	28	0.37	..	0.053	1.59	37	2.80	0.33
0.33% C steel	32	0.6	1.25	0.083	1.47	32	2.45	0.32
Do	33	...	..	0.25	1.11	10	1.70	0.16
Do. (H. T.)	32	0.6	1.25	0.083	1.92	48	2.45	0.63
0.46% C steel (H. T.)	34	1	2	0.188	1.35	26	1.90	0.39
0.49% C steel (H. T.)	24	0.275	0.40	Sharp corner	2.04	51	..	..
0.53% C steel (H. T.)	34	1	2	0.188	1.42	30	1.90	0.47
Do. (H. T.)	34	2	2.5	0.156	1.37	27	2.07	0.35
0.60% C steel (H. T.)	32	0.6	1.25	0.083	1.62	38	2.45	0.43
Chrome-nickel steel: (0.37% C; 0.60% Cr; 1.30% Ni)	32	0.6	1.25	0.083	1.67	40	2.45	0.46
Chrome-nickel steel: (3.5% Ni, 0.8% Cr)	28	0.3	..	0.267	1.21	17	1.66	0.32
Do	28	0.3	..	0.062	2.17	54	2.67	0.70

* Numbers in column refer to footnotes in body of paper.
† Values of K from photoelastic investigations (Fig. 4).
H. T. = heat treated.

7—Alternating-Torsion Fatigue Tests

Tests have shown that the ratio of the endurance limits of ordinary specimens (no stress concentration) in alternating bending and alternating torsion is substantially the same in the case of fillet specimens.[22] These results seem to indicate that for certain cases the stress-concentration factors may be the same in bending and in torsion.

Tests have been made of model crankshafts in alternating torsion.[23] The effect of holes and fillets on the torsional endurance limit has also been investigated.[20]

8—Data and General Remarks

In addition to the data presented in Figs. 1–7, further fatigue stress-concentration factors are given in Tables 1–4. Additional references covering these data are given below.[24–34]

[22] Ludwik, P., "Kerb- und Korrosionsdauerfestigkeit," *Metallwirtschaft-Metallwissenschaft-Metalltechnik*, vol. 10 (1931), p. 708.

[23] Matthaes, K., "Kurbelwellenbrüche und Werkstoffragen," *Luftfahrtforschung*, vol. 8 (1930), p. 91.

[24] Moore and Kommers, "An Investigation of the Fatigue of Metals," Univ. of Ill. Exp. Sta. Bulletin No. 124, p. 105.

[25] Lehr, E., "Oberflächenempfindlichkeit und innere Arbeitsaufnahme der Werkstoffe bei Schwingungsbeanspruchung," *Zeit. f. Metallkunde*, vol. 20 (1928), p. 78.

[26] Moore and Jasper, "An Investigation of the Fatigue of Metals," Univ. of Ill. Exp. Sta. Bull. No. 152.

[27] Moore, Lyon, and Inglis, "Tests of the Fatigue Strength of Cast Iron," Univ. of Ill. Exp. Sta. Bulletin No. 164.

The notations used in Tables 1–4 are those discussed in Sec. 1 (Terminology). It will be noted that in all cases the stress-concentration index (q) is less than unity, i.e., fatigue stress-concentration factors are less than the corresponding theoretical stress-concentration factors.

It is clear that much work, practically of routine character, remains to be done before satisfactory systematic data for design purposes will be available. Further investigations must also be made of certain problems of fundamental character, as indicated by the indefiniteness of some of the conclusions of the foregoing sections.

II—SOME OBSERVATIONS CONCERNING FATIGUE FRACTURES

1—Effect of Stress Concentration on Mode of Failure

The examination of a large number of specimens broken in fatigue tests has led to some rather interesting observations concerning the appearance of the fracture. In ordinary fatigue specimens where stress-concentration effects are not present, the crack usually begins at one place and progresses across the speci-

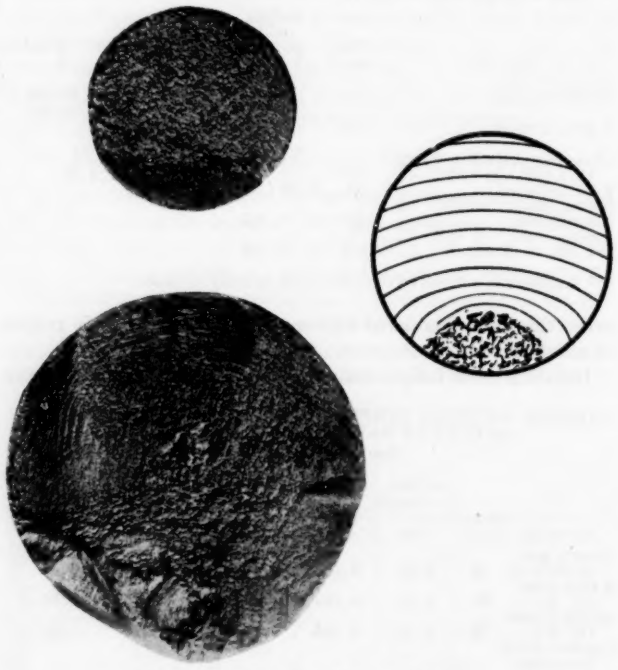
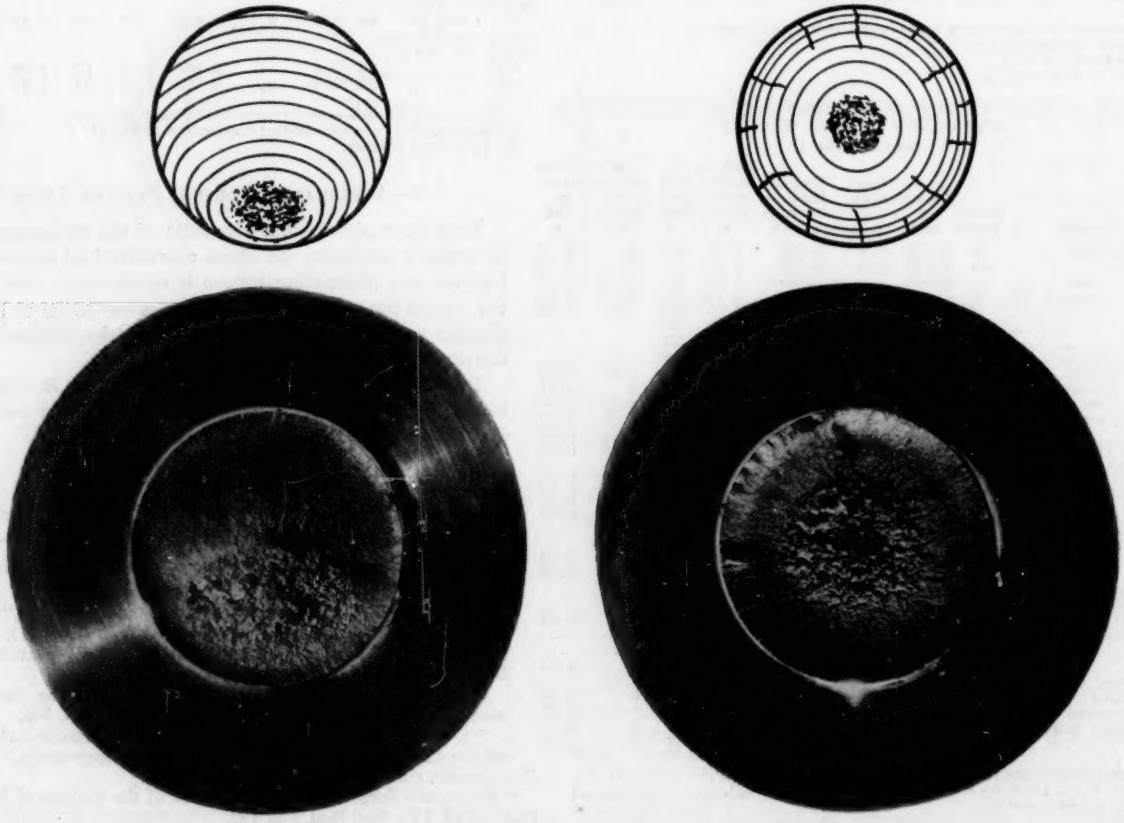


Fig. 9 Fracture—Uniform Stress



(a) Low Stress (b) High Stress

Fig. 10 Fractures—Stress Concentration

[28] Ludwik, P., "Ermüdung," First Communications of New Internat. Soc. Test. Matls., p. 131.

[29] Lea, F. C., "The Strength of Materials as Affected by Discontinuities and Surface Conditions," *Engineering* (London), vol. 134 (1932), p. 258.

[30] Kaufmann, E., "Über die Dauerbiegefestigkeit einiger Eisenwerkstoffe und ihre Beeinflussung durch Temperatur und Kerbwirkung," Springer (Berlin), 1931, p. 65.

[31] Inglis, C. E., "Stresses in a Plate Due to the Presence of Cracks and Sharp Corners," Trans. Inst. Naval Arch. (Brit.), 1913.

[32] Timoshenko, S., "Stress Concentration Produced by Fillets and Holes," Proc. 2nd Congress for Applied Mechanics, Zurich (1926).

[33] Timoshenko, S., and Lessells, J. M., "Applied Elasticity," W.T.N.S. Press, East Pittsburgh, Pa., p. 481.

[34] Moore, Lyon, and Alleman, "A Study of Fatigue Cracks in Car Axles," Univ. of Ill. Exp. Sta. Bulletin No. 197.

men, so that the area of final rupture has a segmental appearance (Fig. 9). If, however, a stress-concentration effect is present (such as, for instance, in a specimen having a small fillet or

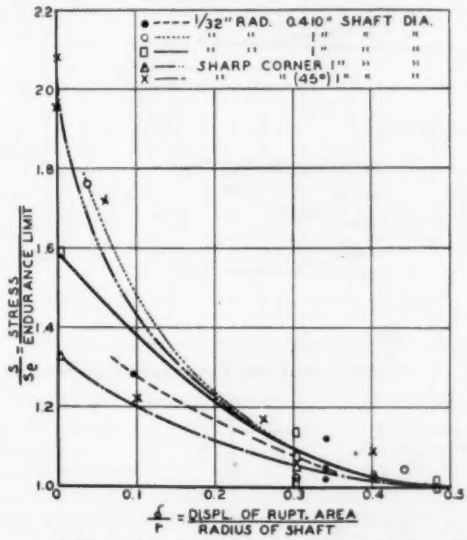


FIG. 11 SHIFT OF RUPTURE AREA AS A FUNCTION OF STRESS
(0.20 per cent C steel specimens.)

fatigue strength of individual grains might be such as represented schematically in Fig. 13(*b*). At a high stress the condition corresponding to (*S'*) could exist, and if stress concentration is present, as in Fig. 13(*a*), failure would be localized in one plane and would progress from all around the periphery and finally rupture with a concentric area, as in Fig. 10(*b*).

Initial failure at many points is evidenced by the presence of radial ridges, called "beach marks" by several fatigue investigators.[34] These radial ridges are formed by the initial cracked portions meeting each other as the test progresses. [Fig. 10(*b*).] Such markings are not likely to occur in specimens uniformly stressed (no stress concentration) since it is improbable that the weakest grains would be located in the same transverse section. "Beach marks" together with a concentric rupture area are generally characteristic of relatively high stress and stress concentration.

Even in the case of severe stress concentration (such as due to a small fillet), the different cracks do not all follow exactly the plane of highest stress, or there would be no steps as evidenced by "beach marks." Several possible explanations can be offered. First, if a cantilever specimen having a fillet be considered, the stress may not vary greatly over a portion of a fillet, since the increase in diameter (causing decrease of stress) can be partly offset by the increase in moment arm (causing increase of stress). Second, cracks might start in adjacent scratches and, in joining, cause beach marks to form.[4] Third, it has been observed that in

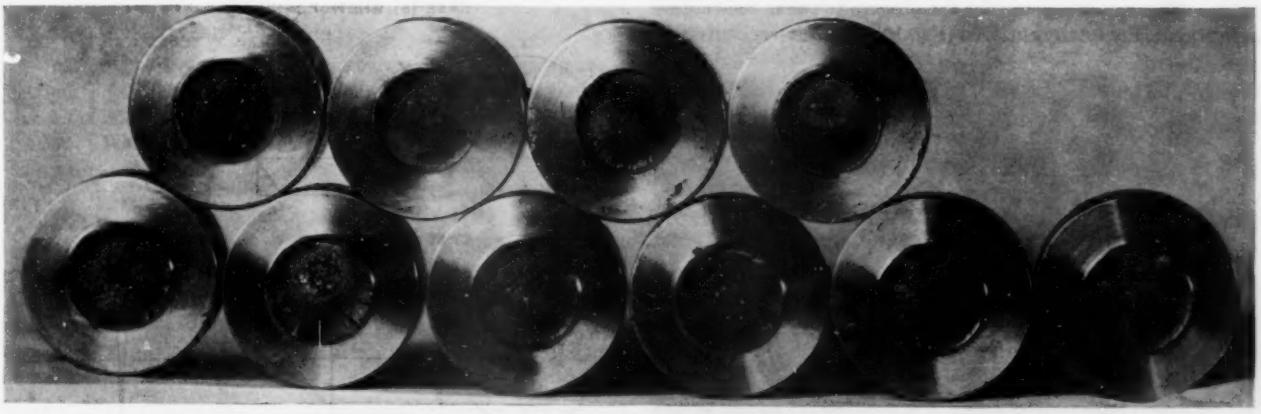
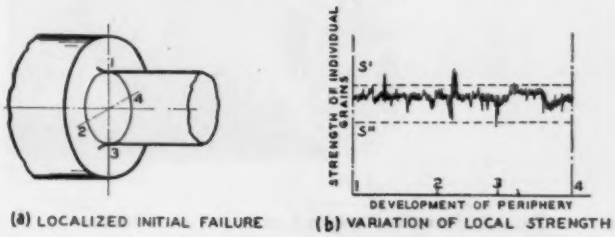


FIG. 12 SHIFT OF RUPTURE AREA
(0.20 per cent C steel.)
(UPPER ROW: Sharp-cornered specimens; stresses, respectively, 19,200, 15,200, 14,900, and 14,000 lb per sq in., reading from left.
LOWER ROW: Fillet radius 1/32 in.; stresses, respectively, 23,800, 17,100, 16,200, 15,100, and 14,900 lb per sq in., reading from left.)

groove), the rupture area will generally be totally enclosed and within the boundary of the cross-section (Fig. 10). The position of the rupture area within the cross-section seems in general to vary with the stress, being centrally located for high stresses and displaced from the center as the stresses are decreased toward the endurance limit.[35] In Fig. 11 are shown data for a low-carbon steel. The shift of the rupture area was also apparent in tests of copper and of chrome-nickel-steel specimens. A medium-carbon steel showed similar results, although not as clearly. A heat-treated carbon steel showed erratic results with respect to this type of measurement.

Fig. 12 shows a series of fractures of low-carbon-steel specimens (sharp corner and 1/32 in. radius). The displacement of the rupture area as related to the applied stress and the degree of stress concentration might be explained as follows: The variation in



(a) LOCALIZED INITIAL FAILURE (b) VARIATION OF LOCAL STRENGTH

FIG. 13

cantilever specimens cracks progress around the periphery in a helical manner. In Fig. 14, reproduced from a previous publica-

[35] A similar observation was made by Dr. J. A. Goff, of the University of Illinois, a number of years ago.

[36] After the manuscript of this paper had been prepared, Professor Bacon, of Swansea, England, sent the author a preprint copy of "Cracking and Fracture in Rotary Bending Tests" to be published in *Engineering* (London). Professor Bacon uses the term "ratchet-faced fracture" to denote what is here called a "beach-marked" surface.

tion,[37] cracks started at each of the holes and progressed helically until they joined midway between the holes, where the outer crack curved in abruptly to meet the inner crack. The helical movement evidently depends on the lateral curvature of the cantilever specimen, due to internal friction.[38] Since the direction of lateral curvature depends on the direction of rotation, the helix should change from right hand to left hand (using screw rotation) with a change of direction of rotation. Experiments have shown this to be the case.[39]

The foregoing remarks have been concerned with a case in which the stress is relatively high. At a lower stress, say, slightly above the "endurance strength of the specimen," a condition corresponding to S'' in Fig. 13(b) could exist, so that failure starts at one place and progresses slowly around the periphery and across the section simultaneously, resulting in a displaced rupture area. In this case the appearance of the displaced rupture area [Fig. 9 or 10(a)] depends on whether or not stress concentration due to form of specimen is present.

If stress concentration is present, the ratio of the rate of crack growth around the periphery as compared with the rate of crack penetration across the section will be higher than if stress concentration is not present, because of the difference in stress fields as shown in Fig. 15. The diagrams of constant-stress lines [Fig. 15(b, d)] represent the original stress fields (not the progress of the crack). It seems reasonable that a stress field such as Fig. 15(d) would be more conducive to relatively rapid crack growth around the periphery than would a stress field such as in Fig. 15(b). This would account for the observation that the rupture area is generally totally enclosed (Fig. 10) for stress-concentration specimens; and is of segmental form for specimens without stress concentration.

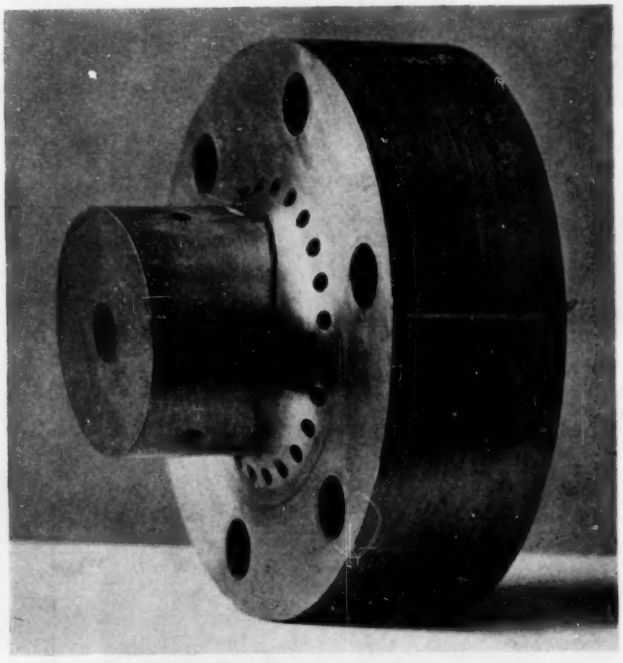


FIG. 14 HELICAL FATIGUE CRACK IN ROTOR MODEL

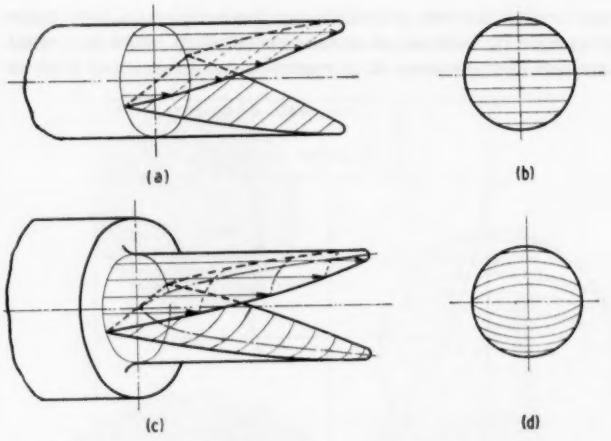


FIG. 15 STRESS FIELDS FOR ROTATING-BEAM SPECIMENS

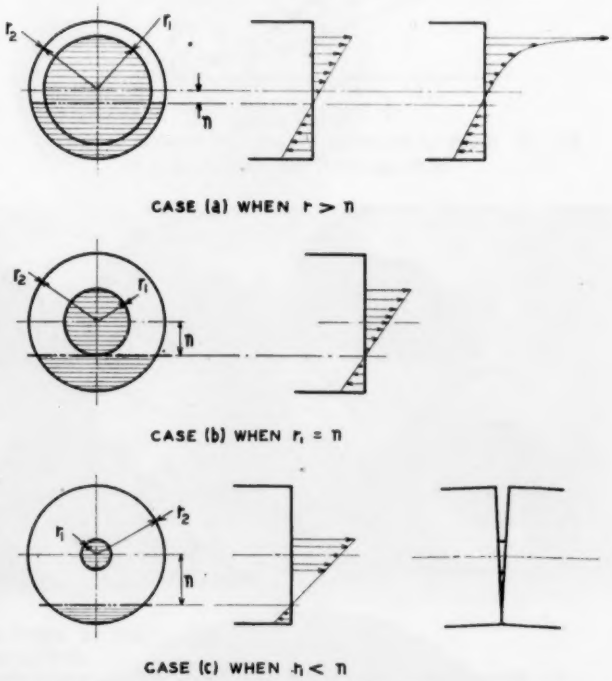


FIG. 16 STRESS DISTRIBUTION FOR CONCENTRICALLY CRACKED SPECIMENS

In connection with the latter type of failure mentioned in the foregoing, a mathematical analysis[40] has been made of the shape of the bottom of the crack at different stages of penetration. It was concluded that the bottom of the crack during penetration tended to conform to a series of elliptic curves.[41] The analysis should be carried further, and should include conditions at the rupture stage and also take account of the cracked area carrying compressive stresses.

[37] Peterson, R. E., "Fatigue Tests of Model-Turbo-Generator Rotors," *Mechanical Engineering*, vol. 53 (1931), p. 211.

[38] Mason, W., "Mechanics of the Wöhler Rotating Bar Fatigue Test," *Engineering* (London), 1923, p. 698.

[39] Professor Bacon (see footnote no. 36) states: "It appears to be a law that the arrowhead defining rotation meets the steep sides of the ratchet face." A check of our specimens confirms Professor Bacon's observations as far as filleted specimens are concerned, but for the specimens with holes, the direction was opposite in all cases.

[40] Bacon, F., "Fatigue Stresses With Special Reference to the Breakage of Rolls," Proc. So. Wales Inst. Engrs., vol. 47 (1931), no. 2, p. 141.

[41] In Professor Bacon's paper (see footnote no. 36) the progress of cracking is followed by ingenious methods. While the course of cracking is different from that indicated by the mathematical analysis[40] (see also Fig. 9), the above comments on the effect of stress concentration are apparently unaffected.

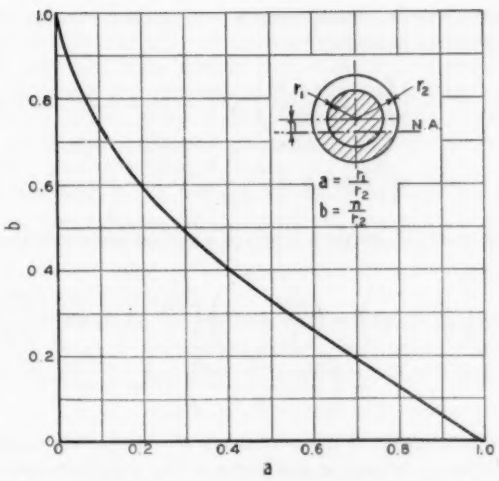


FIG. 17 POSITION OF NEUTRAL AXIS FOR CONCENTRICALLY CRACKED SPECIMEN

A study of fractures is not only of scientific interest, but of immediate practical value as well. In examining broken shafts and parts sent to us we have been able to reach a number of conclusions concerning the degree of stressing and the nature of failure.

2—NOTES ON CONCENTRIC CRACK PROPAGATION

As pointed out in the foregoing section, concentric cracking is produced by relatively high stress and stress concentration. In the following analysis of stress conditions in terms of crack penetration, it will be assumed that: (1) the stress distribution is linear (as in ordinary bending) during the progress of crack penetration. While this assumption is not correct due to stress concentration at the bottom of the crack, it is thought that the stress peak is quite narrow [Fig. 16(*a*)] so that the position of the neutral axis will not be changed greatly and the stress phenomena will be represented as essentially correct from a qualitative point of view; (2) that the cracked portion of the bent specimen carries compressive stresses, but not tensile stresses [Fig. 16(*a*)].

The position of the neutral axis is found as follows:

$$n = \frac{\int y\,dA}{\int dA}$$

$$n = \frac{\displaystyle\int_{-r_2}^{-n} y\sqrt{r_2^2 - y^2}\,dy + \int_{-n}^{0} y\sqrt{r_1^2 - y^2}\,dy + \int_{0}^{r_1} y\sqrt{r_1^2 - y^2}\,dy}{\displaystyle\int_{r_2}^{-n} \sqrt{r_2^2 - y^2}\,dy + \int_{-n}^{0} \sqrt{r_1^2 - y^2}\,dy + \int_{0}^{r_1} \sqrt{r_1^2 - y^2}\,dy}$$

$$n = \frac{-\frac{2}{3}\left[\sqrt{(r_2^2 - n^2)^3} - \sqrt{(r^2 - n^2)^3}\right]}{\left(n\sqrt{r_1^2 - n^2} - \sqrt{r_2^2 - n^2}\right) + r_1^2 \sin^{-1}\left(\frac{n}{r_1}\right) - r_2^2 \sin^{-1}\left(\frac{n}{r_2}\right) + \frac{\pi}{2}\left(r_1^2 + r_2^2\right)}$$

The solution for the position of the neutral axis can be expressed independently of size by taking $a = r_1/r_2$ and $b = n/r_2$, whence the above equation can be simplified to

$$(2a^2 + b^2)\sqrt{a^2 - b^2} - (b^2 + 2)\sqrt{1 - b^2} + 3b\left(a^2 \sin^{-1}\frac{b}{a} - \sin^{-1} b + \frac{\pi}{2}a^2 + \frac{\pi}{2}\right) = 0 \quad \ldots\ldots [1]$$

By using a trial-and-error method and plotting results graphically, the position of the neutral axis can be found to three decimal places in a comparatively short time. The moment of inertia is found as follows:

$$I = \int y^2 dA$$

$$I = 2\int_n^{r_2} (y-n)^2\sqrt{r_2^2 - y^2}\,dy - \int_n^{r_1} (y-n)^2\sqrt{r_1^2 - y^2}\,dy + \frac{\pi r_1^4}{8} - \frac{\pi r_1^2}{2}\left(\frac{4r_1}{3\pi}\right)^2 + \frac{\pi r_1^2}{2}\left(n - \frac{4r_1}{3\pi}\right)^2 + \frac{\pi r_1^4}{8} - \frac{\pi r_1^2}{2}\left(\frac{4r_1}{3\pi}\right)^2 + \frac{\pi r_1^2}{2}\left(n + \frac{4r_1}{3\pi}\right)^2$$

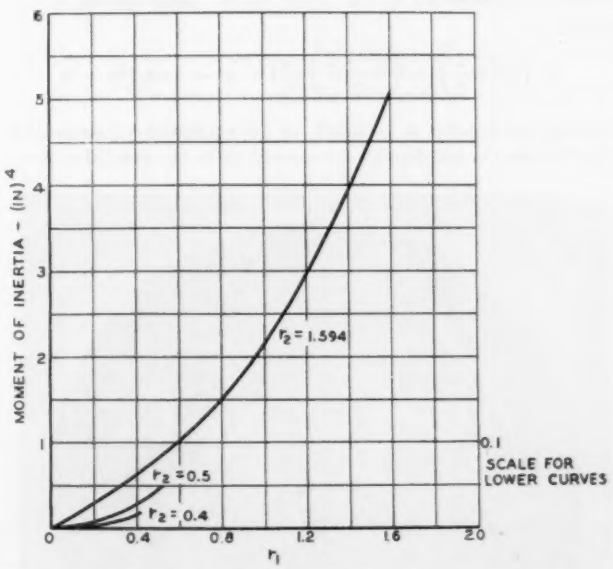


FIG. 18 MOMENTS OF INERTIA OF CONCENTRICALLY CRACKED SPECIMENS

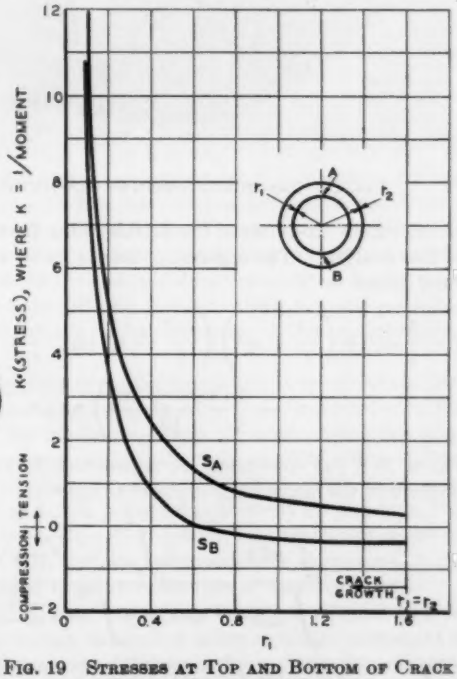


FIG. 19 STRESSES AT TOP AND BOTTOM OF CRACK DURING GROWTH OF CRACK
(For r_2 = 1.6 in.)

$$I = \left(\frac{r_1^2}{4} + n^2\right)\left(\frac{\pi r_1^2}{2} + r_1^2 \sin^{-1}\frac{n}{r_1}\right) + \left(\frac{r_2^2}{4} + n^2\right)\left(\frac{\pi r_2^2}{2} - r_2^2 \sin^{-1}\frac{n}{r_2}\right) + \left(\frac{13nr_1^2 + 2n^3}{12}\right)\sqrt{r_1^2 - n^2} - \left(\frac{13nr_2^2 + 2n^3}{12}\right)\sqrt{r_2^2 - n^2} \quad [2]$$

The foregoing equations are valid for $n \leqq r_1$. For $n = r_1$ the condition represented in Fig. 16(*b*) exists. Substituting $b = a$ in [1],

$$\pi a\left(a^2 + \frac{1}{2}\right) - \left(\frac{a^2 + 2}{3}\right)\sqrt{1 - a^2} - a \sin^{-1} a = 0$$

Solving graphically, $a = 0.397$, or approximately, four-tenths. The "critical crack depth" corresponding to the condition mentioned will be six-tenths of the outside radius for the assumptions of this analysis. The moment of inertia for $n = r_1$ becomes by substitution in [2]



FIG. 20 FRACTURED SPECIMEN (ARTIFICIAL CRACK)

$$I = \frac{5}{4}\pi r_1^4 + \left(\frac{r_2^2}{4} + r_1^2\right)\left(\frac{\pi r_2^2}{2} - r_2^2 \sin^{-1}\frac{r_1}{r_2}\right) - \left(\frac{13r_1r_2^2 + 2r_1^3}{12}\right)\sqrt{r_2^2 - r_1^2} \quad [3]$$

For $n > r_1$ the conditions are represented in Fig. 16(*c*). The position of the neutral axis is found as follows:

$$n = \frac{\int y\,dA}{\int dA} = \frac{2\int_n^{r_2} y\sqrt{r_2^2 - y^2}\,dy}{\pi r_1^2 + 2\int_n^{r_2}\sqrt{r_2^2 - y^2}\,dy}$$

$$\pi n\left(r_1^2 + \frac{r_2^2}{2}\right) - \left(\frac{2r_2^2 + n^2}{3}\right)\sqrt{r_2^2 - n^2} - nr_2^2 \sin^{-1}\frac{n}{r_2} = 0$$

Dividing by r_2^2 and substituting $b = n/r_2$ and $a = r_1/r_2$ a dimensionless equation results:

$$\pi b\left(a^2 + \frac{1}{2}\right) - \left(\frac{2 + b^2}{3}\right)\sqrt{1 - b^2} - b \sin^{-1} b = 0$$

$$a = \sqrt{\frac{1}{\pi}\sin^{-1} b + \frac{2 + b^2}{3\pi b}\sqrt{1 - b^2} - \frac{1}{2}} \quad [4]$$

The moment of inertia is found in a manner similar to the foregoing:

$$I = \pi r_2^4\left(\frac{r_1^2}{4} + n^2\right) + \left(\frac{r_2^2}{4} + n^2\right)\left(\frac{\pi r_2^2}{2} - r_2 \sin^{-1}\frac{n}{r_2}\right) - \left(\frac{13nr_2^2 + 2n^3}{12}\right)\sqrt{r_2^2 - n^2} \quad [5]$$

The displacement of the neutral axis is given by b in terms of a (independently of size) in the curve of Fig. 17 drawn according to Equations [1] and [4]. The moment of inertia is not independent of size and requires a separate curve for each outside diameter as in Fig. 18. Equations similar to the foregoing have also been derived for displaced fracture areas, but these equations are not essentially different and are omitted because of space limitations.

If stress concentration is neglected, the stress variation during crack penetration can be plotted as shown in Fig. 19.

$$S_A = \frac{M}{I}(r_1 + n) \quad [6a]$$

$$S_B = \frac{M}{I}(r_1 - n) \quad [6b]$$

When the crack has penetrated to the "critical crack depth" $0.6r_2$ (for the assumptions of this analysis), the condition $r_1 = n$ exists, so that $S_B = 0$. For greater crack penetration the central zone becomes entirely in tension.

One might expect the above condition to reveal itself in the surface markings; before the "critical crack depth" is reached the crack opens and closes once per revolution, and in the closed position carries compressive stresses, the entire action resulting in the usual smooth, battered, and somewhat corroded surface; below the "critical crack depth" the bottom of the crack is not subjected to compression at any time [Fig. 16(*c*)] and consequently should not show a battered appearance. A number of fractured specimens were examined, and in some cases three zones were evident. (Fig. 20.) Although the "critical crack depth" was generally indistinct, measurements of a few fractures did not show it to be 0.6 r_2 in accordance with the foregoing analysis.[42]

However, if the analysis offers a qualitative explanation, it may be considered to be of some value.

The stresses acting on the central zone (Fig. 19) can also be resolved into a variable stress S_v and a steady stress S_0 as in Fig. 21. At the outset the stress is all S_v, but at the time of failure is largely S_0.

It was thought that it would be possible to calculate the size of the fracture area by using this method,[43] but computations

[42] Professor Bacon (see footnote no. 36) has found by means of an etching procedure that the crack does not close as completely as assumed in the foregoing analysis, due to slight plastic flow at the crack bottom. This would cause the "critical crack depth" to be less than that found by analysis, a condition borne out by observations.

[43] The above-mentioned discovery by Professor Bacon would also explain the lack of quantitative check in the case of size of rupture area.

did not generally agree with measurements. Data are needed on the tensile strength of specimens having circumferential cracks, also data from bending tests. The procedure for calculating the size of the fracture area does not seem unreasonable.

In order to test a severe case of stress concentration, specimens were made with artificially produced circumferential cracks.[44] A very narrow V-bottomed groove was turned in a bar, which latter was heated and compressed axially to close the groove. The bar was subsequently annealed and machined into a test piece having a circumferential crack. The material used was a 0.44 per cent C steel.[17] The appearance of a fractured specimen is shown by Fig. 20.

The results of the fatigue tests of the cracked specimens are given in Table 5, the computations being made according to Equations [1], [2], and [6a]. The stress-concentration factors

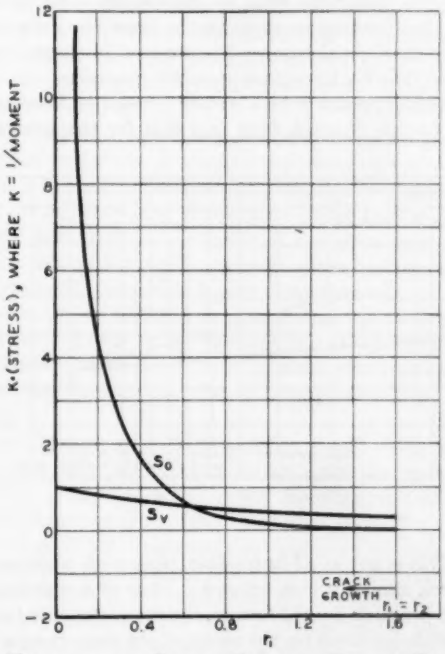


FIG. 21 VARIABLE AND STEADY STRESSES DURING GROWTH OF CRACK
[For r_2 = 1.6 in. (See Fig. 19.)]

are found by dividing 31,700 (which is the endurance limit without stress concentration) by the crack-endurance limits (Table 5). The stress-concentration factors as a function of size are shown in Fig. 8.

TABLE 5 FATIGUE OF SPECIMENS HAVING ARTIFICIAL CRACKS—SUMMARY OF DATA

	Endurance limits* lb per sq in.	Stress-concentration factor, k
Without crack	31,700	..
r_2 = 1.60; r_1 = 1.18	7,900	4.00
r_2 = 0.40; r_1 = 0.297	11,200	2.83
r_2 = 0.40; r_1 = 0.160	4,700	6.75

* Based on Equations [1], [2], and [6a].

ACKNOWLEDGMENTS

The author is indebted to Mr. F. Cassel for considerable assistance in the collection of data for Part II; and to Dr. S. Timoshenko and Messrs. J. M. Lessells and I. W. Chubb for their interest and support of research on mechanics of materials.

[44] The method was suggested by Mr. P. G. McVetty, of the Westinghouse Research Laboratories.

Discussion

JOHN A. GOFF.[45] In section 2 of the paper there are certain observations regarding fatigue fractures, and, in particular, the author points out that when a stress-concentration effect such as that produced by a small fillet or groove is present in the specimen being tested, the final rupture area is generally totally enclosed within the boundary of the cross-section at which failure occurs. Moreover, the author states that the location of this final rupture area seems to vary with the nominal stress imposed upon the specimen and that it appears to be centrally located for high nominal stresses, but displaced toward the circumference for stresses approaching the endurance limit for the given stress-concentration effect.

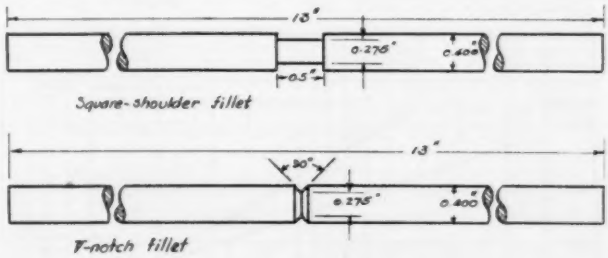
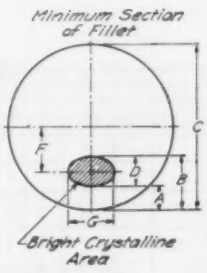


FIG. 22



SQUARE SHOULDER FILLET

Specimen Number	Nominal Stress	Dimensions in mm. D	G	F
Upper Curve				
C39B	46,010	3.25	9.4	0.58
A195B	33,700	2.05	2.8	1.03
A39C	29,300	1.80	2.6	1.20
C91C	25,400	1.80	2.7	1.85
C177A	25,313	1.50	2.2	1.65
D08	24,270	1.50	3.8	1.65
D91B	23,990	2.30	3.4	1.75
D182A	23,990	1.60	2.2	1.68
Lower Curve				
C91B	40,160	2.55	2.55	0.23
G195A	34,040	2.30	2.30	0.40
D182B	24,600	1.60	1.60	0.90
A39A	24,970	1.60	1.70	0.98
G130D	23,800	2.50	2.50	1.05

FIG. 23

In a footnote there is a reference to some unpublished data obtained by the writer at the University of Illinois in 1921. At that time the writer was working under the direction of Profs. H. F. Moore and J. B. Kommers to determine the effect of fillets on the resistance of steel to repeated stress by reversed bending as a part of the work of the joint investigation of fatigue of metals. The principal results on lowering of endurance limit due to stress concentration by the different fillets studied have been published in Bulletins 124 and 142 of the University of Illinois Experiment Station.

Early in the investigation just mentioned, it was observed that two of the fillets studied exhibited the features described in the paper; that is, the final rupture area (or bright crystalline area) was totally enclosed within the cross-section where failure occurred. These fillets are referred to as the V-notch and square-shoulder fillets, and their principal dimensions are shown in Fig. 22. The steel was 0.49-carbon sorbitic, heat treated as described in Bulletin No. 124, and the specimens were those used at the time in the Farmer rotating-beam type of machine.

In almost every case the rupture area was distinct and its boundary sharp enough to admit of rather careful measurement of its principal dimensions. Consequently a little jig was con-

[45] University of Illinois, Urbana, Ill. Assoc-Mem. A.S.M.E.

structed for holding the fractured specimen in front of a Brinell hardness telescope, and in this way dimensions A, B, C, and G (Fig. 23) were easily read to one-tenth millimeter.

After miscellaneous attempts to correlate the measurements obtained in this manner with other data such as number of repetitions of stress etc., the writer finally hit upon the idea of plotting on logarithmic coordinate paper the displacement of the center of the rupture area from the center of the specimen (dimension F, Fig. 23) against the nominal stress S, and in this way obtained the representation of Figs. 24 and 25.

One of the most striking and unusual features presented by Figs. 24 and 25 is the fact that to each fillet belongs a pair of distinct straight lines. For the square-shoulder fillet, for example, this means that a specimen loaded with a nominal stress of

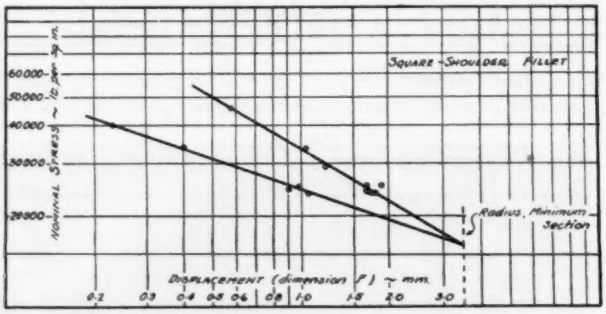


Fig. 24

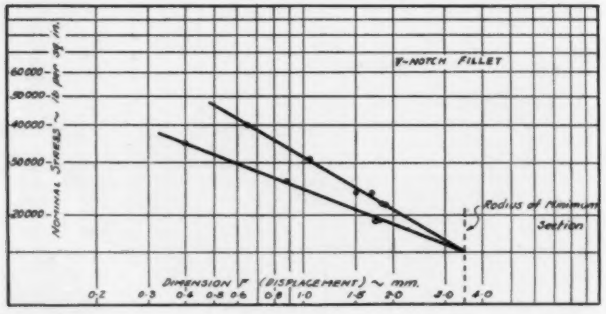


Fig. 25

40,000 lb per sq in. will break in either one of two possible ways; that is, the center of its final-rupture area will be displaced either 0.23 mm or 0.74 mm from the center of the cross-section.

Again, it is quite conspicuous how, for either fillet, the two straight lines intersect at a common value of F very near to 3.5 mm; but 3.5 mm is almost exactly the radius of the minimum cross-section of the fillet. Hence such a displacement, if possible, would place the center of the rupture area right on the circumference of the minimum cross-section.

From the distribution of test points on the pair of straight lines for either fillet, it would appear that neither mode of failure is more probable than the other. The grouping of points near the endurance-limit stress is explained by the fact that a relatively large proportion of specimens broken were loaded with nominal stresses in this region, the primary object of the experiments being to determine the endurance limit for each fillet as accurately as possible.

The data present in Figs. 24 and 25 confirm the author's observation that the location of the rupture area seems in a general way to depend upon the nominal stress; but at the same time they indicate a kind and order of dependence which might never be suspected from an examination of his Fig. 11. The way in which the test points fall along the straight lines of Figs. 24 and 25 would seem to argue that the mechanism of fatigue failure in a specimen having sufficient stress-concentration effect to insure that the final rupture area is totally enclosed within the cross-section is not such a random phenomenon as it is sometimes imagined to be.

Yet another interesting feature of the results is brought to light by an examination of the measurements for the square-shoulder fillet, Fig. 23. In the table of Fig. 23, the specimens are grouped according as their displacements F and nominal stresses S place them on the upper or on the lower straight line. Comparing dimensions D and G, then, shows very clearly that the fractures on the lower line are nearly circular, while those on the upper line are decidedly oval in shape. The measurements for the V-notch specimens seem to exhibit the same peculiarity—namely, that the rupture areas on the lower line are nearly circular, while those on the upper line are oval in shape, but less decidedly so than for the square-shoulder specimens.

There also appears to be a certain correlation between the pair of lines for the V-notch fillet and that for the square-shoulder

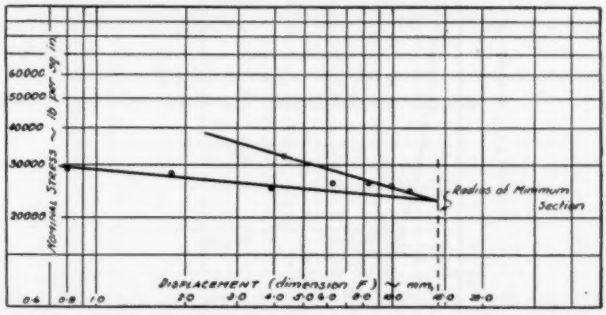


Fig. 26

fillet. This is not at all impossible, since both were made in the same steel, similarly heat treated. Thus at a nominal stress of 24,720 lb per sq in. for the former, the radial distance between the two possible locations for the center of the rupture area is greater than that at any other stress. For the square-shoulder specimen this greatest radial distance is found at almost exactly the same nominal stress—namely, 24,850 lb per sq in.

Corroborating these results described are similar measurements furnished the writer in 1925 by Mr. J. Muller, who at that time was engaged with the joint investigation of fatigue of metals. At the writer's request, Mr. Muller kindly measured the rupture areas of larger cantilever-type specimens which were being tested for endurance limit. These specimens were made with a square-shoulder fillet whose small diameter was in every case $1^1/_8$ in. (28.6 mm). This shoulder determined the critical section and introduced sufficient stress-concentration effect to insure that the rupture areas were totally enclosed within the critical cross-sections.

The results of measurements on the cantilever-type specimens are shown in Fig. 26, which again exhibits the characteristic pair of straight lines on logarithmic coordinate paper, intersecting at a common value of F very nearly equal to 14.3 mm—just half the diameter of the minimum cross-section. A check on the principal diameters of the rupture areas, however, fails to yield the interesting conclusions regarding shape of rupture area which obtain for the V-notch and square-shoulder fillets as described.

The writer offers the foregoing data without any attempt at physical explanation. It would be gratifying to him, however, to have these data attract the attention of those actively inter-

ested in the study of fatigue of metals to the extent possibly of influencing their views on the mechanism of fatigue failure. It seems not unreasonable to suppose that the ultimate theory of fatigue must include an explanation of the results presented herewith.

R. Eksergian.[46] The author's tests and interpretation on stress concentration as affecting fatigue are very interesting, and his introduction of a "fatigue stress concentration factor," with his discussion of its variation as a function of different physical properties of materials and for fillet sizes, is an important contribution.

It has been known that stress-concentration effects on endurance are lower than would be expected from the estimating of endurance from calculated or photoelastic data, and the tests of Haigh and Wilson for holes in flat plates indicate a marked reduction with mild steel plates. The comparison of grooved specimens with ordinary fatigue specimens shown in the author's Fig. 1 also shows that with mild steels the effect of stress concentration is greatly reduced.

This opens up a very interesting question, inasmuch as on the one hand ductility appears to play an important role in reducing the effect of dimensional stress concentrations on endurance and on the other hand it has been consistently shown that endurance limits appear essentially only a factor of the ultimate strength. Furthermore, we know that the range of stress becomes greatly reduced when oscillating about higher mean stresses.

It appears to the writer that some explanation can be offered by considering the ultimate stress concentration to be divided into two components: (1) an intermolecular stress concentration and (2) a dimensional stress concentration arising from fillets, holes, notches, etc.

Both would be expected to be reduced by ductility due to local plastic flow in a redistribution and spreading of the local stress concentration. With the more brittle materials the effect of stress concentration becomes more emphatic.

If now we regard the endurance limit governed by stress concentration, the relative ductility, and the yield point rather than the ultimate strength, it would be necessary to explain the effect of these factors in maintaining with a simple fatigue specimen the apparent constant percentage on the ultimate strength where ordinary steels of various carbon contents are used. Raising the carbon contents raises the tensile strength, increases the ratio of yield to ultimate, and reduces the ductility. On the other hand, the molecular stress concentration would increase with decreasing ductility. With mild steels, the molecular stress concentration due to the plastic-flow spreading action would be small, so that the endurance should approximate the yield value. With decreasing ductility the molecular-stress-concentration factor would be anticipated to increase, lowering the endurance with respect to the yield value. But since the ratio of the yield to ultimate strength increases with decreasing ductility, the ratio of the endurance with respect to the ultimate would tend to maintain a constant ratio. On this basis, therefore, we would anticipate that by the use of an alloy steel, by raising the ratio of the yield to ultimate but maintaining very good ductility, the endurance with respect to the ultimate strength would be raised. It is further likely with mild steels having good ductility, with the beginning of the spreading action of the local stress concentration; likewise local cold working takes place during the first few cycles, raising the yield and resulting in understressing.

With dimensional stress concentrations the effect of ductility in reducing the importance of stress concentration would likewise be anticipated. With increasing tensile values shown in Fig. 1, the effect of dimensional stress concentration would be more pronounced, and with very brittle materials an approach to the photoelastic values would be anticipated. This condition is indicated in the author's Fig. 4 for Si steel. The discrepancy of the endurance based on photoelastic or calculated fillet stresses appears to show that endurance limits are really dependent on the peak stress going into the yield with some spreading action even with small ductility.

The writer recently had some experience with failures of hypercompressor crankpins occurring at the fillet. The pin and web assimilated more of a curved-beam characteristic than the symmetrical construction shown in Fig. 4. For this reason the stress concentration resulting from the fillet joining the web was calculated, and a comparison was made with stress-concentration factors obtained by Timoshenko and Dietz on a symmetrical construction shown in Fig. 4. This comparison is shown in the accompanying Fig. 27. Some weight was given to the greater tendency toward the curved-beam action, but at best the stress was far below the ideal endurance values. The fillets were located at the center core of the forging, and the forging was of massive dimensions. Proper working of the forging was impossible in the vicinity of the fillet. For this reason both the local ductility and grain structure were likely poor. The molecular stress concentration, accentuated by the dimensional stress concentration, or in other words the writer's "fatigue-stress-concentration factor," was of high order, and with possible internal stress superimposed could account for the failure. This is characteristic of the mass feature of a forging.

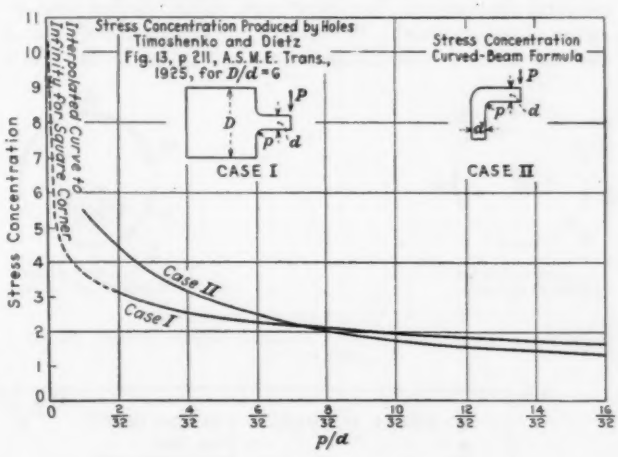


Fig. 27

In locomotive axles failure usually occurs at the fillet connecting the axle with the hub fit which is stepped up in the wheel center. With axles in the order of 12 to 14 in., fillets with $^1/_4$-in. radius and even up to $^1/_2$ in. are likely to fail. Such failures occur mostly on the main axle where the maximum loading is due to piston thrusts. The author has shown in the very interesting test, Fig. 8, that the stress-concentration factor, with a severe stress concentration, increases with the size of specimen. It would be anticipated, therefore, that the stress concentration factor shown in Fig. 4 for fillets would increase considerably with massive forgings as in axles. Therefore, until tests on large forgings have been made, the data shown in Fig. 4 should not be used for interpreting the effect of fillet stresses on endurance for large forgings.

The effect of mass size and fatigue with stress concentrations is a very important matter, and aside from the cost and difficulties for carrying on such tests, the matter is greatly tied up with the large variations in the conditions of forgings. Cut-out test

[46] Consultant, Engineering Department, E. I. du Pont de Nemours & Co., Wilmington, Del. Mem. A.S.M.E.

specimens are entirely misleading and in no way indicate the true conditions in a forging itself. But the important point is that the data supplied by the author for estimating stress concentrations are hardly applicable for assimilating the conditions in a massive forging. Statistical experience indicates that sharp fillet ratios are much more severe, resulting in higher stress concentration factors with large forgings than are indicated by such results as shown by the author's Fig. 4.

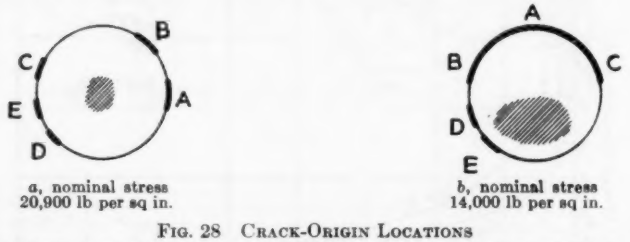
The writer would suggest that a cooperative test program under the supervision of a committee sponsored by the A.S.M.E. be established for planning some suitable tests for large forgings with typical fillet conditions. The survey plan would first gather as much statistical data as possible before planning a series of tests and the cost for them.

Author's Closure

Dr. Goff has found that for a series of fatigue tests of "stress-concentration specimens" two modes of failure may occur, one resulting in a round rupture area and the other resulting in an elliptical rupture area, the latter being shifted farther from the center of the specimen. A check of our specimens does not reveal the striking relationships shown in Figs. 24 to 26. However, some of the observations mentioned in the paper in connection with "stress-concentration" specimens are not incompatible with Dr. Goff's data. The idea was put forth that, at high stresses, cracking started from numerous sources around the periphery and progressed concentrically to form a centrally located circular rupture area. This is illustrated by the following log of a test run of a $^1/_{32}$-in. fillet, 1-in.-in-diameter specimen at relatively high stress (see Fig. 28*a*):

43,000 cycles, very tiny crack at *A*
48,000 cycles, *A*, 0.19 in., cracks appear at *B* and *C*
50,000 cycles, *A*, 0.19 in.; *B*, 0.18 in.; *C*, 0.05 in.; crack at *D*
52,000 cycles, same
54,000 cycles, *A*, 0.19 in.; *B*, 0.40 in.; *C*, 0.05 in.; *D*, 0.08 in.
56,000 cycles, *A*, 0.31 in.; *B*, 0.40 in.; *C*, 0.23 in.; *D*, 0.08 in.
58,000 cycles, *A*, 0.31 in.; *B*, 0.40 in.; *C*, 0.23 in.; *D*, 0.08 in.; *E*, crack appeared
60,000 cycles, *A*, 0.31 in.; *B*, 0.40 in.; *C*, 0.23 in.; *D*, 0.08 in.; *E*, crack appeared
64,000 cycles, *D* and *E* merged, very fine hairlike crack from *A* to *D*
68,000 cycles, same
78,000 cycles, *B* and *C* merged, *C* and *E* merged
80,000 cycles, *A* and *B* merged
398,000 cycles, failed

It was further proposed that at relatively low stresses (just above the endurance limit), cracking starts at one place, corresponding to the weakest point on the periphery (see Fig. 13), and then develops in a manner, described in the paper, so as to result in a displaced elliptical rupture area. This type is illustrated by the following log of a test run of a specimen similar to that above, but at relatively low stress (Fig. 28*b*).



a, nominal stress 20,900 lb per sq in. *b*, nominal stress 14,000 lb per sq in.

Fig. 28 Crack-Origin Locations
(0.20C steel; $^1/_{32}$-in. fillet; 1 in. in diameter; endurance limit, 13,700 lb per sq in.)

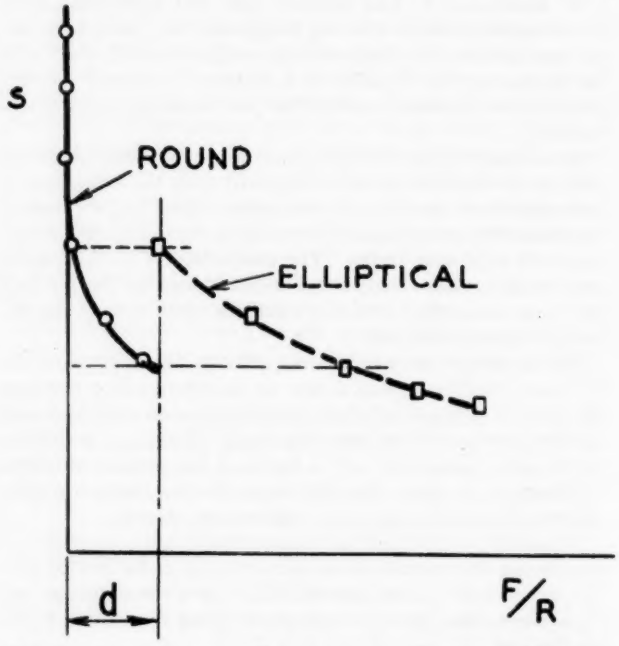


Fig. 30 Illustration of Explanation of Different Modes of Failure

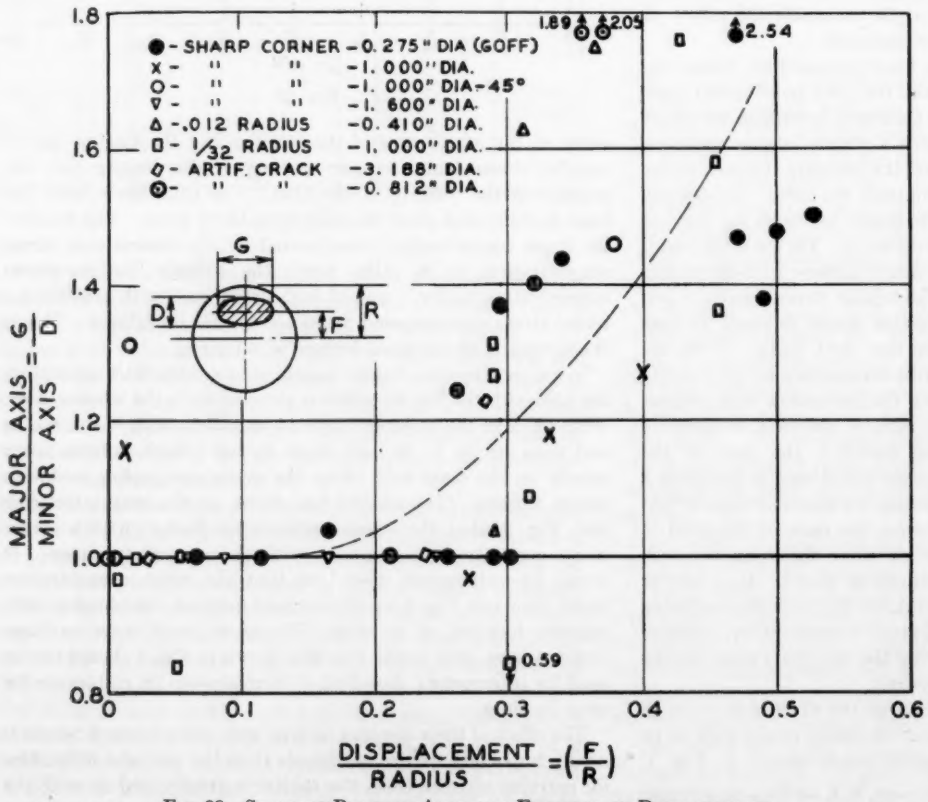


Fig. 29 Shape of Rupture Area as a Function of Displacement

2,871,000 cycles, crack 0.5 in. long discovered at *A*
3,094,000 cycles, crack *A*, same; approx. 0.2 in. from *A*, crack *B*, 0.15 in. long
3,183,000 cycles, *A* and *B* same; approx. 0.15 in. from *A*, crack *C*, 0.1 in. long
3,256,000 cycles, *A*, 0.55 in.; *B*, 0.27 in.; *C*, 0.20 in.
4,005,000 cycles, *A*, *B*, *C* merged, 1.3 in. long
6,459,000 cycles, *A*, *B*, *C* 1.67 in. long; approx. 0.15 in. from *A*, *B*, *C*, crack *D*, 0.17 in. long; approx. 0.1 in. from *D*, crack *E*, 0.1 in. long
7,094,000 cycles, failed

It is suggested that between the above-mentioned extremes a transition range may occur where for any given stress failure may be either concentric or eccentric, depending on the "point-by-point" strength distribution around the periphery (Fig. 13) for the individual specimens. That the elliptical rupture areas are farther displaced than the round areas is shown in Fig. 29, which also includes Dr. Goff's data. One might expect as a consequence of the foregoing remarks, some such diagram as shown in Fig. 30. Then at a displacement d one might have either round or elliptical areas. Perhaps the region near the vertical line at $0.3\ F/R$ in Fig. 29 represents such a condition. Or it may be that the transition from round to elliptical rupture areas is more gradual, following a general trend indicated by the dotted curve of Fig. 29. The wide scattering of points makes it difficult to draw conclusions without a large number of tests.

It appears to the author that much light can be thrown on the foregoing questions concerning the mode of failure by carefully recording crack progress and also simultaneously recording crack penetration by periodic insertion of some type of coloring medium or by some other suitable method of attaining the same end.

Mr. Eksergian has contributed some interesting remarks, particularly with regard to locomotive axles. The author agrees that direct data on the fatigue strength of fillet specimens of 12 to 14 in. diameter are not available. The largest specimens of this type which have been tested are $2^1/_8$ in. in diameter. Considerable extrapolation of uncertain character must be done in order to estimate what happens for large sizes. Mr. Eksergian apparently realizes the difficulty in making tests on large specimens; his suggestion of a cooperative test program under the supervision of a committee sponsored by the A.S.M.E. seems like a worth-while move.

APM-55-20

Working Stresses for Columns and Thin-Walled Structures

By S. TIMOSHENKO,[1] ANN ARBOR, MICH.

In metallic structures such as airplanes, airships, ships, bridges, etc. slender bars, thin webs, and thin-walled tubular members are very often used. In choosing working stresses for such structural elements, not only the mechanical properties of the material but also the elastic stability of these elements should be considered. The method of choosing working stresses for such structures is illustrated in this paper by several examples.



1—Introduction

IN METALLIC structures such as airplanes, airships, ships, bridges, etc. we very often use slender bars, thin webs, and thin-walled tubular members. In choosing working stresses for such structural elements, not only the mechanical properties of the material but also the elastic stability of these elements should be considered. Take, for instance, a centrally compressed column (Fig. 1). In this case the failure may occur at a comparatively low direct stress, and the magnitude of this stress then depends not on the strength of the material, but only on its modulus of elasticity and on the slenderness of the column. The failure occurs not as a result of crushing of the compressed material, but as a result of lateral buckling of the column. We may have analogous conditions in the case of thin plates subjected to the action of compressive or shearing forces in their middle plane, and, in general, in the case of thin-walled structures. The methods of choosing working stresses for such structures will now be illustrated by several examples.

2—Prismatic Columns

Let P = compressive force
E = modulus of elasticity
σ = normal stress
$\sigma_{y.p.}$ = yield-point stress
l = length of the column
r = least radius of gyration of the cross-section
A = cross-sectional area
I = smallest moment of inertia of the cross-section
S = section modulus
$m = S/A$
$k = \sqrt{(P/EI)}$
n = factor of safety
h = thickness of the plate
τ = shearing stress
μ = Poisson's ratio
R = radius of a cylindrical tube.

In the case of a slender column, the *critical* value of the centrally applied compressive force P_{cr} at which lateral buckling occurs is given by Euler's well-known formula:

$$P_{cr} = \frac{\pi^2 EI}{l^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

The corresponding critical compressive stress is

$$\sigma_{cr} = \frac{P_{cr}}{A} = \pi^2 E \left(\frac{r}{l}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots [2]$$

in which r is the least radius of gyration of the cross-section and l the length of the column. The ratio l/r is usually called the "slenderness ratio" of the column.

In Fig. 2 the curve AB represents the critical stress σ_{cr} as a function of the slenderness ratio calculated from Equation [2] for steel ($E = 30.10^6$ lb per sq in.).

The derivation of Equation [1] is based on Hooke's law, and the curve AB can be used only if σ_{cr} does not exceed the proportional limit of the material. Taking, for instance, in our example a proportional limit equal to 30,000 lb per sq in., we find that Euler's formula can be used only if the slenderness ratio l/r is not less than 100.

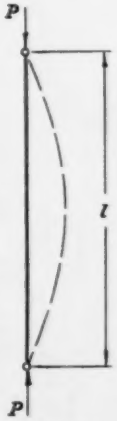


FIG. 1

Further investigations have shown that the application of Equation [2] can be extended also to shorter columns, in which σ_{cr} is above the proportional limit of the material, by substituting, instead of the constant modulus E, a reduced modulus E_1 varying with the slenderness ratio of the column. This reduced modulus can be determined from the compression stress-strain diagram of the material. Investigations made with structural steel show that for comparatively short columns ($l/r < 60$) the critical stress σ_{cr} can be taken equal to the yield point of the material, and for intermediate slenderness ($60 < l/r < 100$) it can be assumed that σ_{cr} varies from the yield point to the proportional limit following a linear law. Taking, for instance, a yield point of the material equal to 40,000 lb per sq in. and the proportional limit equal to 30,000 lb per sq in., we obtain in Fig. 2 the line DCB, which, together with Euler's curve, determines the values of σ_{cr} for any value of the slenderness ratio of a column.

In discussing working stresses for a column we must consider the critical stress given by the diagram $ABCD$ (Fig. 2) in the same way as ultimate strength in the case of simple tension. A compressive stress equal to the critical stress brings the column to failure, and the working stress must be much less than this stress. The magnitude of the factor of safety depends on the

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Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

possible errors in the magnitude of the load P (Fig. 1), and also on the possible errors in the central application of the load and on possible initial crookedness. The simplest assumption is to take a certain constant factor of safety. Taking, for instance, a factor of safety of 2.5 and using the critical stresses from Fig. 2, we obtain the diagrams $ABCD$ in Figs. 3(*a*) and 3(*b*) which give working stresses for columns of various slenderness ratios.

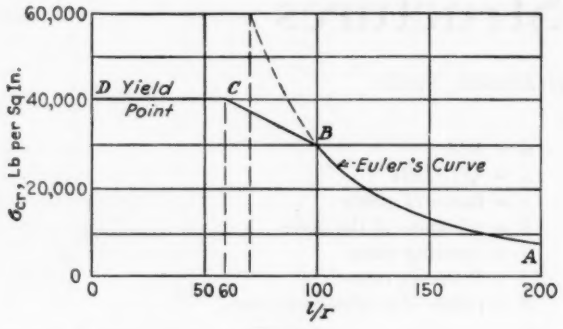


FIG. 2

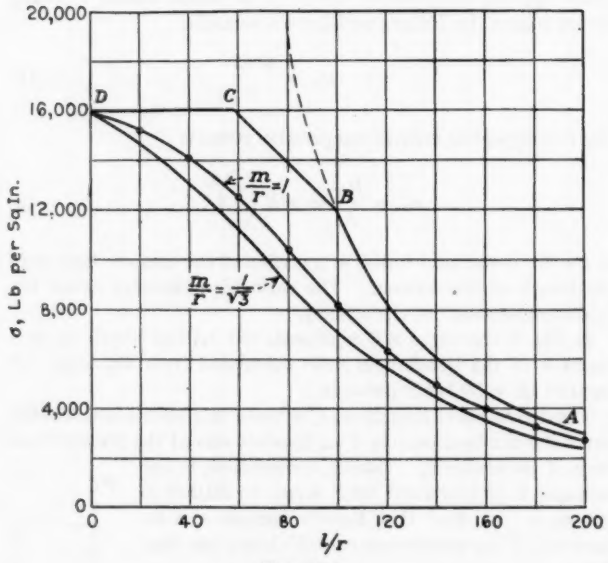


FIG. 3(*a*)
($\sigma_{y.p.}$ = 40,000 lb per sq in.; n = 2.5; e = l/400.)

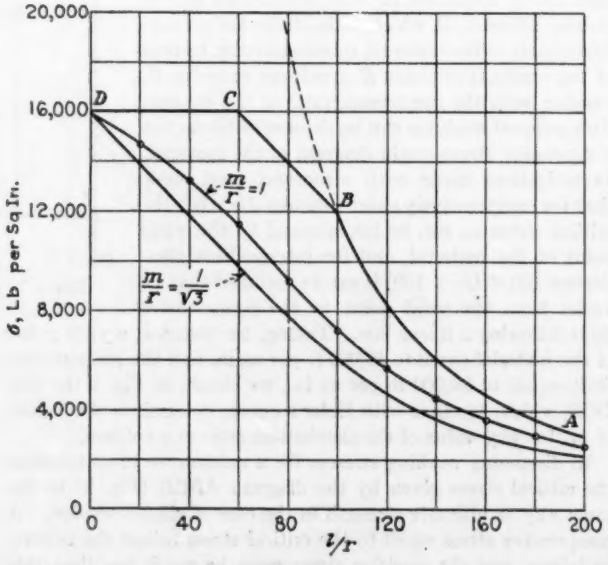


FIG. 3(*b*)
($\sigma_{y.p.}$ = 40,000 lb per sq in.; n = 2.5; e = l/250.)

Experiments show that errors in application of the load and the magnitude of initial deflection of the column usually increase with an increase in the slenderness ratio. This justifies the use of a variable factor of safety which increases with the slenderness of the column. German specifications for bridges assume, for instance, a factor of safety of 3.5 for slender structural-steel columns ($l/r > 100$) and 1.7 for very short columns. In the interval $0 < l/r < 100$ the factor of safety varies according to a linear law. In this manner a table of working stresses for columns of various degrees of slenderness can be established.

3—Eccentrically Loaded Columns

To give some theoretical justification to the practice of using a variable factor of safety, let us consider the case of a column subjected to a load having an eccentricity e at both ends (Fig. 4).

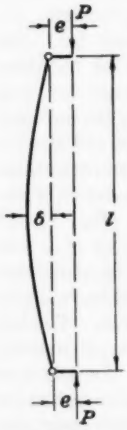


FIG. 4

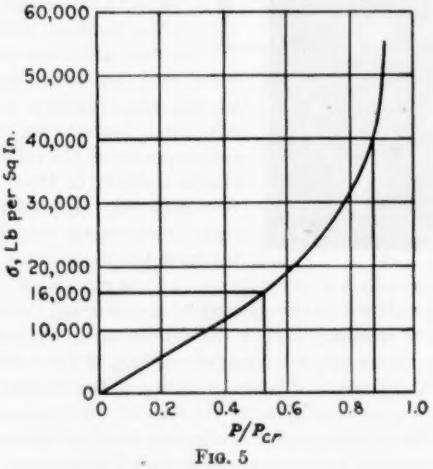


FIG. 5

In this case there is no question of critical load because the column begins to bend once the load is applied. The maximum deflection at the middle is

$$\delta = \frac{e\left(1 - \cos\frac{kl}{2}\right)}{\cos\frac{kl}{2}} \quad \ldots\ldots\ldots\ldots\ldots [3]$$

in which

$$k = \sqrt{\frac{P}{EI}} \quad \ldots\ldots\ldots\ldots\ldots [4]$$

The maximum bending moment is

$$M_{max} = P\,(\delta + e) = Pe \sec\frac{kl}{2} \quad \ldots\ldots\ldots\ldots [5]$$

and the maximum compressive stress,

$$\sigma_{max} = \frac{P}{A} + \frac{Pe}{S} = \frac{P}{A}\left(1 + \frac{e}{m}\sec\frac{kl}{2}\right) \quad \ldots\ldots\ldots [6]$$

in which S = section modulus

$m = S/A$ = core distance of the cross-section.

Remembering that k is a function of P (Equation [4]), it can be concluded from Equation [6] that in this case the stress is not proportional to the load. The relation between the load and the

stress σ_{max} can be represented by a curve which has a vertical asymptote, corresponding to the Euler load (Fig. 5). From such a curve we can see at once that in designing the column we cannot proceed as in the case of simple bending, and determine the safe dimensions from Equation [6] by substituting in it $\sigma_{max} = \sigma_{y.p.}/n$, where n is the factor of safety. Assuming, for instance, $\sigma_{y.p.} = 40{,}000$ lb per sq in. and $n = 2.5$, we find that the load producing the maximum stress, equal to $40{,}000/2.5 = 16{,}000$ lb per sq in., is larger than half of the load producing yield-point stress; i.e., by substituting in Equation [6] for σ_{max} the quantity $\sigma_{y.p.}/2.5$ the factor of safety for the load P becomes less than 2. In order to ascertain a given factor of safety n for the load P, we must take such cross-sectional dimensions of the column that the substitution of nP for P in Equation [6] will give us σ_{max} equal to $\sigma_{y.p.}$ Then the equation for the design of the column becomes

$$\sigma_{y.p.} = \frac{nP}{A}\left(1 + \frac{e}{m}\sec\frac{l}{2}\sqrt{\frac{nP}{EI}}\right) \quad [7]$$

This equation derived for columns compressed by a load with a given eccentricity can be used also in the case of centrally loaded columns. Due to imperfections in manufacturing we must always expect some eccentricity in the application of the load, and also some initial deflection. It can be assumed that such deviations from perfect conditions increase in the same proportion as the length of the column, and that their effect on the maximum stress is the same as that of an eccentricity e proportional to the length. Assuming for the ratio e/l a certain value, we can, for a given $\sigma_{y.p.}$ and for a given factor of safety n, calculate from Equation [7] the values of safe average stress P/A for any slenderness ratio. The results of such calculations[2] made for

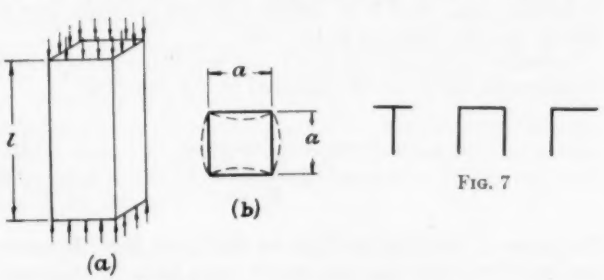


FIG. 6

FIG. 7

$e/l = 1/400$ and $e/l = 1/250$ are given in Figs. 3(*a*) and 3(*b*). The calculations are made for a built-up section with $m/r = 1$ and also for a solid rectangular section for which $m/r = 1/\sqrt{3}$. In all cases $\sigma_{y.p.}$ is taken equal to 40,000 lb per sq in., and the factor of safety $n = 2.5$. Comparing these results with the ordinates of the diagram in Fig. 2 we find that the factor of safety with respect to the critical stress in the case of the eccentricity $e = l/400$ [Fig. 3(*a*)] and for a built-up cross-section ($m = r$) varies in the inelastic region ($l/r < 100$) from 2.5 to 3.7. For a larger eccentricity, $e = l/250$ [Fig. 3(*b*)], the same factor varies from 2.5 to 4.2. Thus the discussion gives some justification for the use of a variable factor of safety in calculating working stresses for columns (see Art. 2). It also shows that the design of columns on a basis of critical stress can be replaced by designing on a basis of the yield point of the material if a suitable choice of the factor of safety n and the eccentricity ratio e/l is made.

If the eccentricities of the application of the load are known, as in the case of compressed members of trusses for which the end moments can be calculated, we derive the deflection curve on a basis of these known eccentricities and finally arrive at an equation of the same kind as Equation [7], from which the proper cross-sectional dimensions of the column can be obtained.[3]

4—Buckling of Walls of Tubular Members

In the previous discussion we have considered the buckling of the column as a whole. If the column is built up of thin plates, the walls of the column may buckle before the column buckles as a whole. Take, for instance, a thin tube of square cross-section and uniformly compressed (Fig. 6). The side plates may buckle as shown in Fig. 6(*b*), and each side can be considered as a long plate simply supported along the edges and uniformly compressed in the longitudinal direction. The critical compressive stress for a comparatively long plate ($l/a > 3$) is

$$\sigma_{cr} = \frac{\pi^2 E h^2}{3(1-\mu^2)a^2} \quad [8]$$

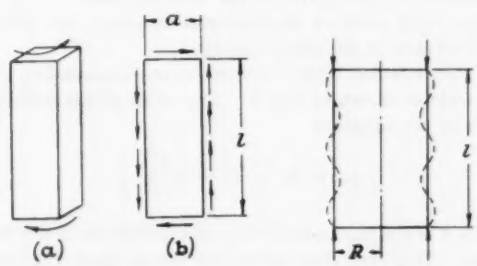


FIG. 8

FIG. 9

in which h is the thickness of the plate and μ is Poisson's ratio. For steel, for instance ($E = 30 \times 10^6$ lb per sq in., $\mu = 0.3$), taking $h = 0.01a$,

$$\sigma_{cr} = 10{,}900 \text{ lb per sq in.}$$

For any other ratio between the thickness and the width of the plate the critical stress is obtained from the equation:

$$\sigma_{cr} = 10{,}900\,\frac{10^4 h^2}{a^2} \text{ lb per sq in.} \quad [9]$$

It is seen that for steel plates, the thickness of which is one-hundredth of the width, buckling occurs at a stress which is far below the proportional limit. Substituting $\sigma_{y.p.}$ for σ_{cr} in [9] and taking $\sigma_{y.p.} = 40{,}000$ lb per sq in., the corresponding value of the ratio is found to be

$$\frac{a}{h} = 51$$

For thicker plates Equation [9] is not applicable and we can assume that σ_{cr} is equal to the yield-point stress of the material.

The question of the buckling of plates arises also in the case of such cross-sections as those shown in Fig. 7.[4]

If a square tube is subjected to the action of twist (Fig. 8), each side of the tube is in a condition of pure shear and may buckle if the shearing stress exceeds a certain critical value. Regarding this critical value we have the following information: If the length of the plate is large ($l > 4a$) and the edges can be considered as simply supported, the critical value of the shearing stress is given by the equation:

$$\tau_{cr} = 5.35\,\frac{\pi^2 E}{12(1-\mu^2)}\,\frac{h^2}{a^2} \quad [10]$$

[2] These calculations were kindly made for the author by D. H. Young, Instructor in Engineering Mechanics at the University of Michigan.

[3] Calculations of this kind were made by D. H. Young, see International Association for Bridge and Structural Engineering Memoirs, vol. 1 (1932), p. 507.

[4] Several formulas for calculating critical stresses in this case are given in the author's book, "Strength of Materials," see p. 604.

If the edges are built in, the critical stress is

$$\tau_{cr} = 8.98 \frac{\pi^2 E}{12(1-\mu^2)} \frac{h^2}{a^2} \quad [11]$$

If the plate cannot be considered as long and the edges are simply supported, the value of the critical stress is given by the equation

$$\tau_{cr} = \alpha \frac{\pi^2 E}{12(1-\mu^2)} \frac{h^2}{a^2} \quad [12]$$

in which α, a numerical factor depending on the ratio between the length and width of the plate, can be taken from the following table:

l/a =	1	1.2	1.4	1.5	1.6	1.8	2.0	2.5	3.0	∞
α =	9.4	8.0	7.3	7.1	7.0	6.8	6.6	6.3	6.1	5.35

If Equations [8]–[12] give for the critical stress a value greater than the yield point of the material in shear, the yield point should be taken as the critical stress.

If a thin circular tube is subjected to compression, the wall may buckle as shown in Fig. 9. The value of the critical stress is given by the equation

$$\sigma_{cr} = \sqrt{1/3(1-\mu^2)}\, E \frac{h}{R} \quad [13]$$

in which h is the thickness of the wall and R the mean radius of the tube. It is seen that within the elastic limit, buckling may occur only in the case of very thin tubes.

5—Long Circular Tubes Submitted to Uniform External Pressure

Another problem in which the question of elastic stability is of great practical importance is the case of a long circular tube submitted to uniform external pressure (Fig. 10). It is well known that if the pressure p exceeds a certain value p_{cr} the circular form of the tube becomes unstable and the tube collapses, as shown in Fig. 10 by the full line. The critical value of the external pressure is given by the equation:[5]



Fig. 10

$$p_{cr} = \frac{E}{4(1-\mu^2)} \cdot \frac{h^3}{R^3} \quad [14]$$

and the corresponding value of the compressive stress in the circumferential direction is:

$$\sigma_{cr} = \frac{p_{cr} \cdot R}{h} = \frac{E}{1-\mu^2} \frac{h^2}{4R^2} \quad [15]$$

in which h is the thickness of the wall and R the radius of the tube.[6] Assuming $E = 30 \times 10^6$ lb per sq in. and Poisson's ratio $\mu = 0.3$ and plotting σ_{cr} against $2R/h$, we obtain a curve AB (Fig. 11) giving the critical stress for steel tubes of various proportions. This curve represents the actual critical stress only if the magnitude of this stress does not exceed the proportional limit of the material. Beyond this limit the curve gives exaggerated values of the critical stress. To get the true values of this stress we must proceed in the same manner as in the case of a column, and work with a reduced modulus of elasticity. In this manner it can be shown that for materials with a pronounced yield point, this yield-point stress must be taken as the critical stress for thicker tubes. Taking, for instance, a steel with yield point $\sigma_{y.p.} = 40{,}000$ lb per sq in. and proportional limit $\sigma_p = 30{,}000$ lb per sq in., we find that the smallest value of $2R/h$ for which Equation [15] can be used is about 33. This corresponds to point B on the curve AB. For thicker tubes ($2R/h < 20$)

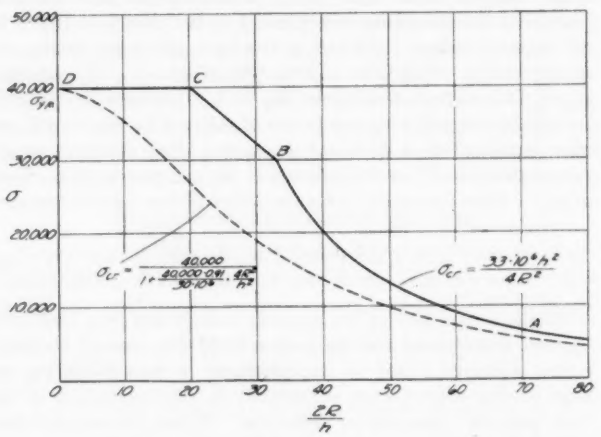


Fig. 11

the horizontal line DC giving $\sigma_{cr} = \sigma_{y.p.}$ can be used. For intermediate thickness ($20 < 2R/h < 33$) we can use, as in the case of columns, the straight line CB for determining the critical stress. Thus the line $ABCD$ gives the critical values of compressive stress for all proportions of the tubes, and if a factor of safety is decided upon, there is no difficulty in each particular case in finding the safe thickness of the tube.

Instead of the broken line $ABCD$, it is sometimes useful to have a continuous curve, as the one given by the equation:[7]

$$\sigma_{cr} = \frac{\sigma_{y.p.}}{1 + \frac{\sigma_{y.p.}(1-\mu^2)}{E} \cdot \frac{4R_2}{h^2}} \quad [16]$$

This curve is shown in the figure by the dotted line. It can be seen that for thick tubes the critical stress given by this curve approaches $\sigma_{y.p.}$, and for thin tubes it approaches the critical stress given by Equation [15]. For usual proportions the curve gives much lower stresses than those given by the line $ABCD$. This additional safety introduced by Equation [16] can be considered as compensating for the effect of any initial ellipticity of the tube which always can be expected in practice.

Let us consider now the effect of an initial ellipticity on the magnitude of maximum stress in a uniformly compressed tube. The deviation of the shape of the tube from a perfect circular form can be defined by the initial radial displacements u_o (Fig. 12). A very simple solution of the problem can be obtained by assuming that u_o is given by the equation:

$$u_o = \delta \cos 2\theta$$

in which δ is the maximum initial radial deviation from the circle and θ is the central angle to the point in question. When θ is varying from 0 to $\pi/2$, the radial deviation is changing from $+\delta$ to $-\delta$, and the general shape of the tube is as shown in Fig. 12 by the full line.

Under the action of external uniform pressure p, there will be an additional flattening of the tube. The corresponding radial

[5] This equation was obtained by M. Levy, *Journal d. Math. pure et Appl.* (Iiouville), ser. 3, vol. 10, p. 5, 1884. The derivation of the equation can be found in the author's "Strength of Materials," vol. 2, p. 598, 1930.

[6] The thickness of the wall is assumed small, and the external radius and mean radius are assumed to be the same.

[7] Such a curve was proposed by R. V. Southwell, *Phil. Mag.*, vol. 29, p. 67, 1915.

displacements will be represented by u. Then to determine u we can use the differential equation:[8]

$$\frac{d^2u}{d\theta^2} + u = -\frac{12(1-\mu^2)}{Eh^3} MR^2 \quad [17]$$

in which M is the bending moment, taken positive when it produces a decrease in the initial curvature of the tube. Following

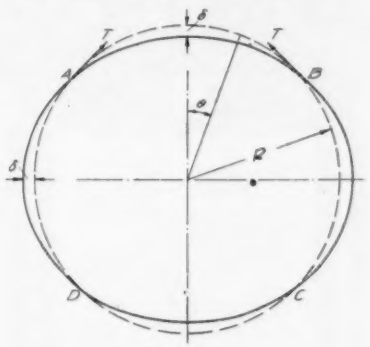


FIG. 12

this rule, we conclude that in the portions AB and DC there will be produced by a uniform external pressure a positive bending moment and in the portions AD and BC a negative bending moment. At points A, B, C, and D the bending moment is zero, and the actions between the parts of the tube are represented by forces T tangential to the circle representing the ideal shape of the tube. This circle can be considered as the funicular curve for the external pressure. The compressive force along this curve remains constant and equal to T. Thus the bending moment at any cross-section is obtained by multiplying T by the total radial displacement $u_o + u$ at this cross-section. Taking an elementary ring from the tube of a width, in the direction perpendicular to the plane of the figure, equal to unity, the compressive force T is obtained from conditions of static equilibrium to be equal to pR. Then the bending moment at any cross-section of the ring is

$$M = pR(u + u_o) = pR(u + \delta \cos 2\theta)$$

Substituting in Equation [17], we obtain

$$\frac{d^2u}{d\theta^2} + u = -\frac{12(1-\mu^2)}{Eh^3} pR^3(u + \delta \cos 2\theta)$$

or

$$\frac{d^2u}{d\theta^2} + u\left(1 + \frac{12(1-\mu^2)}{Eh^3} pR^3\right) = -\frac{12(1-\mu^2)}{Eh^3} \delta pR^3 \cos 2\theta$$

The solution of this equation satisfying the conditions of continuity at the points A, B, C, and D is

$$u = \frac{\delta p}{p_{cr} - p} \cos 2\theta \quad [18]$$

in which p_{cr} is given by Equation [14]. It is seen that at points A, B, C, and D, u and $d^2u/d\theta^2$ are zero. Hence the bending moments at these points are zero, as was assumed above. The maximum bending moment occurs when $\theta = 0$ and $\theta = \pi$. At this cross-section:

$$M_{max} = pR\left(\delta + \frac{\delta p}{p_{cr} - p}\right) = pR\delta \frac{1}{1 - (p/p_{cr})} \quad [19]$$

[8] This equation was established by J. Boussinesq, *Comptes Rendus*, vol. 97, p. 843, 1883. The derivation of this equation can be found in the author's "Strength of Materials," vol. 2, p. 457, 1930.

It is seen that for small values of the ratio p/p_{cr} we can neglect the change in the ellipticity of the tube due to pressure p and multiply the compressive force pR with δ in calculating the maximum bending moment. When the ratio p/p_{cr} is not small, the change in the initial ellipticity should be considered and Equation [19] must be used in calculating M_{max}.

The maximum compressive stress is now obtained by adding to the stress produced by the compressive force pR the maximum compressive stress due to bending moment M_{max}. Thus we find:

$$\sigma_{max} = \frac{pR}{h} + \frac{6M_{max}}{h^2} = \frac{pR}{h} + \frac{6pR}{h^2} \frac{1}{1 - (p/p_{cr})} \quad [20]$$

The pressure p at which yielding of the material begins is found from the foregoing equation by substituting $\sigma_{y.p.}$ for σ_{max}. Then

$$\sigma_{y.p.} = \frac{pR}{h} + 6p \cdot \frac{R^2}{h^2} \frac{\delta}{R} \frac{1}{1 - (p/p_{cr})} \quad [21]$$

Using the notation

$$\frac{R}{h} = m; \qquad \frac{\delta}{R} = n \quad [22]$$

the equation for calculating p becomes

$$p^3 - \left[\frac{\sigma_{y.p.}}{m} + (1 + 6mn)p_{cr}\right] p + \frac{\sigma_{y.p.}}{m} p_{cr} = 0 \quad [23]$$

It should be noted that the pressure p determined in this manner produces a stress equal to $\sigma_{y.p.}$ only in the weakest spot. This pressure is smaller than the pressure at which the collapsing of the tube occurs, and it becomes equal to the latter only in the case of a perfectly round tube. Hence the use of Equation [23] for calculating critical values of p is on the safe side. Without any difficulty a curve analogous to the dotted curve in Fig. 11 can be constructed for any value of δ/R. Having a series of such curves for various values of initial ellipticity, the determination of the necessary thickness of the tube can be easily made in each particular case.

We have given here only a few examples of buckling of thin-walled structures. More problems of this kind are discussed in the author's paper in the Reports of the Third International Congress for Applied Mechanics, Stockholm, 1930. The literature on the subject is given there also.

Discussion

GEORGE E. BEGGS.[9] Since the experiments on buckling of wrought-iron compression members were made in connection with the design of the Britannia and Conway bridges over eighty years ago, no equally thorough tests have been conducted to guide bridge engineers in selecting a proper plate thickness for cellular compression members of steel having one or more squares in their cross-section. [See Fig. 13 (*b*) and (*c*)]. The most elaborate tests of this character in recent years are the web plate compression tests made at the U. S. Bureau of Standards and reported in its Technologic Paper No. 327. In the writer's opinion, these tests do not exhibit accurately the structural behavior of a cellular column, nor do the test results support the mathematical formula for critical buckling stress, as will be explained below. The best approach to the mathematical solution of the problem

[9] Professor of Civil Engineering, Princeton University, Princeton, N. J.

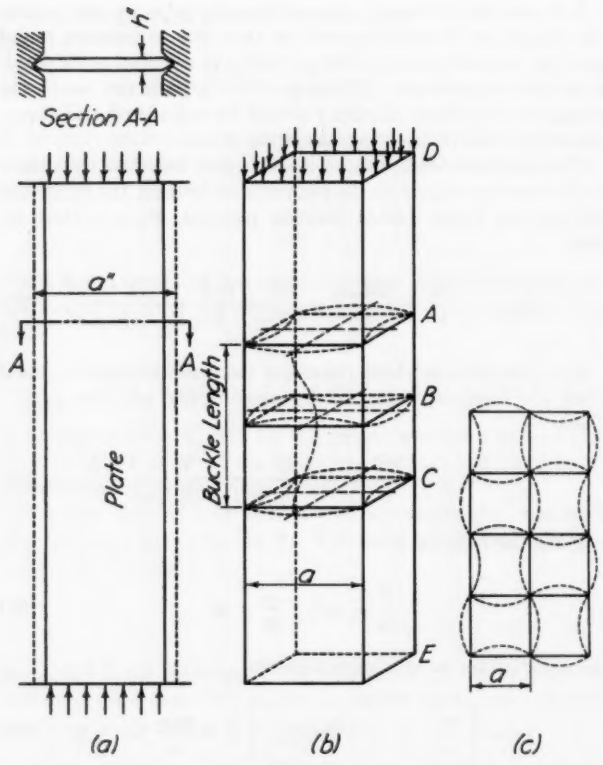


FIG. 13

of plate buckling was made originally by George H. Bryan (Proc. Lond. Math. Soc., vol. 22, pp. 54–67), in which will be found Equation [8] of the paper, being the same as the one quoted in Fig. 15.

Bryan assumed a truly flat and perfectly elastic plate of infinite length, supported freely along its longitudinal edges and loaded in compression, as indicated in Fig. 13 (*a*). He developed for this ideal case a formula giving the maximum unit stress that this plate would carry without starting to become crooked. The resulting expression and Euler's column formula are strikingly similar:

Bryan's Web-Plate Buckling Formula:

$$\sigma_{cr} = [\text{constant}]\pi^2 E/(a/h)^2$$

Euler's Column Formula:

$$\sigma_{cr} = [\text{constant}]\pi^2 E/(l/r)^2$$

The plotted curve for Bryan's formula (Fig. 15) is closely similar to Euler's curve in Fig. 2 of the paper. The criticisms of the application of Euler's formula to the design of columns of usual slenderness ratios seem to apply to the general use of Bryan's formula in designing web plates of usual thinness ratios. By reason of the ever-present initial crookedness of web plates, it appears that Bryan's formula is inapplicable for an accurate computation of buckling resistance, particularly when the critical stress found by this formula has a value near the yield point of the material. Bryan's formula tells only the maximum load that a single truly flat plate [Fig. 13 (*a*)] will carry before starting to buckle. The engineer desires a formula that will give him the buckling resistance of a plate or assemblage of plates of steel, all of which have initial Bryan buckles in them due to cooling at the rolling mill or irregularities due to handling. From published data about such initial crookedness of web plates, the writer has observed that these plates may be expected to have initial buckles of the Bryan type with the crests of the buckles $a/350$ above the dents (Fig. 13). It is suggested that the mathematician assume this degree of initial crookedness of the plates, the yield point, and other properties of the material, and derive a practical formula or curve for the buckling resistance of a square tube or cellular column in compression.

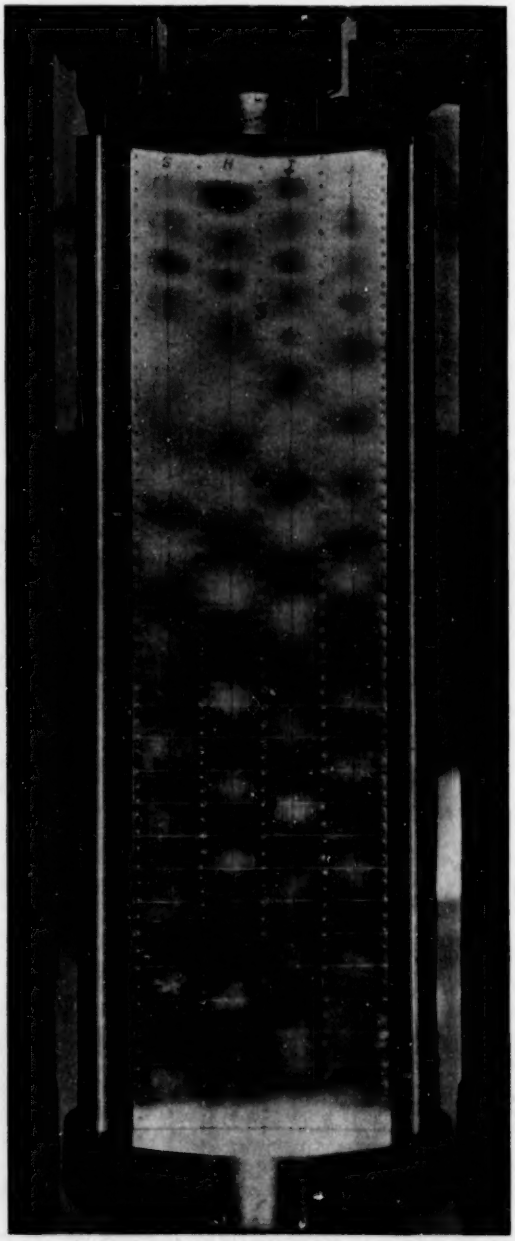


FIG. 14

Fig. 14 shows a cellular column under load. The cross-section consists of eight cells [see Fig. 13 (*c*)]. The buckling load has a value such that the column yields without added load at an average unit stress well below the yield-point value of the material. The crests and dents of the buckles alternate both vertically and around the cross-section. The vertical distance center to center of buckle dents (Fig. 14) averages 1.38 times the full width of a partition, as was noted for all partitions of this and a similar

specimen, whereas Bryan's theory indicates that the buckle length should not exceed the full width of partition. The pattern of the buckles became apparent at a unit stress about $^3/_4$ of that indicated by Bryan's formula for a single plate. [Fig. 13 (*a*).] Nature's tendency to produce a buckle pattern was so strong that several initial dents reversed their curvature and became crests, in order to accommodate the prevailing pattern. At one elevation, as at *A*, Fig. 13 (*b*), the plates bent in and out around the tube, as shown by the dotted line. At a lower elevation, as at *B*, a similar but opposite order of dents and crests occur around the sides of the square cell. The reversal from dents to crests at successive elevations evidently subjects each long corner, as *ABC*, of the square cell to a torsion, alternately clockwise and counterclockwise in plan. If the partitions are joined by steel angles in the corners of the cell, these angles will be subjected to an alternating torsional strain and will offer some resistance thereto. However, since angle sections have extremely little

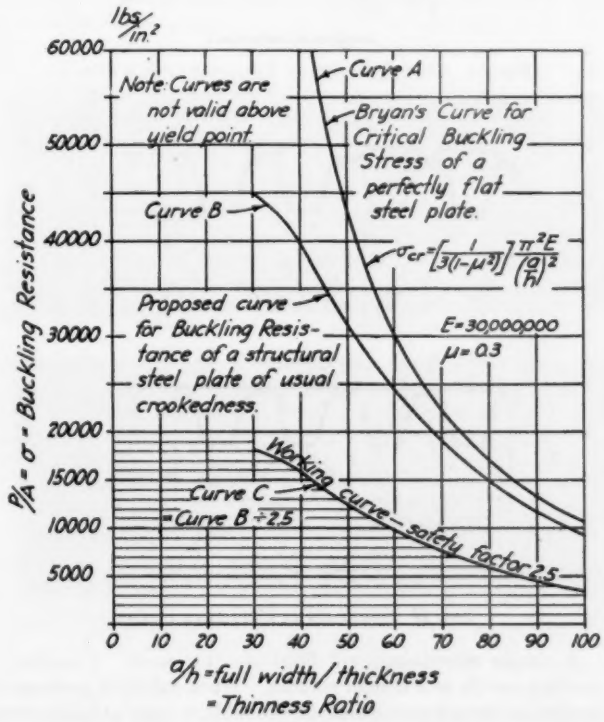


FIG. 15

torsional resistance, they would afford no great support against the buckling of the web plate. In the web-plate compression tests at the U. S. Bureau of Standards (Tech. Paper 327), this natural alternating torsional movement at the two vertical edges of a web plate was prevented by the attachment of heavy steel stiffeners to the flange angles of the tested partition, with the result that the natural buckling of the webs was hindered. In the one test where these stiffeners were not very tightly attached, failure occurred at a comparatively low stress. These tests therefore give results on the unsafe side, even though the actual structure is safeguarded by an ample safety factor. A further criticism of the test report referred to is that the conclusions there given support a misquoted formula of Bryan instead of the correct one. This misquoted formula is incorrect by 25 to 30 per cent on the unsafe side, this estimate depending on the value taken for Poisson's ratio.

Just as mathematicians have developed formulas and curves to predict the actual capacity of columns of usual crookedness and eccentricity, so should they also give their attention to deriving similar ones to predict the buckling resistance of thin, but somewhat initially crooked, web plates of cellular compression members. Engineers should also carry out buckling tests with cellular columns of steel comparable to the experiments made by Fairbairn with wrought-iron cellular columns. Only by such studies can a reliable guide be found for the efficient design of the webs of such members. Until then, part of the factor of safety will remain a factor of ignorance.

In the absence of adequate theory and tests, the writer proposes curves *B* and *C* of Fig. 15 as a guide in selecting thickness of webs for cellular compression members of structural steel, Curve *B* fits reliable test data more safely than curve *A*, but such data are meager.[10]

The writer hopes that the author may be able to extend the excellent studies he has made of the ideal behavior of plates subject to buckling and to modify the plate-buckling theory so as to allow for some specified initial crookedness of the plates. A practical formula or design curve taking such initial crookedness into account will be of value.

R. EKSERGIAN.[11] The choice of suitable working stress is governed essentially by such factors as eccentricity of loading, degree of constraints at the ends, initial deflections, etc., and the writer feels that to provide for such contingencies reference to full-size tests approximating actual conditions is always desirable when possible.

In choosing a factor of safety, the preliminary limits are shown by the boundary curve of Fig. 2 which extends over two regions, one the yield limit for low values of l/r since in this region the critical stress exceeds the yield value, and the other the column-stability region where the limiting stress depends upon the critical column stress. The former region is dependent upon the physical properties of the material, i.e., the yield value, while the latter is independent of all except the modulus of elasticity E. Now in any actual column eccentricities must exist, and such eccentricities introduce a transition curve from the yield-limit region into the column-stability region. Equation [7] represents this transition curve, and it will be noted the peak stress, after a suitable factor of safety on the load is used, is limited to the yield value. At first glance we might conclude there is some inconsistency since with the boundary-curve idea for the higher values of l/r, the stress is totally independent of any yield limit, whereas the column equation for any value of l/r appears to always depend on the yield value as well. A little study of the equation, however, will show that for the higher values of l/r, the working stress P/A is very little affected by large changes in yield values, or, in other words, the ratio of the critical stress to the working stress, i.e., the factor of safety on the column stability, is little affected by any choice of yield value for the higher values of l/r. For instance, if we write Equation [7] in the form,

$$\sigma_{y.p.} = n\sigma\left(1 + \frac{e}{m}\sec\frac{l}{2}\frac{n\sigma}{\sigma_{cr}}\right), \quad \sigma_{cr} = \frac{2E}{(l/r)^2}$$

and assume $n = 2.5$ and $l/r = 200$ for a slender column, and $m/r = 1$, with $e/l = 1/400$, then when $\sigma = 2500$ lb/per sq in., $\sigma_{cr.} = 7400$ lb per sq in., $\sigma_{y.p.} = 2.5 \times 2500\,(1 + 1/2\sec 82.8°) = 31{,}200$ (lb per sq in.), and $n' = 7400/2500 = 2.96$; and when

[10] A safety factor of 2.5 was used to obtain curve *C* from curve *B* only because this factor was used by the author in obtaining his working curve for columns. However, if further tests and study show that curve *B* approximates closely the true ultimate buckling resistance of commercial web plates, a smaller factor of safety between 1.5 and 2.5 may be chosen, depending on conditions and individual judgment.

[11] E. I. du Pont de Nemours & Co., Wilmington, Del. Mem. A.S.M.E.

σ = 2700 lb. per sq in., $\sigma_{y.p.}$ = 2.5 × 2700 (1 + 1/2 sec 82.2°) = 57,800 (lb per sq in.) and n' = 7400/2700 = 2.74.

Thus nearly doubling the yield value changes n' by only $7^1/_2$ per cent. For this reason, with the higher values of l/r, raising the yield value has very little effect on increasing the allowable working stress.

In the case of rods subjected to alternating or cyclic stresses, as connecting rods in engines, it would seem that the endurance limit should replace the yield value in the boundary curve *DC* in Fig. 2, and likewise in the column Equation [7].

In the design of rods for high-speed engines, particularly locomotive rods, due to the bending caused by centrifugal whip action, the rods are either of I- or rectangular section with a large depth of section in the plane of reciprocation. In the latter the rod acts as an ideal pin-connected column, while in the lateral plane, due to the bearing constraint of the pins, it approaches a column with fixed ends. Due to the inertia bending action, the depth of the rod with rectangular sections is never less than 2 and frequently from 3 to 4 times its width. Thus if b is the width and h the depth of the rod, the critical compression stresses σ_v for the plane of reciprocation and σ_h for the lateral plane have the relation, $\sigma_h/\sigma_v = 4b^2/h^2$, so that when $h = 2b$, failure can only occur by lateral buckling perpendicular to the plane of reciprocation. It is to be noted that maximum compressive loadings occur at low speeds where the inertia forces can be neglected.

Some years ago the late Professor Lanza conducted at the Massachusetts Institute of Technology a series of tests on full-sized locomotive rods having various values of l/r. Up to a value of l/r = 160, which the writer believes was the limiting value used in the test, failure occurred practically at the yield value, around 38,000 lb per sq in. A fully constrained column would give a compressive stability with l/r = 160 and with $E = 30 \times 10^6$ equal to 47,000 lb per sq in. Since the rods failed at the yield value rather than by actual column instability, these tests assure us that at least the effect of the lateral bearing at the pins gives sufficient constraint to approximate over $^3/_4$ the stability of a fully constrained column, which is equivalent to a pin column of length 0.58 l. Professor Lanza recommended an allowable working compressive stress at 8000 lb per sq in. with l/r = 160. In practice stresses up to 10,000 lb per sq in. have been used with this value. This corresponds to a factor n = 3.5. The fact that the lateral constraint of the pin bearing is very effective is evident from the fact that with side rods, bearing centers may have considerable eccentricity with no allowance for this eccentricity.

J. M. Frankland.[12] One aspect of instability in elastic systems not discussed in the paper, seems to the writer of practical importance in present-day construction of thin-walled structures.

Owing to a certain rather widespread misconception of Euler's theory of buckling, it has unfortunately been extended into a range for which it was never intended. This may best be illustrated by referring to Fig. 16, where P is the load and d the deflection at the end of the strut. From O up to A the load is less than the critical Euler load and the strut remains straight. At A the critical load P_{cr} is reached. Now the statement is often made that at this load the strut becomes unstable and that any deflection greater than or equal to d_{cr} is possible; in other words, the column should continue to deflect along AB at constant load until it folds up completely. But the correct interpretation of the theory is that the strut begins to bend at the critical load and the deflection begins to increase more rapidly along some such curve as AC. This means that there is a perfectly definite deflection for each load, even when the load exceeds the critical value. Provided that the combined stresses due to bending and direct compression nowhere exceed the elastic limit, the strut can be loaded above the critical point A without producing any permanent deformation, the column becoming straight again as soon as the load is reduced below the critical value.

[12] 37 W. 16th St., New York, N. Y.

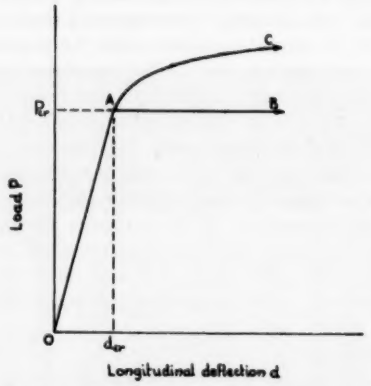


Fig. 16 Load-Deflection Relations for a Strut

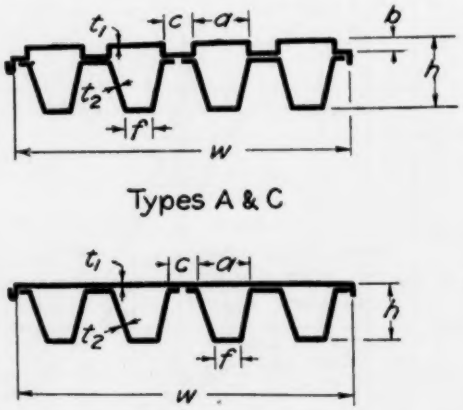


Fig. 17 Sections of Cellular Beams

A simple experiment will illustrate this point. Consider a knitting needle as a slender column. When sufficient pressure is applied at the top as it stands on the table, a large lateral bowing is produced, the amount being quite definite for each pressure exerted. That the needle has only been elastically deformed is obvious, for it becomes quite straight when the pressure is removed. It is perhaps unfortunate that this relatively stable state of affairs should be known as "elastic instability."

There is consequently in slender members a range of loads above the Euler critical load for which the structure receives no permanent deformation. This range, of course, will increase with the slenderness ratio. It is obvious that little use can be made of this characteristic property when dealing with columns such as are customarily used in construction, since in none of their parts is the slenderness ratio high and the added range of loads is negligible. There are, however, certain types of structures coming into increasing use today in which the necessity for economizing weight has led to the use of slender parts—in particular, sections built from sheet metal. The author's box column is an example. Given a sufficiently thin wall, considerable elastic buckling can develop before crippling or permanent buckling occurs. The theoretical analysis of such conditions is difficult and edge conditions are so important that it is hard to apply

elastic theory to structures of this kind. Experimental investigation will probably yield more satisfactory results. With Prof. H. L. Whittemore of the Bureau of Standards the writer has recently been engaged in making mechanical tests of a structure[13] which brought these points vividly to our attention. It is a cellular beam of sheet steel intended to be used in floor construction. Fig. 17 shows some typical sections. The beam is assembled from light-weight corrugated sheets and made into panels 2 ft wide and of the desired span length. The depth h is $5^5/_8$ in. The thickness of the steel in the experimental panels ranged from 0.05 in. to 0.11 in. (U. S. Standard gages 18 to 12), various combinations of gages being used to obtain the necessary range of section properties. The panels were tested as beams, so the tops of the sections were in compression. Buckling occurred only in the portion labeled a. Now the only reason to fear elastic buckling in the cell tops was that it might possibly increase the transverse deflection of the beam. The tests showed, however, that there is no sensible effect on the deflection until permanent buckles begin to appear.

22,000 lb per sq in. and the other with a yield point of about 43,000 lb. That is to say, the high yield point had increased the stress range of elastic buckling by about 10,000 lb per sq in. This occurs for a/t_1 equal to 52, which the author has shown to give a relatively stiff wall in a box column. Such an increase in the crippling strength is by no means to be neglected; in fact, the increased yield point has practically doubled the crippling strength. The theory of elastic buckling would indicate the same allowable working load no matter what the yield point was. Such a section after testing is shown in the lower part of Fig. 18. Earlier stages of buckling showed much greater uniformity. What Fig. 18 shows is a late stage in failure.

The flat-top section with a/t_1 equal to 62 showed permanent buckling at stresses about 50 per cent of the yield point.

The author's formula for the secondary elastic buckling of a box column does not apply to the case of these cellular beams, since here the stresses are more complicated, there being shear as well as compression in the top of the cell. Also one cannot assume that the compressive stresses are uniform across the width

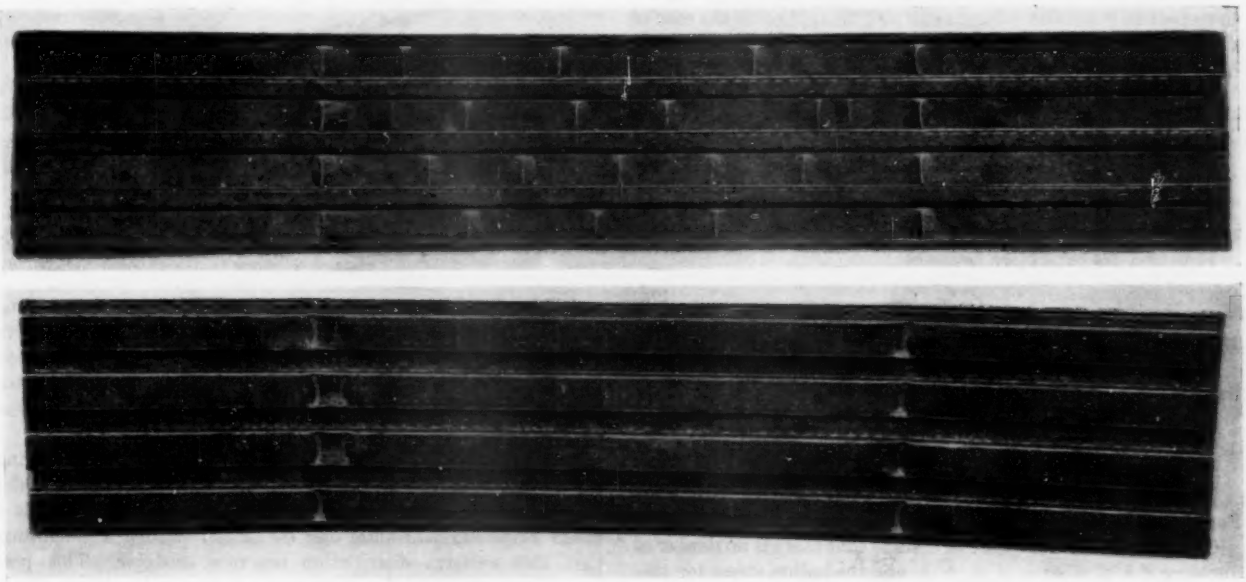


FIG. 18 TOP VIEW OF CELLULAR BEAMS AFTER TESTING

As seen in Fig. 17, the section is not symmetrical about a horizontal axis, the neutral axis being 33 to 40 per cent of the depth below the top of the beam. This gives a range of maximum compressive stresses from 50 to 67 per cent of the maximum tensile stresses. The values of a/t_1 ranged from 36 to 62. At the lowest ratio of 36 in a specimen with a corrugated top, there was little buckling of the cell tops and the compressive yield point of the material was developed. The upper half of Fig. 18 shows the top surface of such a specimen after testing. The dents at the quarter-points mark the places where the load was applied. Strain lines in the mill scale at 45 deg to the axis of the panel can be seen in the cell tops, which indicates a simple compressive yield. When a/t ranged from 52 to 62 in a corrugated-top type and the compressive stress was 60 per cent of the tensile stress, compression crippling began at about the same loads that developed the yield on the tension side. It was found for this type of section and for these values of a/t_1 that permanent buckling took place at stresses about 55 per cent of the yield point. This was true for two materials, one with a yield point of about

[13] Research Paper 463, Bureau of Standards *Journal of Research*, vol. 9 (1932), p. 131.

of the cell top. This, no doubt, explains why buckling was obtained with such low values of a/t_1. Further details of these tests can be found in the report published in the August, 1932, number of the Bureau of Standards *Journal of Research*.

The points the writer wishes to emphasize are:

1 There exists in general a range of elastic buckling before crippling occurs. Experiment shows that in certain cases the load range may be considerable.

2 The upper limit to this load range is set by the yield point, the lower limit by the critical load for elastic buckling. This latter is determined by the slenderness ratio and the modulus of elasticity. Thus slender members in high-strength steel and in duralumin should show this effect to a pronounced degree.

3 The general phenomenon is, however, characteristic of all cases of buckling in one degree or another, so that its consideration need not be restricted to design in sheet metal. It is suggested that these considerations might play a part in the design in stiffeners in plate-girdle webs (though they probably would not apply to stiffeners under the points of concentrated load). It may often be possible for the designer to use this range of loads up to the point of crippling to increase the economy of his struc-

ture without sacrificing the safety, particularly when using sheet metal in which local buckling can occur.

G. H. MacCullough.[14] In the present paper, the author has utilized the results of some European investigators showing that Euler's column analysis can be extended[15] into the region beyond the proportional limit and the results of this logical extension are very closely approximated by the line *ABCD* of Fig. 2. He also gives a theoretically correct formula for the design of eccentrically loaded columns. Perhaps his Equation [7] ought to be supplemented by further investigation whenever the axis about which the bending moment acts is not the axis about which resistance to bending is least. The danger of buckling as in pure column action about the other axis (the axis of least radius of gyration) with the load P treated as a centrally placed load should also be considered. The actual stress P/A in this case should not exceed the allowable stress given by the line *ABCD* of Fig. 3 with l/r based on the least radius of gyration.

Markus Reiner.[16] The reasoning which the author applies in respect to prismatic columns can also be applied to the case of the buckling of walls of tubular members. Equation [8], originally due to G. H. Bryan, can be written as

$$\sigma_{cr} = \frac{\pi^2 E}{3(1-\mu^2)}\left(\frac{h}{a}\right)^2 \quad [24]$$

The expression $\pi^2E/3(1-\mu^2)$ is a constant (c) of the material. The ratio a/h, put equal to α, is analogous to the slenderness ratio and may be called the "thinness ratio."

Equation [24] therefore becomes

$$\sigma_{cr} = c/\alpha^2 \quad [25]$$

which is the equation of a third-order hyperbola.

Equation [25] is not valid for stresses above the proportional limit ($\sigma_{p.l.}$). The limiting thinness ratio is therefore

$$\alpha_B = \sqrt{\frac{c}{\sigma_{p.l.}}} \quad [26]$$

For thicker plates Bryan's equation is not valid.

On the other hand it is safe to assume that there is no danger of buckling if $h \leqq a$, or $\alpha \leqq \alpha_c = 1$, and the failure stress for this condition is therefore equal to the yield point.

Substituting the values for E, $\sigma_{p.l.}$, $\sigma_{y.p.}$, and μ given in the paper,

$$c = \frac{3.14^2 \times 30 \times 10^6}{3(1-0.09)} = 1.083 \times 10^8 \quad [27]$$

and

$$\alpha_B = 10\sqrt{\frac{1083}{30}} = 60 \quad [28][17]$$

Plotting σ_{cr} against α in Fig. 19, we get, using the above values, the parts of the graph between points D and C and between B and A. Between points B and C a theoretical law could be derived in a way similar (but more complicated) to that in which Engesser and Kármán derived their law for short prismatic columns. However a straight line connecting points B and C is on the safe side and probably a sufficient approximation, and a linear law

$$\sigma_{cr} = C_1 - C_2\alpha \quad [29]$$

is therefore proposed instead of Bryan's law for plates thicker than $h_B = a/60$, or, more generally, $h_B = a\sqrt{(\sigma_{p.l.}/c)}$.

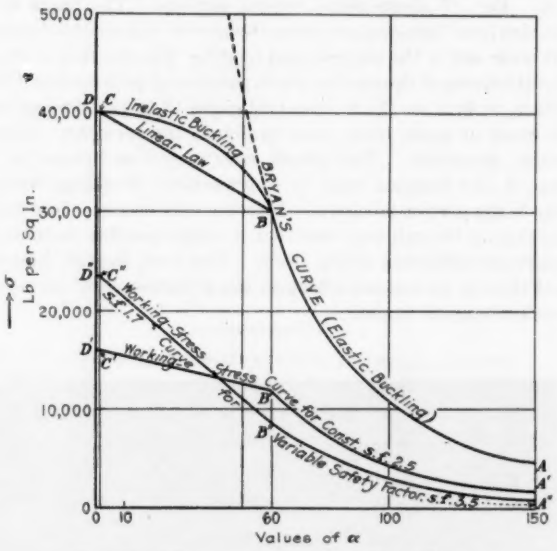


Fig. 19 Buckling of Column Web Plates
(E = 30,000,000; μ = 0.3; $\sigma_{p.l.}$ = 30,000; $\sigma_{y.p.}$ = 40,000; α = width of plate ÷ thickness.)

If we assume a constant factor of safety of 2.5, the working stresses for different thinness ratios are given by the curve $A'B'C'D'$. However, it is better practice, for reasons explained by the author in detail in this paper, to assume a variable factor of safety of from 1.7 to 3.5. The working stresses are then given by the curve $A''B''C''D''$.

It may, however, be doubted whether the concept of buckling failure can be applied at all to the failure of column plates under thrust as occurring in actual practice. The buckling stress is a stress *uniformly distributed over the section* of such a magnitude that this uniform distribution becomes unstable. This presupposes in the case of a plate that the plate is perfectly plane. However, it is well known that large plates exhibit considerable initial crookedness. The stresses in such plates under thrust are therefore not uniformly distributed from the beginning, and failure will occur after the maximum stress has reached the yield value. In order to determine the working stresses in a column web plate, an analysis of the stresses developed in a *curved plate* is therefore necessary. The design of such plates would have to proceed on a basis of the yield point, the factor of safety, and a suitable choice of an initial crookedness of the plate.

Apart from the theoretical difficulties of the problem, there is that of the choice of the initial crookedness, which seems to offer infinite possibilities. If, however, the initial web-plate contours, which have been very carefully measured up by R. S. Johnston,[18] are inspected, distinctive regularities can be noted, which may find their explanation from the process of rolling the plates.

The writer has in hand a theoretical investigation along these lines, the results of which he hopes soon to publish elsewhere.

Author's Closure

In writing the paper on thin-walled structures for the symposium on working stresses, the author had in mind only to bring

[14] Professor of Engineering Mechanics, Worcester Polytechnic Institute, Worcester, Mass. Mem. A.S.M.E.

[15] For an explanation of the method of extending Euler's analysis, see Timoshenko's "Strength of Materials," vol. II, arts. 37 and 38.

[16] Visiting Research Professor, Lafayette College, Easton, Pa.

[17] The author gives for the limiting ratio a value of 51 by using $\sigma_{y.p.}$ instead of $\sigma_{p.l.}$ as a criterion.

[18] Technologic Papers No. 321, Bureau of Standards (1926).

the attention of designers to the cases in which not only the strength of the material but also the stability of the structure should be considered in determining the allowable stresses. The problems of columns, thin plates, and thin shells were briefly discussed as examples without going into details.

Prof. G. E. Beggs, in his discussion, gives an interesting example of a cellular column buckled under the action of uniform compression. He states that buckling started at a load three-fourths of that indicated by Bryan's formula for a single plate. Such a discrepancy can be explained if we keep in mind that certain initial crookedness of thin plates is always present at the very beginning of experiments and by application of a compressive load further increase of this initial crookedness is produced. This makes it difficult to establish experimentally the magnitude of the critical load. To eliminate this difficulty, the writer and several other experimenters, who worked with stability of thin plates, used the method in which deflections of the plate are measured and curves are plotted representing deflections as functions of the load. These curves have the form of hyperbolas, and by drawing asymptotes to these curves critical loads can be determined with a better accuracy than can be done by simply observing when buckling begins.

Professor Beggs mentions also that the length of waves in which the buckled sides of the column are subdivided was equal to 1.38 the width of a partition, "whereas Bryan's theory indicates that the buckle length should not exceed the full width of partition." It should be noted that if the measurements were taken between two consecutive crests, the distance, following Bryan's theory, should be twice the width of a partition. If Professor Beggs has found shorter distances, it can be explained only by the stiffening action of steel angles in the corners of the cell. It is theoretically proved that any fixity of the edges results in a decrease of wave length.

The author agrees completely with Professor Beggs that the theory of thin-walled columns is not sufficiently developed and that further theoretical and experimental investigations of thin-walled structures are necessary.

Mr. J. M. Frankland, in his discussion, mentions that the Euler load does not represent the maximum load which a compressed slender bar can carry and that to produce an increase in the deflection of a buckled compressed bar certain increase of the load is necessary. This is a very well-known fact. It is also known that the relation between the load and deflection will be represented by a curve which is very different from the curve AC of Mr. Frankland. The true curve is of such a kind that for an increase of the load of 1 per cent above Euler's value, the deflection becomes of the amount of 9 per cent of the length of the bar. Practically, a slight increase of the load above Euler's load produces complete failure in a column of usual proportions.

The Euler load is called critical and indicates instability, because if the column is compressed above this load the two forms of equilibrium are possible—one straight form, which is unstable, and the other curved, which is stable.

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Progress in Fuels

Fuels and Fuel Utilization in 1932

THE last year has been characterized by a general slackening of technical progress in the fuel industries, a condition which has not been unexpected. Industrial research in most branches has been confined principally to operating problems, and much caution has been observed toward new developments. The trends of recent years have for the most part continued in the same direction, although some changes are noticeable.

Of particular importance has been the increased competition between solid, liquid, and gaseous fuels. This intense competition has created an economic problem difficult of solution. Overproduction and competition within the separate branches of the industry have been serious enough, and the present situation is causing considerable concern on the part of fuel producers. The availability of fuel oil and natural gas at prices competitive with solid fuel has been responsible for a number of technical developments, chief of which is the demand for combination burners that can handle any of the fuels. Installation of these burners permits changing from one fuel to another as the relative prices fluctuate.

Activity in the domestic field has been greater than usual, due partly to the competition between fuels and partly to the greater stability of the domestic market as compared with the industrial.

As in all industries, employment conditions in the fuel industries are unsatisfactory. Because of the almost complete suspension of new construction, there is practically no demand for designing or construction engineers. Industrial research has been drastically curtailed, and even university research has been reduced considerably. There are a few openings from time to time for operating engineers, particularly for those having behind them records of reduced operating costs. The outlook for the future, however, is hopeful. The depression has brought a keener realization of the importance of fuel economy, and increased efforts in this direction may be expected.

Natural Fuel Production

Bituminous-coal-mining statistics indicate the unsatisfactory economic condition of the industry. The situation has been bad for some years, but it has been aggravated by the general curtailment of industrial activity. In the first eight months of 1932 there were mined 184,910,000 tons of bituminous coal, a decrease of 26.1 per cent from the 1931 production in the same period.

Differences in production costs in the various mining sections, due partly to a lack of standardization of wages according to the union or non-union character of the labor,[1] an absence of any agency to equalize these differences, and a diminishing market have been responsible for an orgy of price cutting. Much of the coal produced has been offered for sale at prices far below the cost of mining.[2]

The effect on those producers who have survived has been to compel the adoption of cheaper production methods and the preparation of a more salable product. Reduction of mining costs is being accomplished by closing high-cost mines and by further mechanization of operations. During 1931 the tonnage loaded by machines increased by 1.3 per cent in spite of a large decrease in total tonnage. A better product is being obtained by increased use of cleaning methods and more uniform sizing.[3,4]

Some short-sighted purchasers of fuel have been pleased with the present situation in the bituminous-coal industry, resulting as it has in a better fuel at a lower price. To the extent that the condition of the industry has forced the preparation of a better product and the adoption of lower cost methods, the results have been beneficial, not only to the consumers, but to the producers. Where the result has been the sale of coal below the cost of production, no permanent benefit can accrue even to the consumer. The reestablishment of the bituminous-coal industry in a position where reasonable profits are possible is of vital importance to producer and consumer alike.

Remedial measures proposed are directed toward the regulation of production and better controlled marketing methods. A majority of operators would welcome some regulative agency provided it is not governmental. The proposed mechanism of the Davis-Kelly bill was actively opposed by most operators, although favored by the labor group. It is generally believed that changes in the anti-trust laws will be necessary before any cooperative regulation within the industry can be effected.[5]

The decline in anthracite production has been slightly less than that in bituminous. Production figures for the first eight months of 1932 indicate a tonnage of 30,653,000, a drop of 23 per cent from the 1931 figure for the same period.[6] This branch of the industry has attempted to solve its own internal problems. A large group of operators have formed a marketing organization which, being attacked by the government on the grounds of restricting free competition, is serving as an experiment which the entire coal industry is watching carefully. Its success would open the way for more organizations of a similar type.

In the face of competition from fuel oil and gas, the anthracite industry is making a strong effort to conserve its market. Through research and development of new equipment, attempts are being made to endow anthracite with the same attributes of cleanliness, convenience, and efficiency possessed by fuel oil and gas. The efforts in this direction have been decidedly successful. It appears that before very long comparisons between the fuels must be made solely on the basis of costs.[7,8] The increased use of automatic equipment has created a greater demand for the smaller sizes of anthracite and has established a better balance between supply and demand for the various sizes produced.[9]

In New England foreign anthracite is a serious competitor. Under existing rates coal can be and is brought by boat from Wales, Germany, and even Indo-China to compete with that sent by rail from Pennsylvania. The foreign coal enjoys all the benefits of the research and development carried on by domestic producers to increase the demand for anthracite without contributing to its support.[10]

Research is under way in an attempt to find uses for anthracite

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[1] *Coal and Coal Trade Journal*, vol. 63, p. 286.

[2] *Coal Age*, vol. 37, p. 201.

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Compiled by T. A. Mangelsdorf, Assistant Professor of Fuel Engineering, Massachusetts Institute of Technology, Cambridge, Mass., and F. B. Broughton, Gulf Refining Company, Boston, Mass.

Personnel of the Executive Committee of the Fuels Division for 1932: R. A. Sherman, Chairman; William G. Christy, Secretary; A. D. Blake, W. J. Wohlenberg, F. M. Van Deventer, and K. M. Irwin. Associate: John Van Brunt.

[3] *Coal Age*, vol. 37, p. 223.

[4] *Ibid.*, vol. 37, p. 193.

[5] *Ibid.*, vol. 37, p. 200.

[6] *Ibid.*, vol. 37, p. 344.

[7] *Coal and Coal Trade Journal*, vol. 63, p. 342.

[8] *Ibid.*, vol. 63, p. 185.

[9] *Coal Age*, vol. 37, p. 221.

[10] *Coal and Coal Trade Journal*, vol. 63, p. 119.

other than as fuel. These other uses, if developed, will be of importance to the fuel industries to the extent that they divert a portion of the anthracite production from the fuel markets. It is reported that anthracite fines have been found superior to sand in water-softening beds, and an investigation is under way to determine the possibility of using anthracite as electrode carbon.[11]

The last year has witnessed an improvement in the position of crude petroleum resulting from the rigid production control set in operation in the fall of 1931. Strict enforcement of the proration laws in the East Texas area has overcome the demoralizing influence which this field exerted on the whole mid-continent territory. The producers now find themselves in the unusual position of enjoying a greater net return from a smaller volume of production.[12]

During the first seven months of 1932, 463,500,000 bbl of crude were produced, as compared with 498,600,000 in the same period of 1931. The result has been a very desirable reduction of crude in storage, although this has been achieved partly by the not so desirable increase in stored refined products. Considerable drilling activity has been manifest, and in these seven months 8464 wells were completed, an increase of 15 per cent. Better prospecting methods reduced the percentage of dry wells from 33 per cent to 23 per cent of the total completed.[13]

While the figures indicate a better statistical position, concern is felt for retaining the gains. Illegal production under the proration laws is hard to prevent and tends to depress prices. The maximum permissible production per well in the East Texas field has been raised recently, and there is some question as to whether the increase can be properly absorbed. The subject of proration has entered the realm of politics, which causes some uneasiness, but it is probable that the benefits are sufficiently evident to insure retention of some form of restriction of output.

"The Impelling March of Natural Gas," as one important series of articles dealing with this subject was entitled, has continued throughout the last year.[14] The changeover from manufactured to natural gas was carried through in the Chicago district. Although statistics indicate a drop in natural-gas sales of about 8 per cent, the extension of pipe lines and distribution systems has continued hardly undiminished.

Natural gas is becoming so widespread in its distribution that there is scarcely a section of the country that has not been influenced by either its advent or its approach. It is now a factor to be reckoned with in all branches of fuel utilization. It has affected the trends in boiler and industrial-furnace design. It has affected the solid-fuel market to such an extent that the Interstate Commerce Commission has given the railways authority to meet competition of Texas natural gas by reducing rates on industrial coal in the Middle West.[15] Such a move is without precedent. It has become so cheap in the Far West that water power has reversed its position, becoming a standby for steam generated by natural gas. Southern California Edison has abandoned some of its hydro sites and has evidenced a willingness to relinquish part of its share of the projected power from Boulder Dam.[16]

The realization that the economics of pipe-line transportation requires a high load factor has put natural gas in the position of usually carrying the base load. It has therefore been necessary to have available a substitute for it that has the same physical characteristics and that can be produced with high flexibility to carry the peaks. This demand has resulted in some new processes. Foremost is the "refractory-screen oil-gas process," which cracks low-grade oils on a refractory bed in a converted water-gas set. Deposited carbon is intermittently burned to supply the necessary heat.[17] There are also two catalytic processes, one continuous and the other intermittent, which utilize butane for the same purpose.[18,19]

The removal of hydrogen sulphide from natural gas is obtained in a new method by scrubbing with a solution of salt and lime. This process is of especial interest, since it has been shown that the addition of relatively small amounts of chemicals to oil-field water produces the equivalent composition of the salt-lime solution and permits elimination of the hydrogen sulphide at a relatively low cost.[20]

Coal Carbonization and Gas Manufacture

There have been few engineering developments in the manufacture of coke because of the uncertainty in that industry. By-product coke production has suffered severely due to decreased blast-furnace output. Likewise the carbonization of coal primarily for the production of gas has felt the effects of the expanding networks of natural-gas transmission lines. Improvements have been in operation rather than in design, since there has been little new construction. In some ovens flue gas is being recirculated in the heating flues to produce a longer flame and provide more uniform heating. There is a trend toward the utilization of blast-furnace gas for heating the ovens, where it is available. The successful utilization of coke breeze by recharging to the ovens has been reported.[21]

The gas industry in general has not been so adversely affected by the depression as have most industries. The decrease in total gas sales has been small and has been due principally to lessened sales of gas for industrial purposes. There have been actual increases during the year in the production of both oil-gas and reformed refinery gas. Both of these, however, still constitute a small percentage of the total.

The by-product situation, which has such an important effect on the economics of coal carbonization, is generally unsatisfactory. New markets are needed for the benzol, ammonia, naphthalene, and tar, the importance of which have been diminished by synthetic ammonia, new anti-knock fuels, and non-phenolic plastics. Some progress has been made toward finding new uses. One example is the comparatively new fertilizer produced by mixing ammonia liquor with superphosphate fertilizer. A use has been discovered for the ammonia thiocyanate, a recently developed by-product formed during the removal of cyanogen from the gas. It has been found that the application of this material to soil results in immediate sterilization, with the destruction of weed growth, but the subsequent decomposition of the material provides a latent fertilizer which prepares the soil for fresh vegetation.

Some interesting experiments are being carried out by the Detroit Edison Company on the carbonization of coal by electricity. The principal appeal of such a process is that it would furnish an outlet for off-peak power. A public utility producing both electricity and gas might find the process profitable if it becomes technically successful.[22]

The United States Bureau of Mines in cooperation with the American Gas Association has recently completed a comprehensive test program on the gas, coke, and by-product making proper-

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[11] *Coal and Coal Trade Journal*, vol. 63, p. 13.
[12] *Oil and Gas Journal*, vol. 31, no. 15, p. 7.
[13] *Ibid.*, vol. 31, no. 16, p. 30.
[14] *Gas Age Record*, vol. 69, pp. 329, 377, 431, 463, 493, 547.
[15] American Gas Assn. Natl. Gas Dept. Circ. no. 35, July 20, 1932.
[16] *Electric World*, vol. 100, p. 355.
[17] *Gas Age Record*, vol. 69, p. 737.
[18] *Ibid.*, vol. 69, p. 758.
[19] *Ibid.*, vol. 69, p. 683.
[20] *Ibid.*, vol. 69, p. 81.
[21] H. C. Porter, paper to appear in December or January in *Ind. and Eng. Chem.*
[22] *Gas Age Record*, vol. 70, p. 27.

ties of American coals. The results of this investigation will be of inestimable value to the carbonization industry.[23]

Steam Generation

Probably the major development in the field of steam generation is the increased competition between fuels, particularly between coal, oil, and nat ral gas, and the determination of operating engineers to use the cheapest fuel available at all times, whether it be one of the three mentioned or some fuel available locally—as, for example, petroleum coke or acid sludge. Fluctuating prices of the various fuels relative to one another has created a strong demand for combination burners that will handle coal, oil, or gas with few or no changes necessary when changing from one fuel to another. The demand for combination burners has not been limited to pulverized-coal installations. It has been found to be a comparatively simple matter to install combination oil-and-gas burners in stoker-fired furnaces, and a number of such installations have been made.

The trend in burner design has been toward larger capacity. Burners that will handle up to 100,000,000 Btu per hr using coal, oil, or gas are in successful operation. Improvements have been made to decrease draft loss across both natural-draft and forced-draft burners. This permits a higher capacity for a given draft with natural-draft burners or a reduction in power consumption for a given capacity in the case of forced-draft installations.

An important development in pulverized-coal firing has been a modification in the design and operation of ball mills to eliminate lag, thus making it possible to handle sudden changes in boiler output when these mills are used with the unit-fired system, without an appreciable change in boiler pressure.

Burners for refinery waste fuels such as acid sludge have been greatly improved. Suitable materials have been found, which together with changes in design permit the atomizer parts to give several thousand hours of continuous service without replacement, where formerly a hundred hours was about the best performance.

A new pneumatic transport system for pulverized coal or other pulverulent material has been developed during the past year and appears to have a number of decided advantages. Claims made for it are ease of installation, low first cost, negligible maintenance charges, and entirely automatic operation. It may be installed for any capacity or distance, using small-diameter standard pipe, and requires no more air per ton of material at low than at high capacity according to the statements of the manufacturer.

The trend toward the unit system of pulverized-coal firing continues. In the first nine months of the year 68 orders for unit pulverizers for steam generation were reported, with no reported orders for the central system.[24,25]

Several improvements in stoker design are noteworthy. Additional installations provided with zoned air control have been made, this system having demonstrated its usefulness and practicability under actual operating conditions. In one of the plants using zoned air control, it has been possible to operate at rates of firing up to 90 lb per sq ft per hr for long periods of time with air temperatures as high as 570 F.

A modification has been made in the design of multiple retort stokers permitting continuous discharge of the ash without the use of a clinker grinder or crusher. This design is especially applicable where head room is limited, since it requires less space than either the dump grate or clinker grinder. The general trend in ash handling is toward a finer clinker, partly for ease of handling the ash hydraulically and partly because of the greater use being made of ash for purposes other than fill. Clinker grinders to meet this requirement have been developed.

Improvements have been made in stoker drive mechanisms, both mechanical and hydraulic, so that it is now possible to obtain almost an infinite variation of speeds within the limits practicable for stoker operation. Stoker sizes continue to increase. The average capacity of those ordered during 1932 was 305 hp, compared with the 1931 average of 252 hp.

During the last few years there has been a decided renewal of interest in stokers, which may be attributed to several causes. First and most important is the enormous progress that has been made in their design to permit higher rates of combustion, larger sizes, and greater flexibility, and to insure greater reliability. A second factor is the realization that the advantages of pulverized coal have in many cases been overemphasized, due perhaps to the more or less revolutionary character of this method of firing and the glamor connected with the earlier stages of its application to steam generation. Still another reason for the increased popularity of stokers is the attention that is being focused on the elimination of stack discharge. Pulverized coal has certain advantages over stokers, particularly for the combustion of coals having a high ash content or a low fusing ash, and there is no danger that this method of firing will be displaced. It is probable, however, that the choice between stokers and pulverized coal in future installations will be made on the basis of sounder engineering judgment and with more attention being paid to the real merits of the two systems.

Several research projects of particular importance have been reported during the last year. The Bureau of Mines completed a study of the use of fluxes in slag-bottom furnaces and is continuing work on the problem of determining the types of coal satisfactory for successful operation.[26] Several excellent papers on the combustion of pulverized coal were presented at the Third International Coal Conference last winter which gave the results of important research in progress.[27] The Bureau of Mines is studying the combustion on grates of coal ignited from the top in an effort to clarify the principles involved in firing with underfeed stokers. The results of these investigations completed and in progress should be of great value in the design and operation of steam-generating equipment.

Industrial Furnaces and Kilns

During the last year the use of luminous flames for the dual purpose of increasing the rate of heat transfer and providing a non-oxidizing atmosphere has grown in popularity. This development has been facilitated by the extension of natural-gas distribution and the availability of liquefied petroleum gases, both of which are admirably suited to this type of combustion. Luminous-flame burners are being used for annealing sheet steel, heating copper billets, normalizing, and many other heating and treating operations.[28]

Important research on luminous flames is under way, and one paper on the determination of true temperature and radiation of luminous flames has been published within the year.[29]

There are no new important developments in fuel application in steel making or in cement manufacture. Efforts are being made, however, in the latter industry to increase the efficiency of rotary kilns—in some instances by preheating combustion air and the raw mixture, by supplying a drier mixture from a slurry filter, by spray feed, and by waste-heat boilers.

[23] U. S. Bureau of Mines, Bull. 344.
[24] *Power*, vol. 76, p. 50.
[25] Survey of Current Business, U. S. Dept. of Comm. 1932 monthly and annual supplement.
[26] A.S.M.E. Meeting, Dec., 1931 (Nicholls and Reid).
[27] Proc. 3d Intl. Conf. on Bit. Coal, vol. II.
[28] *Gas Age Record*, vol. 69, p. 573.
[29] *Ind. and Eng. Chem.*, Analytical Edition, 4, p. 166.

There have been some changes from pulverized-coal to natural-gas firing.[30]

Marine and Locomotive Developments

Probably the item attracting the most attention in the marine field is the use of colloidal fuel by the Cunard liner *Scythia* on one of her trips. Colloidal fuel was a development of the oil-conservation activities during the war, but not much has been done with it since then. It is not truly colloidal, but consists of finely pulverized coal suspended in heavy oil in the ratio of 40 parts coal to 60 parts oil. It has a heating value per pound intermediate between the two, and per unit volume a 3–6 per cent increase over oil. Because of the coal, the flame is highly luminous, while the operating ease of oil is retained. The marine field has always been considered its sphere of greatest applicability. It is reported that the trial was very satisfactory.[31]

Of interest also is the boiler installation planned for the proposed super-liners of the United States Lines. The boilers are believed to have greater unit output than any now afloat, yet the number of boilers is such that ample flexibility of power is possible. Rated power is developed with two of the 18 boilers out for cleaning. The total steam demand will be 1,650,000 lb per hr. The installation follows the trend ashore of fewer and larger steam-generating units.[32]

There have been no new developments in the railway-fuel situation. The railroads are continuing their efforts toward higher efficiencies, and the yearly figures will probably show a further reduction in coal per unit output. In 1931 there was about a 2 per cent efficiency gain in both freight and passenger services.

Motor Fuels

Developments in gasoline production have progressed steadily in the direction of better knock rating, increased volatility, and reduction of gum content.

The better knock rating has been obtained chiefly by increasing the percentage of cracked gasoline. In 1931 cracked gasoline constituted 39.8 per cent of the total produced, which increased to 42.4 per cent for the first half of 1932. So great has been the demand for cracked gasoline that additional cracking capacity has been installed, although refining operations in general are at somewhat less than two-thirds of capacity. In the new cracking installations the trend has been, as it has in many fields, toward larger units. One unit of 20,000 bbl per day capacity has been completed.[33]

The hydrogenation of petroleum, which has been mentioned in previous reports, has progressed considerably. Two large-scale commercial plants are in operation and are apparently entirely satisfactory. In April of this year one of the plants completed a run which had been in continuous progress for a year.[34]

The Knowles coking process for the carbonization of heavy petroleum residuals seems to be finding favor. Removal of the coke from the Knowles stills or ovens is by means of a mechanical pusher similar to that used for the removal of coke from by-product ovens. The product is an excellent domestic fuel, having a low volatile content and an ash content of around 1 per cent. It can be manufactured at a price to compete with anthracite.

Among new processes is that of treating with boiling zinc-chloride solution. The process is countercurrent with the oil in the vapor phase. The resulting product has high color stability and low sulphur and gum content. Costs are said to be quite low, and commercial application seems to be successful.[35]

The most important Diesel-fuel development is the production by several leading refiners of anti-knock oils. Surprising as it may seem, conditions which prevent knock in gasoline engines produce knock in the Diesel engine. The usual types of knock inhibitors are therefore harmful rather than beneficial in a Diesel fuel. Progress is being made in determining oil characteristics of importance to good operation, with special attention being paid to "ignitibility." There is still a need for satisfactory tests and specifications for Diesel fuels.[36]

Domestic Heating

Probably in no branch of the fuel industry has there been greater activity during the last year than in the field of domestic heating. Although a large percentage of the total fuel produced is consumed for this purpose, domestic furnaces have been permitted to languish at a low state of efficiency and convenience for many years. Increased competition between solid, liquid, and gaseous fuels and a belated realization that the domestic fuel market is the least affected by periods of low business activity have brought this field to the attention of both producers of fuel and equipment manufacturers.

The attempt of the anthracite industry to retain its hold on the domestic market by the development of automatic equipment has already been mentioned. Stokers have been developed for bituminous coal also, and there are now over 60 manufacturers of household stokers. There is no complete report available of the number of stokers sold, but a major manufacturer of feed-screws reports a 120 per cent increase in sales.

Oil burners have shown considerable improvement during the past year. Completely automatic burners which will handle the heaviest grades of fuel oil have been designed and should find application in apartment houses and office buildings. Several self-contained oil burners and furnaces have made their appearance. The construction of burner and furnace as a unit has permitted the incorporation of some desirable features which are not possible when the burner is designed to be used in a boiler previously fired with coal. Combined with the desirable technical features is a neatness of appearance which should insure their popularity.

One reputable manufacturer has ready for market a pulverized-coal unit for domestic boilers. Some 20 of these units were in operation during the winter of 1931–1932.

The American Gas Association reports an increase of 4 per cent over 1931 in the number of customers using gas for house heating and an increase of 2.8 per cent in gas sold during the first six months of 1932 as compared with the corresponding period in 1931. Installations include completely new units especially designed for gas and conversion burners which are used to adapt coal-fired boilers to gas firing.

A development which may completely change the economic situation of the fuel industries is the conditioning of air for homes, including cooling and dehumidifying during the summer as well as heating and humidifying during the winter. Since the increased demand for fuel would come during the summer, when activity in most lines of the fuel industry is now at a minimum, the new load would be of inestimable value. All branches of the fuel industry in addition to equipment manufacturers are actively engaged in exploring this new field. Some equipment is already on the market, and the predictions are that before many years

[30] *Cement and Cement Mfg.*, vol. 3, no. 11, and vol. 4, nos. 1, 2, 4, 5, 7, 9, 10, 11.

[31] *Power*, vol. 76, p. 58.

[32] T. E. Ferris, Meeting Soc. of Naval Architects, Nov. 19–20, 1931.

[33] *Refiner and Natl. Gas Mfr.*, vol. 11, p. 469.

[34] *Ind. and Eng. Chem.*, vol. 24, p. 1129.

[35] *National Petroleum News*, vol. 23, no. 45, p. 32E.

[36] P. H. Sweitzer, 5th National Meeting of Oil and Gas Power, discussion, A.S.M.E., June 8, 1932.

have passed, summer air conditioning for homes will be considered as essential as heating during the winter.

Of interest in the heating field is the application of the refrigeration principle, or heat-pump action, to furnish heat in winter and cold in summer to a Los Angeles office building. The same equipment does both jobs, apparently with economic satisfaction. The possibilities of this application are greatest in localities where the seasonal fluctuations are small.[37]

Smoke and Dust Abatement

The work of smoke elimination is being carried on. There are no outstanding developments in this field, but the continued watch on stacks and on the installation of new equipment is slowly effecting an improvement in conditions. Atmospheric pollution from dust and sulphur fumes is still a serious problem, and extensive research, hampered somewhat by the economic situation, is being carried on for its elimination.

The electrical precipitator has been found the best agent for dust removal, but its operation is expensive. Scrubbing methods, not yet entirely satisfactory, show promise of success. With these, corrosion is a very serious handicap, so that much of the research on scrubbers consists in studies of the life of materials of construction. The work on sulphur fumes, which are responsible for most of the corrosion, has not yet given much indication that they can be properly eliminated. There is, rather, a growing conviction that preparation of the coal is a more fertile field for the solution of the sulphur problem than is scrubbing the stack gases.

The question of adequate combustion space and the proper height of settings is being investigated in the interests of smoke prevention. A serious obstacle to any smoke studies is the unsatisfactory state of development of test methods and instruments. There is a real need of practical apparatus for measuring the smoke in the air and its effect on transmitted radiation.

Among interesting smoke studies has been the microscopic identification of smoke particles from different sources. It is claimed that the particles from different stacks have individual physical characteristics which definitely connect them with their source. On the basis of such studies it is said that nine-tenths of the smoke particles in the New York City atmosphere come from small fires in apartments, homes, and small office buildings.[38] Any improvement in conditions, if this is the case, would come slowly and only as a result of education of the public.

Engineering Employment

From a study of the employment situation in the fuel field, it is apparent that the industry has openings for men of particular ability only, whose past records give promise of achieving lower operating costs, improved salability of product, or development of profitable processes. While it is probable that the effect of the economic situation on most fuel industries has been less than in other fields, nevertheless the economic pressure has been sufficient to cause abandonment of a number of worthy but not immediately profitable enterprises. Consequently, the demand for men is mainly defined by the promise of quick profit.

Of the industries surveyed, the public utilities have shown, perhaps, the steadiest output, but the margin of profit to which they are restricted has required the exercise of caution in expansion. The metallurgical industries are especially sensitive to economic pressure. The best employment conditions seem to be in those industries where competition is intense and where new developments lead to widespread changes. Such industries are oil refining, gas- and oil-burning equipment manufacturing, both industrial and domestic, and air conditioning, among others.

With the revival of industrial activity, it is expected that the demand for engineers in the fuel industries will be greater than ever before. The depression has brought a better realization of the importance of fuel economy, which will be translated when conditions warrant into extensive investigations of the best method of performing any particular heating operation. This in turn will require research workers, designing and construction engineers, and operating engineers trained in the principles of fuel processing and utilization.

[37] *Power*, vol. 76, p. 29.

[38] E. E. Free, Meeting of Illuminating Group of A.I.E.E., March 7, 1932.

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Progress in Steam-Power Engineering

General

NEW installations during the past year have been few, since the depressed state of general business has brought about a reduction in the utilities' loads and no immediate need exists for increased capacity in either the utility or industrial field.

In general power-station design, the tendency toward the use of 1200- to 1400-lb steam pressure with the necessary reheat is not so strong. The tendency is more toward the use of pressures as high as can be practically used without reheat. One plant is now in operation using a pressure of 700 lb and 850 F temperature and no reheat. Still higher pressures and temperatures are imminent, as large turbine builders declare themselves ready to build equipment for 1000 lb pressure and 1000 F temperature without reheat. The tendency is, then, toward simpler design than is accomplished with the higher pressure and reheat.

The special high-temperature turbine being tested by The Detroit Edison Company was actually operated for considerable periods of time at 1000 F temperature for the first time during this year.

There is increasing activity among the larger industrials toward the consideration of by-product power production, with or without the cooperation of the utility companies.

Developments in the use of higher pressures and temperatures have stimulated the idea of "rehabilitation" of older plants by the superposition of a high-pressure boiler and turbine in connection with existing equipment. In some cases, notably at the Burlington Generating Station of Public Service Electric and Gas Company, this expedient has, at a comparatively moderate cost, converted what was a stand-by peak-load plant into the most efficient on the system. Such a rehabilitation may also be accomplished by the use of the mercury-vapor system in conjunction with existing equipment. This may in some cases even be more attractive than the high-pressure steam system.

The tendency toward further simplification and cheapening of building design continues. Some utilities have given serious study to the development of designs omitting the conventional building to the utmost extent possible. However, these studies do not at present give any general accepted answer to this question. In certain specific cases the cost of such an "outdoor" power plant is apparently not materially less than a conventional but inexpensive building.

In order to test the possibilities of this type of plant, the General Electric Company has now under construction an "outdoor" installation at the works in Schenectady. This is a full-sized power plant providing a 20,000-kw mercury turbine and also a steam boiler. Its operation will be watched with considerable interest, as it should indicate the operating difficulties to be encountered with outdoor construction in a fairly severe climate.

There are still comparatively few plants built with only one boiler per turbine. However, with the greater reliability of boiler and fuel-burning equipment which modern design provides, the tendency toward this ideal is more pronounced. Three boilers for two turbines seems to be the minimum seriously considered at present, except in cases possessing exceptional transmission-line interconnection advantages.

The Bureau of Standards has completed its determination on the temperature-pressure relations of saturated water, and the results of this important experimental work are being made available to steam engineers through the A.S.M.E. Transactions.

Boilers

No important changes in boiler design have appeared, the tendency being rather to consolidate the new ideas of the last few years and to improve details. The use of fusion-welded drums has practically supplanted riveted construction.

More attention is being given to means and devices to prevent the carry-over of solids with the steam from the boiler. One manufacturer has developed a steam washer in which the steam, before leaving the drum, passes through a spray or rain of incoming feedwater. The elimination of solids from the steam passing to the superheater is of greater importance than is the elimination of moisture below 0.8 to 1 per cent.

The increase in boiler exit-gas temperature resulting from increasing steam pressure and rates of output per square foot of surface has made the use of economizers in addition to air preheaters more essential. Economizers when used are generally of the steaming type because of the high temperature of the feedwater coming from the regenerative feed-heating system. The trend in economizer design is toward the omission of all joints, as they have in many cases been the greatest single source of trouble. The latest designs proposed by the manufacturers consist of practically continuous welded loops.

Boiler Furnaces and Fuel Burning

Water walls have become almost universal for all larger units. Progress has been made in the design and construction of so-called wet-bottom or slag-tap furnaces. Furnace shapes and burner designs more suitable for this type of furnace have been developed. The use of water-cooled furnace floors has greatly reduced the operating difficulties present with this type of furnace. However, the "dry" versus "wet" furnace is still a moot question, each type having its apparent advantages and disadvantages.

Stokers and pulverized-fuel equipment have undergone no radical changes of design in the past year.

An inherent handicap against the use of pulverized fuel, at least in populated districts, is the cost of providing satisfactory fly-ash eliminators. The matter of fly-ash and cinder elimination is receiving more attention than heretofore. The electric precipitator seems still to lead in general effectiveness. However, a type of wet scrubber has been developed and used by one power company which indicates a degree of removal of the same order as that of the electric precipitator.

Cinder emission from stoker plants is less difficult, and both dry and wet type washers are used, depending somewhat upon the requirements of location. A large majority of the steam-power plants throughout the country are still without eliminators of any kind.

Stokers suffer somewhat from a certain limitation in kinds of fuel that may be used successfully and the increasing difficulty of providing sufficient grate area under very large boilers without undesirable complications in design.

The changing market prices of fuel oil and gas have increased the use of these fuels in steam-power plants. A number of coal-

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burning plants have installed oil-burning equipment. Most of the plants that have taken advantage of the economies at present available through this conversion have been those designed for pulverized fuel, because of the relative simplicity and small expense of making the change in a pulverized-fuel furnace.

Sponsored research at Purdue University has apparently borne fruit in the development of a fairly simple process for the manufacture of building brick from pulverized-fuel ash. This scheme contemplates the setting up of equipment to manufacture the brick at the power station where the ash is produced.

Turbines and Generators

The tendency in large turbo-generator design is distinctly toward single-shaft machines. The largest of these so far installed are two 165,000-kw tandem machines in the Hudson Avenue Station of the Brooklyn Edison Company. The order has been placed for a similar unit for the Richmond Station of the Philadelphia Electric Company. No increase in steam pressure over that of the previously existing equipment was made in either case.

Research work on the design and materials of low-pressure blading has been carried on in an effort to minimize the wear due to moisture. Special steels, chromium plating, and stellite protection have been tried. At the present time stellite seems to offer the greatest promise.

Fabricated welded-steel construction is to a large extent replacing the use of castings. Welded-steel construction has not yet been used for the cylinders of large turbines. A number of small turbines have been built in this way, and its use for large units is receiving serious consideration.

Motor-operated turning gear to keep the machines slowly rotating when out of service for comparatively short periods are now almost invariably used on the larger units. The rotation serves to eliminate inequalities of temperature in the turbine rotor and thereby permit more rapid starting. On the larger units auxiliary high-pressure oil systems are being provided to float the rotating element before starting from rest and thus avoid damage to the bearings.

In general, the progress in turbine design has been toward greater refinement of detail and increase in size of single-shaft units.

Generator efficiencies have increased somewhat, but the margin still available leaves but little room for further improvement.

Condensers

It appears that the use of fusion-welded steel shells for condensers will soon entirely supplant cast-iron construction. The 65,000-sq ft condenser for Kearny Generating Station is the largest so far built of welded steel. The rolling of tubes in both tube sheets is becoming common practice. Vertical circulating pumps are gaining in favor.

Power-Station Piping

Probably the greatest progress in steam-power engineering that has occurred within the past year has been the increasing use of fusion welding in the fabrication of the piping systems. In the latest work cast fittings have almost entirely been eliminated and flanged joints reduced to a minimum. Valve bodies of considerable size have been welded directly into the pipe lines. This is unquestionably a distinct and important step toward greater reliability and decreased maintenance.

Special Cycles

Two installations of the mercury-vapor cycle are now going forward—one at General Electric Works in Schenectady and the other at Kearny Generating Station of Public Service Electric and Gas Company of New Jersey. Each will generate 20,000 kw from the mercury turbine and supply sufficient steam to generate approximately 30,000 kw in steam turbines. The Kearny installation is rapidly nearing completion. The cost of this plant only slightly exceeds that of a high-pressure steam installation, and the heat rate will be about 20 per cent less. The principal new development in this installation compared with that at Hartford, Conn., other than an increase in capacity, is the use of mercury-cooled furnace-wall construction. Commercial operation of the Hartford mercury boiler and turbine unit continues, and a net efficiency of slightly less than 10,000 Btu per kw-hr has been maintained. This is a new economy record for this country.

The use of diphenyl oxide has been considered for a binary cycle, but no actual progress is known to have been made along this line. One installation has been made using diphenyl oxide as a heat carrier to convey heat absorbed from the flue gases to an air preheater, thus eliminating a large amount of cumbersome and expensive duct work.

Your committee has no information as to any special progress with the Benson or Loeffler cycles, though operation and experiment continue in Europe.

FSP-55-3a

Steam-Turbine-Plant Practice in the United States

By VERN E. ALDEN[1] and W. H. BALCKE,[2] BOSTON, MASS.

From the great volume of available data on the design, equipment, service requirements, and operating results of the steam-electric-generating stations serving the larger power systems in the United States, the authors have endeavored to present a cross-section of existing turbine-plant practice. The principal subjects discussed include the selection and arrangement of steam turbine-generators, their condensers, and auxiliaries, together with an analysis of published test results for many large turbines and the service-reliability records of modern installations. While some economic considerations and other controlling factors underlying the principal features are outlined, the purpose of the authors is that of describing existing practice rather than of offering any recommended solution for the many problems involved. To assist in recognizing typical rather than exceptional practices, and widespread rather than personal or minority opinions, advantage has been taken of comments and suggestions from many individuals closely associated with the development of the art.



W. H. Balcke



V. E. Alden

AMERICAN steam-turbine-plant practice has in the main been dominated by the desire to gain four objectives:

1 Improvement in reliability in order that generating capacity may, to the maximum extent possible, be available for service when needed
2 Reduction in fuel consumption per kilowatt-hour of net station output
3 Reduction in the overall investment per kilowatt of system peak load carried
4 Reduction in cost of operating labor and maintenance.

A very large percentage of all the new steam-turbine-plant capacity installed since 1920 has gone into plants serving the electric light and power systems. Practice in industrial power plants has usually followed or paralleled that in utility plants. The rapid growth in load and the extensive use of interconnections have had a most important influence as they have helped to justify the use of turbine-generators of increasing capacity.

Increases in Steam Pressure and Temperature and Capacities of Turbines

The most significant trends are the increase in the size of turbine-generators and the increases in steam pressure and temperature. The graphs shown in Figs. 1, 2, and 3, somewhat modified from the form as presented in an earlier paper,[3] may help to visualize the rate of progress. The cross-hatched areas show the limits within which representative steam-station practice has fallen. An attempt has been made to indicate what might be called conservative practice, or trend, by means of a curve drawn through each cross-hatched area. The horizontal lines extending from 1928 to 1932 in Figs. 2 and 3 indicate the views of several schools of thought as to the best steam pressure and temperature.

Reheating

Another important development has been the use of a single stage of reheating to raise the temperature of the steam, after partial expansion, to its initial temperature, or slightly higher. This development dates from the initial operation early in 1925 of two 40,000-kw turbine-generators in the Philo Station of the Ohio Power Company, with steam delivered to the turbine throttle at a pressure of 550 lb per sq in. gage and 725 F, the steam being withdrawn from the turbine after being expanded to 155 lb per sq in. gage, returned to the boiler plant, reheated to 725 F, and then expanded through the remaining turbine stages to the condenser. Since this initial development there has been installed, or is now in the course of construction, in 19 generating stations in the United States a total of 2,800,000 kw of turbine

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] "Rehabilitation of Steam Power Plants," by C. F. Hirshfeld, vol. 48, Quarterly Transactions, American Institute of Electrical Engineers, Oct., 1929.

capacity for operation on the reheating cycle. All of this capacity is for operation at initial steam pressures of 550 lb per sq in. gage or higher, and 1,150,000 kw is for operation at pressures of 1200 lb per sq in. gage or higher. For approximately 80 per cent of this capacity flue-gas reheaters are used, and for the remaining capacity high-pressure steam reheaters are used. Where high-pressure steam reheaters have been used, the reheat temperature has been limited to approximately 500 F. Approximately 30 per cent of the capacity of all turbines 20,000 kw and larger placed in service and ordered since January 1, 1925, have been designed for operation on the reheating cycle.

During the past five years there has been an appreciable

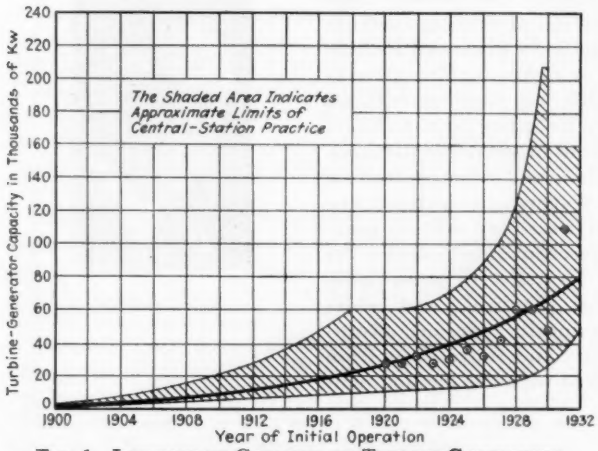


FIG. 1 INCREASE IN CAPACITY OF TURBINE-GENERATORS
(Graph shows both single and multi-cylinder units; circles are average turbine capacities from table at bottom of Fig. 5.)

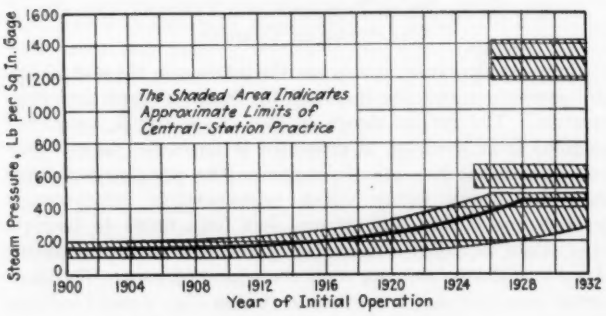


FIG. 2 INCREASE IN STEAM PRESSURE

amount of sporadic operation of large turbines with steam at temperatures from 750 to 800 F. The limit of good practice was generally considered, however, to be 750 F at the throttle. Orders were placed in 1929 with three manufacturers for turbines to operate with steam at temperatures from 800 to 850 F. An experimental installation of a 10,000-kw tandem-compound condensing turbine is now in operation with steam temperatures ranging up to 1000 F at 390 lb per sq in. gage in the Delray No. 3 Station of the Detroit Edison Company.

OPERATION ON NON-REHEAT CYCLE AT 650 LB PER SQ IN. AND 825 F

Some considerable amount of satisfactory experience has been had in the past five years with large turbines operating on a non-reheat cycle with steam delivered to the throttle at pressures 550 to 600 lb per sq in. gage and temperatures of 675 to 725 F. The percentage of moisture in the turbine exhaust is about the same for this condition as with steam delivered to the turbine throttle at 200 lb per sq in. gage and 480 F, an operating condition with which we have had considerable experience in the older stations. There has been some weight of opinion to the effect, however, that for these operating conditions the friction losses and blade erosion in the low-pressure stages, resulting from the relatively high moisture content of the steam, were too high for good practice. The raising of the steam temperature to the still higher value of 825 F makes it possible with a steam pressure of 650 lb per sq in. gage and no reheat to limit the moisture in the steam at the exhaust to the somewhat more reasonable value of approximately 12.5 per cent. This combination, having none of the complications of reheat, is a most attractive one. The new Sheboygan Station of the Wisconsin Power and Light Company and the extension to the Burlington Station of the Public Service Electric and Gas Company of New Jersey are to operate with these steam conditions. The somewhat older Michigan City Station of the Northern Indiana Public Service Company operates with steam delivered to the 60,000-kw single-cylinder turbine at 650 lb per sq in. gage and 750 F.

MULTI-STAGE FEEDWATER HEATING

One of the most striking developments is the revival and almost universal acceptance since 1924 of the practice of multi-stage

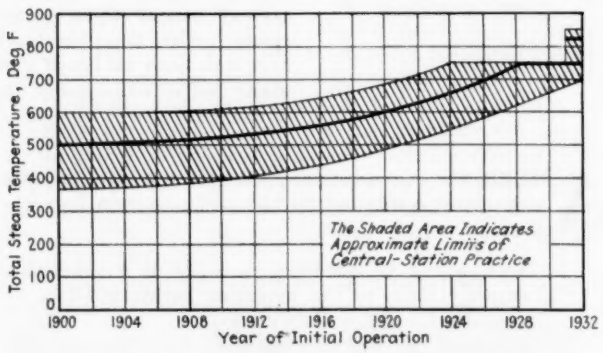


FIG. 3 INCREASE IN STEAM TEMPERATURE

feedwater heating with steam extracted from the main turbines at a number of points during its expansion. It has become common practice to extract steam from the main turbines at from three to five points and to heat the feedwater to temperatures ranging from 250 to 425 F, depending on the initial throttle steam pressure and the extent to which an economizer is depended upon to reclaim heat from the boiler-flue gases. This use of multi-stage feedwater heating places the generating station on a regenerative cycle, and insures the generation of the maximum amount of by-product electrical energy (at an overall heat rate of approximately 4200 Btu per kw-hr) from the steam extracted from the turbine at each of the different pressures. Aside from its marked effect in reducing the amount of heat rejected to the condenser circulating water and thus improving the overall station efficiency, the use of multi-stage heating of feedwater with steam extracted from the main turbines has had a most important effect, as outlined later, in making it possible to increase the maximum capacity obtainable in single-cylinder turbines.

TREND TOWARD USE OF HIGH-CAPACITY TURBINES

The marked trend toward the use of turbine-generators of higher capacity is shown by Fig. 1. It is true in general that the larger utility companies have been willing to purchase units of higher and higher capacities as soon as the manufacturers have carried their development work forward so as to permit them to build these larger machines. Table 1, showing for 14 large sta-

TABLE 1 CAPACITIES OF TURBINE-GENERATORS INSTALLED IN 14 LARGE STATIONS

Dates indicate year of initial operation. Capacities are given in kilowatts. Types of turbine construction are indicated as follows: s, single-cylinder; te, tandem-compound; ce, cross-compound; shp, single-cylinder turbine for expansion of steam from 1200 lb gage to pressure at which it is delivered to larger low-pressure turbines

	1921	1922	1923	1924	1925	1926	1927	1928	1929	1930	1931
Charles R. Huntley Station, Buffalo, N. Y.[a]	2 35,000-s 40,000-te 60,000-ce 2 20,000-s	40,000-te 60,000-ce 30,000-s	35,000-s	2 35,000-s 30,000-s 30,000-s	2 50,000-ce 30,000-s 50,000-te 50,000-s 60,000-ce	60,000-s		75,000-s		75,000-s 80,000-c	80,000-s
Hell Gate Station, New York, N. Y.								160,000-ce	165,000-ce		
Colfax Station, Pittsburgh, Pa.								2 41,250-s			
Lakeside Station, Milwaukee, Wis.						30,000-s 7,000-shp		60,000-s	7,700-shp	60,000-s 2 7,700-shp	
Cahokia Station, East St. Louis, Ill.			35,000-s	2 35,000-s	40,000-s			50,000-s	75,000-s[b]		115,000-te
Waukegan Station, Waukegan, Ill.			25,000-s		35,000-s		50,000-te			65,000-te	2 160,000-te
Hudson Ave. Station, Brooklyn, N. Y.				50,000-te	50,000-te	80,000-ce		110,000-ce		110,000-ce	
Crawford Ave. Station, Chicago, Ill.				50,000-s 60,000-ce	50,000-ce 52,000-ce	77,000-ce	91,500-ce	104,000-ce			
Philo Station, Philo, Ohio				40,000-s	40,000-s				165,000-ce		
Long Beach Station, Long Beach, Calif.				37,800-s	37,800-s	52,800-s		94,000-te		94,000-te	
Charles Leavitt Edgar Station, Boston, Mass.					2 32,000-s 1 3,150-shp	65,000-s 10,000-shp			12,500-shp		
East River Station, New York, N. Y.						2 60,000-s			160,000-te		150,000-tc[c]
State Line Station, Hammond, Ind.									208,000-ce		125,000-tc[c]
Powerton Station, Pekin, Ill.								55,000-te	55,000-te	105,000-te	105,000-te

[a] Three 20,000-kw single-cylinder turbines were installed in 1916 and one 35,000-kw single-cylinder turbine in 1918 in the Charles R. Huntley Station.
[b] This 75,000-kw turbine took the place of the first 35,000-kw turbine installed in Cahokia Station in 1923.
[c] These two units for State Line Station will be placed in operation in 1932.

tions the units installed each year, indicates the pressure which the utilities have brought to bear on the manufacturers to develop the higher-capacity units.

Reasons for using the larger units have been lower investment per unit of installed capacity in the turbine-generator itself and in the related equipment and structures, slightly lower fuel consumption, and lower costs for operating labor and maintenance per unit of station output. The reduction in the investment per unit of installed capacity incident to the use of larger turbine-generators is in part offset by the investment for spare capacity, which is increased as the capacity of the largest turbine unit increases. A controlling reason in connection with stations built in metropolitan areas has been the desire to install the maximum possible capacity per square foot of expensive real estate.

The capacity of the largest turbine-generator installed has, as an extreme case, been as high as 120 per cent of the peak load for some of the smaller systems. For the larger systems it has been considered good practice to install units having a capacity from 10 to 25 per cent of the system peak load. Where interconnections with other systems have been adequate and the prospects have been good for rapid load growth, the tendency has been to increase the size of the generating units. An example which serves to exemplify this trend toward the use of very large units is the initial installation in the new 1200-lb Port Washington Station of the Milwaukee Electric Railway and Light Company of one 80,000-kw tandem-compound turbine-generator receiving its steam from one boiler. In few, if any, cases have the utility companies departed from the practice of providing sufficient installed generating capacity to permit carrying the peak load with the largest unit out of service, and for the larger systems conservative operating practice has required a somewhat greater margin of installed generating capacity. The law of diminishing returns is coming into play as relating to the savings in fixed charges and operating costs resulting from the use of larger generating units, and it is doubtful whether justification can be found in the near future for the use of turbine-generators appreciably in excess of 200,000 kw capacity, even on the largest systems. It is also doubtful whether many generating units of a capacity less than 30,000 kw will be installed in the future by the electric light and power companies under discussion.

Multi-Cylinder Turbines

The marked trend toward use of multi-cylinder construction for the larger turbine units is indicated by Table 1, which shows 20 cross-compound turbines,[4] 18 tandem-compound turbines, and 35 single-cylinder turbines. The advantages of multi-cylinder construction are:

1 The lower leaving loss or residual kinetic energy in the steam resulting from the use of two or more sets of low-pressure blades, each having its own exhaust to the condenser, a design feature usually embodied in multi-cylinder turbines.

2 For a cross-compound turbine, the ability to operate the high-pressure section of the turbine spindle at high speed and the low-pressure turbine spindle at lower speed. This in turn permits a smaller diameter for the high-pressure casing and in other respects results in a more favorable mechanical design.

3 Where the generator design has limited the capacity of the unit, it has been possible to raise the maximum capacity by going to cross-compound construction with two or more generators.

4 Cross-compound construction has been advantageous in permitting the installation of very large units in an existing turbine room of limited width.

5 With multi-cylinder construction only a portion of the

[4] The turbines for operation at 1200 lb per sq in. gage in the Edgar Station and the Lakeside Station are considered as compounded with the low-pressure turbines.

temperature drop, due to complete expansion of steam, occurs across each cylinder. This minimizes troubles from expansion of mechanical parts and tends to make it possible to start the turbine from a cold condition in less time.

6 Operating experience has tended to indicate that for a given capacity, with all other conditions the same, a multi-cylinder turbine will be available somewhat more hours per year than will a single-cylinder unit of the same capacity.

Single-cylinder turbine construction has, in general, the advantage of lower weight and lower investment in the turbine with its related equipment and structures for a given capacity. It tends toward more capacity per square foot of floor space in the turbine room, and undoubtedly does have an advantage in this respect over a tandem-compound machine of the same capacity. It is doubtful, however, if this advantage exists when the contrast is made between large single-cylinder turbines and large cross-compound turbines. The steeple cross-compound construction

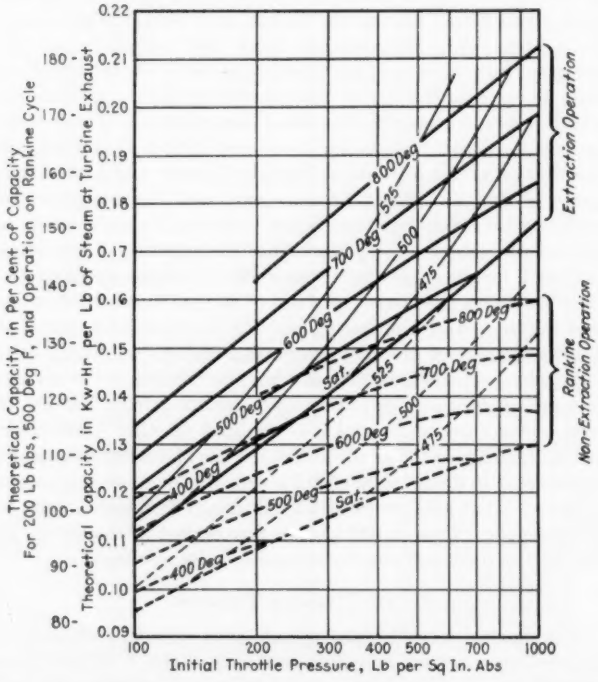


FIG. 4 THEORETICAL CAPACITIES FOR OPERATION ON EXTRACTION AND NON-EXTRACTION STEAM CYCLES AT VARYING STEAM PRESSURES AND TEMPERATURES

(Curves are for various initial total steam temperatures in deg. Fahr. as indicated. Back pressure 1 in. Hg. Specific volumes of exhaust steam in cu ft per lb may be determined by interpolation between constant volume lines. Curves are based on Goodenough, 1917.)

is extremely economical of floor space. Two 25,000-kw, two 50,000-kw, and one 110,000-kw turbines of this type are in operation at 1200 lb per sq in. gage. The high-pressure turbine and its generator are, in the case of the four smaller machines, mounted on top of the generators for the low-pressure turbines. In the case of the larger unit a 55,000-kw high-pressure turbine and its generator are mounted directly on top of the 55,000-kw low-pressure turbine and its generator.

SINGLE-CYLINDER TURBINES

Low-pressure blade areas and exhaust areas to the condenser are limited for single-cylinder machines, and this in turn either places a limit on capacity or makes it necessary to accept a higher leaving loss than that which can be rather readily obtained in a multi-cylinder design. The use of unidirectional double-flow exhaust blading, as embodied in the latest 60,000-kw turbine installed in the Lakeside Station in Milwaukee, serves to increase the low-pressure exhaust blade area which can be obtained in a single-cylinder turbine. In general it has been the experience in the United States that the manufacturers have first built machines for the higher capacities as multi-cylinder machines, and later, as they have gained more experience and have had available better materials and better shop facilities, they have built single-cylinder turbines for these same capacities.

The capacity of a steam turbine of given dimensions is limited by the ability of the last row of spindle blading to pass steam at velocities which will correspond to a reasonable leaving loss. It is not generally realized how rapidly the potential capacity of a turbine rises with increase in steam pressure, increase in steam temperature, and particularly as the low-pressure stages are unloaded and more work is shifted into the upper stages of the turbine by the extraction of steam from the main turbine for multi-stage heating of the feedwater. Fig. 4, taken from an earlier paper,[5] shows the magnitude of this effect. This increase in potential turbine capacity is much more rapid than the increase in turbine-plant efficiency. Fig. 4 shows clearly the way in which increases in steam pressure and temperature and the use of the regenerative cycle have facilitated the building of larger and larger single-cylinder turbines and have actually tended to reduce the cost of modern turbines.

There is a considerable group of engineers that have come to believe that for maximum load, wintertime conditions of operation, and a 29-in. Hg vacuum at the exhaust, a leaving loss ranging from 5 to 6 per cent of the heat available in the steam on the basis of adiabatic expansion is none too high. This is particularly true for the reason that the leaving loss for maximum-load summertime conditions of operation will, with the much lower specific steam volumes corresponding to lower vacuum for this same machine, be of the order of 2 per cent, and the average leaving loss for the year resulting from operation at partial loads and with vacuums corresponding to average circulating-water temperatures throughout the year will be of the order of 1.5 per cent. This willingness to accept turbines having a leaving loss approximately twice as great as that which has been considered in some cases to be good practice has made it possible to increase the maximum capacity of single-cylinder turbines by perhaps as much as 40 per cent. The acceptance of a higher per cent leaving loss is usually accompanied by the reduction on a per cent basis in the high-pressure packing and blade-tip leakage losses, the disk-friction losses, and other losses usually associated with the head end of the turbine, so that the decrease in overall efficiency is not as much as might be expected even for wintertime operation.[6]

During the last five years there have been installed nine turbines of 60,000 kw capacity, one of 65,000 kw capacity, five of 75,000 kw capacity, and two of 80,000 kw capacity, all single-cylinder machines. At least one of the manufacturers has expressed willingness to build single-cylinder turbines for operation at 1800 rpm and steam conditions of 650 lb per sq in. gage and 825 F at the throttle for a capacity of 100,000 kw.

No single-cylinder turbines for the expansion of steam from a pressure higher than 650 lb per sq in. gage to vacuum have been built in this country, and it appears unlikely that such machines will be built in the near future. Throttle pressures in excess of 650 lb per sq in. gage and steam temperatures in excess of 825 F seem to call for multi-cylinder construction. In general, it ap-

[5] "The Margins of Possible Improvement in the Central-Station Steam Plant," by Ernest L. Robinson, Trans. A.S.M.E., 1923, p. 644, paper no. 1913a.

[6] For a more complete discussion of the relation of leaving loss to the economic rating of a turbine, see paper "Some Recent Developments in Large Steam-Turbine Practice," by K. Baumann, Journal of The Institution of Electrical Engineers (London), June, 1921.

pears preferable to use multi-cylinder construction for the reheating cycle. There are, however, in operation in the Philo Station, the Twin Branch Station, and the Stanton Station, six 40,000-kw single-cylinder condensing turbines operating successfully on the reheating cycle.

Turbine Speeds

Most of the large turbine-generators supply power at a frequency of 60 cycles and operate at 1800 rpm. Some consideration was given a number of years ago to the construction of large turbine units for operation at 1200 rpm, but most of the recent development work has been done at 1800 rpm for the reason that this resulted in more compact and less expensive machines. Quite recently the manufacturers have developed fair-sized 3600-rpm turbine-generators.[6] Three 15,000-kw single-cylinder, 3600-rpm turbines with 18,750-kva generators were placed in operation in 1930 in the Baton Rouge Station of Louisiana Steam Products, Inc. These turbines expand steam from a throttle condition of 600 lb per sq in. gage and 725 F to a pressure of 141 lb per sq in. gage at the exhaust. Two 15,000-kw tandem-compound, 3600-rpm condensing turbines with 18,750-kva generators were placed in operation in 1931 in the Bremo Station of the Virginia Public Service Company. At least one of the manufacturers is prepared to build a 3600-rpm single-cylinder turbine of 15,000 kw capacity to operate under steam conditions as high as 650 lb per sq in. gage, 825 F, 28½ in. Hg vacuum, and this same company will build a tandem-compound machine for the same steam conditions to deliver a normal capacity of 30,000 kw with overloads up to 37,500 kw. The incentive for this 3600-rpm development is lower cost and greater compactness.

Developments in Generator Design

Generator design has to an appreciable extent in the past limited the capacity of turbine-generators that the manufacturers could build. At least one manufacturer is now prepared to build a 250,000-kva, 80 per cent power factor generator for operation at 1800 rpm and a 35,000-kva, 80 per cent power factor generator for 3600 rpm. The developments which have made it possible in the last seven years to quadruple the capacity of a single 1800-rpm generator constitute a separate story and have been covered by earlier papers.[7] The almost universal use since 1924 of the closed system of ventilation with finned-tube surface air coolers has had a most important influence on generator design as it has insured clean air. It has had another important influence on turbine-plant design as it has reduced the noise. The use of the closed system of ventilation has very much facilitated the rapid smothering of a fire resulting from an insulation breakdown. Generators built for capacities higher than 75,000 kva at 1800 rpm have been equipped with external ventilating fans. One manufacturer is now building a 99,000-kva generator and has indicated that he is willing to build a 150,000-kva, 1800-rpm generator with internal fans. The use of hydrogen for cooling generators would permit of still higher capacities. Several large turbine-generators have been built with provision for future use of hydrogen cooling, but none are now in operation with that type of cooling. A number of the larger generators are designed for 22,000 v. Six generators ranging from 83,333 kva to 166,667 kva have been built with double windings, three of these generators being wound for 22,000 v. The higher voltage and the double-winding feature make the problem of generator switching somewhat easier. An alternate solution is the use of an auto-transformer connected solidly to the generator leads.

It would appear that the manufacturers have caught up by their development work with the desires of the utility companies and are now prepared to build any capacity of turbine-generator that we may reasonably expect can be justified by economic studies.

Operation of Generators as Synchronous Condensers

Provision has been made in many stations for uncoupling the generators from the turbines and operating them as synchronous condensers. This procedure makes it possible, during the time when the turbine is being overhauled, to carry a substantial block of reactive kva on its generator, leaving the capacity of the other turbine-generators available for carrying true power load. It is also advantageous to operate the generators at load centers as synchronous condensers on systems that have large amounts of water power available for transmission into these load centers at certain seasons of the year.

Obtaining the Maximum Capacity Inherent in Turbine-Generators

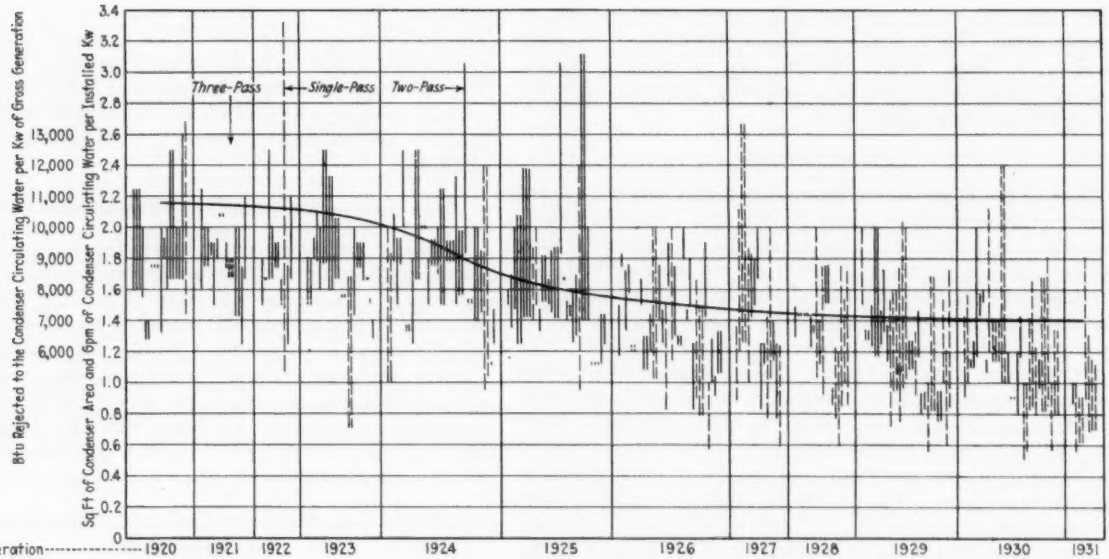
As operating engineers have more fully appreciated the fact that for most stations the fixed charges allocated to each kw-hr of output are appreciably more than the fuel cost, they have striven to reduce the fixed charges by planning operations so as to require less reserve capacity, while still providing adequate protection for the load, and in addition they have attempted to more effectively use all the capacity inherent in the equipment. It has been found in a number of cases that from 10 to 20 per cent of capacity in excess of the normal rating was available in both the turbine and the generator, even for summertime conditions of operation. In many such cases the turbines have been rated up to higher capacities for normal operation.

The peak load for the large systems usually occurs late in November or during December. For a typical case the power factor at the time of the peak load is 92 per cent or higher. Circulating water for the generator air coolers and the condenser is at a temperature of 45 F or less. This set of operating conditions makes available at the time of the system peak 5 per cent more turbine capacity and at least 25 per cent more generator capacity than for August operation with a system power factor of 85 per cent. In some cases operating engineers have availed themselves of at least a part of this surplus generator capacity by shutting off steam to the high-pressure feedwater heaters at the time of the system peak load and sending this steam on through the turbine to the condenser. In order to carry out such a procedure, it is necessary to have sufficient boiler-plant capacity to carry the increased load at somewhat reduced efficiency. Operating engineers are very generally using temperature recorders as guides in determining safe loads that can be carried on the generators. Surplus generator capacity for wintertime operation at the high power factor coincident with the system peak load is now being taken into account by designing engineers in proportioning the relative capacities of the turbine, the generator, and the boiler plant.

The lighting load is of such magnitude in certain of the systems serving metropolitan centers that the peak loads occurring as the result of thunderstorms in the summer present a difficult operating problem, especially since they may occur when one or more turbine units are being overhauled. In such cases generators and surface condensers of more liberal design are being installed in order to provide the required amount of summertime capacity.

It has been found advantageous in some stations to make the necessary changes in physical equipment to permit of using the

[7] "Recent Improvements in the Construction of Large Turbine-Generators," by S. L. Henderson and C. R. Soderberg, A.I.E.E. Quarterly Transactions, vol. 47, April, 1928, p. 549. "Design Features That Have Made Large Turbine-Generators Possible," by W. G. Foster and M. A. Savage, A.I.E.E. Quarterly Transactions, vol. 49, January, 1930, p. 60.



Initial Year of Operation	1920	1921	1922	1923	1924	1925	1926	1927	1928	1929	1930	1931
Three-Pass Installations	-	2	1	1	-	1	-	-	-	-	-	-
Two-Pass Installations	17	13	8	19	31	29	24	7	11	16	12	1
Single-Pass Installations	1	0	1	2	4	2	10	8	7	13	19	7
TOTAL INSTALLATIONS	18	15	10	22	35	32	34	15	18	29	31	8
Installed Kilowatts (1000)	500	417	330	605	1,078	1,187	1,158	678	1,091	1,752	1,503	885
Average Turbine Capacity in Kw	27,800	27,800	33,000	27,500	30,800	37,000	34,000	41,800	60,600	60,500	48,500	110,000
Capacity and Type	60,000 cc	60,000 cc	60,000 cc	37,500 tc	60,000 cc	52,000 cc	80,000 cc	91,500 cc	160,000 cc	208,000 cc	110,000 cc	160,000 tc
Largest Turbine in Kw: Avg Area in Sq Ft per Installation	43,600	46,700	50,800	42,200	45,000	50,000	37,300	42,500	62,500	53,200	44,500	76,500
Sq Ft of Area in Largest Condenser	4-25,000 two-pass	4-25,000 two-pass	4-25,000 two-pass	62,000 two-pass	2-37,500 two-pass	70,000 two-pass	80,000 two-pass	2-35,250 two-pass	137,500 two-pass	8-22,000 single-pass	88,500 single-pass	100,000 single-pass
Average Sq Ft per Kw	1.58	1.72	1.54	1.51	1.52	1.37	1.21	1.1	1.15	0.99	0.975	0.71
Average Gpm per Kw	2.06	1.93	2.07	1.92	1.98	1.87	1.56	1.74	1.51	1.49	1.44	1.15

FIG. 5 DESIGN CHARACTERISTICS OF SURFACE CONDENSER INSTALLATIONS FOR TURBINES OF 20,000 KW AND LARGER

(Graph covers approximately 90 per cent of all installations. Curve shows trend of design and operating practice for heat rejected to the condenser circulating water. Each line, representing one condenser installation, indicates by point at lower end "Sq ft of area per kw of gross generation," and by point at its upper end "Gallons per minute of circulating water per kw of gross generation." The installations for each year are plotted in the order of turbine efficiencies, those for the larger and more efficient turbines being placed to the right.)

additional capacity inherent in the very liberally designed low-pressure ends of the older turbines. New generators were installed and the four 25,000-kw turbines in the West End Station in Cincinnati were modified so as to increase their normal capacity to 36,000 kw each. The two 45,000-kw tandem-compound turbines in the Columbia Station below Cincinnati were converted to cross-compound machines by the installation of new high-pressure turbine-generators and their capacities increased to 63,000 kw each.[8]

NINE TURBINES IN THREE STATIONS HAVE THE SAME CHARACTERISTICS

The turbine plant of the Trenton Channel Station of the Detroit Edison Company is a rather interesting contrast to the diversified practice shown in Table 1. There are installed in this station six 50,000-kw, 1200 rpm single-cylinder turbines. Two machines of the same characteristics are installed in the Delray No. 3 Station, and another machine of the same characteristics is installed in the Marysville Station of the same company. The turbine spindles, the generator fields, and the other principal parts of these nine turbines are exactly interchangeable. The use of the spare turbine spindle which has been provided has, in a number of cases, very much shortened the time of an enforced outage due to turbine-spindle trouble. One of the 50,000-kw turbines is able to carry approximately 9 per cent of the maximum system peak of the Detroit Edison Company.

[8] For more complete discussion of work done in West End and Columbia Stations see discussion by E. S. Fields, p. 1120, vol. 48, Transactions American Institute of Electrical Engineers.

MISCELLANEOUS TURBINE DEVELOPMENTS

Other important developments in turbine practice in this country have been the very general use of water-sealed glands with resulting elimination of condensation in the wintertime on the ceiling of the turbine room; the development of speedier governor gear which will prevent the speed of the turbine from rising to a point where the overspeed trip will operate on loss of load by the generator; the almost complete elimination of lateral vibration of turbine wheels as a cause of turbine troubles; the rather extensive use of a vacuum trip attachment on the turbine throttle as a substitute for an atmospheric relief valve on the condenser; and the use of multi-valve construction so as to give good light-load efficiency on the very large turbines.

TIME REQUIRED FOR STARTING AND LOADING TURBINE-GENERATORS

The time required for bringing turbines of 20,000 kw capacity and larger from a cold condition up to synchronous speed varies from 30 to 180 min, and the rate at which load is applied varies from 1000 kw to 3000 kw per min. Representative practice for a 50,000-kw single-cylinder turbine would require almost two hours for warming up, bringing up to speed, and applying the rated load. Operating engineers seldom consider it to be good practice to operate large turbine-generators at less than 20 per cent of their rated capacity. Special provision can, however, be made on machines to be operated in parallel with water-power plants for the bypassing of sufficient steam around the governor valves to permit of keeping the turbine spindle cool, and under these conditions the turbine may be operated at appreciably lower

loads. The provision of a turning gear which will permit of rotating the turbine spindle at a speed of from 1 to 25 rpm keeps the turbine spindle straight and reduces the time required for starting. It has been considered good operating practice in a number of stations to close the stop valve in the turbine lead and open a valve in a vent connection between the stop valve and the throttle valve in order to definitely prohibit the leakage of steam into the turbine while it is not in service. This aids quick starting. The measures which most effectively facilitate quick starting are the use of steam at low temperatures and perhaps the adoption of a turbine of multi-cylinder design. One of these measures tends, however, to increase the fuel consumption, and the other tends to increase the fixed charges per unit of station output. The solution adopted, therefore, must be a compromise.

Condenser Installations

The broad trends in the design of surface-condenser installations for turbine plants are shown by Fig. 5. The importance of delivering feedwater having a minimum content of solids to the boilers has almost entirely eliminated the use of jet condensers, and not more than six or eight turbines of 20,000 kw capacity or higher, installed since 1920, have been so equipped. Surface condensers have increased in size as the turbines have increased in capacity, but the increase has not been nearly in proportion. The use of higher initial steam pressures and temperatures, reheating, extraction of steam from the turbine at a number of points for heating feedwater on the regenerative cycle, and an increase in the Rankine cycle efficiency ratio (the efficiency of converting heat available in the steam into electrical energy) have all contributed, as shown by Fig. 5, to decrease the amount of heat rejected to the condenser circulating water for each kw-hr generated. This rejected heat has, on an average, over a period of the last 12 years, been decreased from about 10,800 Btu to about 7000 Btu per kw-hr of gross generation. The area of condensing surface has, however, on the average decreased from about 1.6 sq ft in 1920 to 0.98 sq ft per kw of turbine capacity installed in 1930, and the amount of condenser circulating water used has decreased from 2.0 gpm to 1.44 gpm per kw of turbine capacity. For the eight large units operated initially in 1931, an average of 0.71 sq ft of condensing area was installed and 1.15 gpm of condensing water circulated per kw of turbine capacity. The use of single-pass condensers has increased very fast.

Much worth-while improvement in condenser design, based on thoroughgoing research work, has been accomplished. The use of better tube arrangements which insure steam penetration to all parts of the tube bank, the continuous scavenging of non-condensable vapors, and low-pressure drop from turbine exhaust to air-pump suction; the use of more effective air coolers usually external to the condenser shell; higher water velocities of the order of 7 to 8 fps through the tubes; the rather extensive use of $^{7}/_{8}$ in. and $^{3}/_{4}$ in. diameter condenser tubes rather than 1 in. o.d. tubes, as was conventional practice prior to 1924, and the use of more efficient steam-jet air-removal pumps have all contributed to improve surface-condenser performance. It has been demonstrated[9] that a 30,000-sq ft condenser of modern design can give the same overall performance as a 50,000-sq ft condenser typical of good design practice in 1919, serving a duplicate turbine and supplied with the same amount and quality of circulating water. More efficient and more frequent cleaning of condensers, facilities for the removal of debris from the inlet tube sheet while the unit is in service, divided water-box construction where there is much debris in the circulating water, and more careful safeguarding against air leakage into the condenser have all had their part in maintaining good surface-condenser performance. All of these improvements in design and operating practice have fallen somewhat short of making it possible, with the modern condensers of reduced surface operating with the much smaller amounts of circulating water per kw, to maintain as high vacuums as with the installations typical of 1920 practice. The acceptance of somewhat lower vacuums justified by the accompanying reduction in investment in the condenser and its related equipment is indicative of the careful proportioning which in recent years has entered more fully into power-station design.

The combined effects of higher initial steam pressures and temperatures, the use of the regenerative cycle, and the rather general

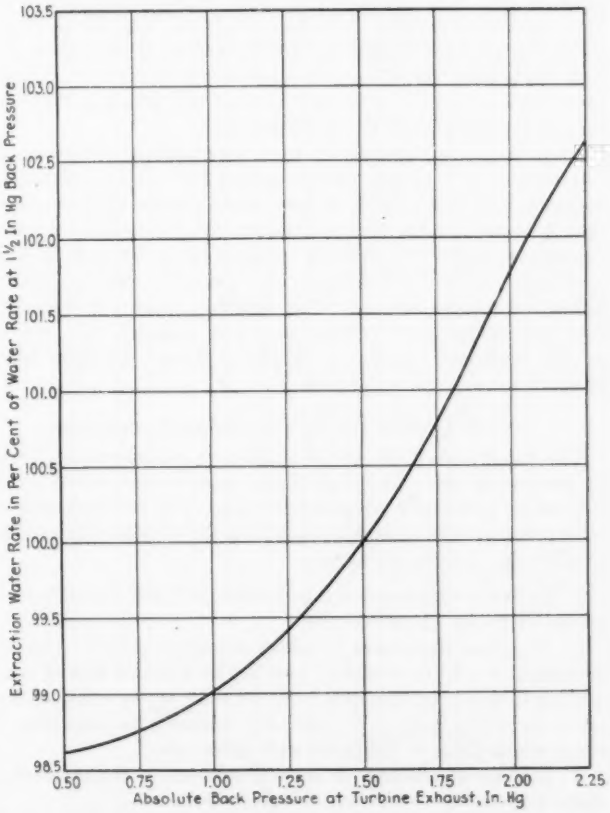


Fig. 6 Water-Rate Characteristics of a Modern Turbine at Its Maximum Rated Output

(Showing effect of relatively high steam pressure and temperature, reheating, use of regenerative cycle, and high leaving loss on the ability of the turbine to avail itself of beneficial effects of high vacuums.)

acceptance of a leaving loss of 5 to 6 per cent at maximum load and 29 in. vacuum have resulted in the shape of the overall turbine water-rate curve on an extraction basis being about as shown by Fig. 6, with the turbine rated at its proper economic capacity. Such a turbine, having but little ability at maximum output to avail itself of the beneficial effects of vacuums higher than 28.5 in. Hg, does not require nearly so large a surface condenser as would a non-extraction-type turbine of the same capacity designed for lower steam pressure and temperature and for a lower leaving loss. It develops, therefore, that the acceptance of higher leaving loss results in a substantial reduction in the investment in the condensing installation as well as in the turbine. Fig. 6 shows a reduction at full load of only 1.4 per cent in the water rate incident to an improvement in vacuum from 28.5 in. to 29.5 in. Hg. However, this same turbine at 70 per cent of its maximum rated output will for the same improvement in vacuum

[9] "Some Results of Condenser Operation," by Edwin B. Ricketts, research engineer, New York Edison Company; paper presented before American Society of Mechanical Engineers, December, 1926.

show a decrease of approximately 3.6 per cent in water rate and at 40 per cent of its maximum output a reduction of approximately 8 per cent in water rate for the same improvement in vacuum. This emphasizes the importance of condenser installations which by reason of low air leakage, efficient air pumps and air coolers, and high maintained water velocities can at partial loads maintain very high vacuums.

The trend toward the use of less surface per kw of installed turbine capacity has appreciably simplified what would otherwise have been a most difficult design problem for large turbine installations. It should not be inferred, however, that the size of condenser shells has borne a direct relation to the installed tube surface. In order to provide for adequate penetration of the steam to all parts of the tube bank, it has been necessary to provide some form of differential tube spacing or its equivalent, and this in turn has reduced the space factor (ratio of total tube cross-sectional area to total tube-sheet area), which in turn increases the diameter of the condenser shell.

From two to eight condensers have been installed to serve each of a number of the large cross-compound and tandem-compound turbines, and in a number of cases these condensers have been closely coupled to the turbine exhaust with the condenser tubes placed vertically. The present trend seems to be back to conventional practice with a horizontal condenser placed directly below the turbine exhaust. Ten turbines, ranging in capacity from 105,000 kw to 165,000 kw, have been installed, each served by one horizontal condenser. Eight of these condensers were designed for a single water pass.

Trend Toward Use of Single-Pass Condensers

The trend toward the use of single-pass condensers has been an interesting one. The four design factors which have made it possible appreciably to reduce the amount of surface required to absorb a certain amount of heat from the turbine exhaust are, in the order of their importance:

1 Better arrangement of tube surface, insuring that effective work will be done by each tube.

2 The use of high water velocities through the tubes. If good performance is to be obtained, then for each square foot of condensing surface approximately 0.005 kw must be expended on the job of scrubbing down to a reasonable thickness the dead film of water which clings to the inside wall of the tube.

3 The use of efficient air coolers, external to the condenser shell.

4 The use of smaller diameter tubes.

As designers have availed themselves of these factors, whether in two-pass or single-pass designs, there has been marked improvement in the effectiveness of the condenser, and later designs have been laid out with less surface to perform the same duty. With relatively less total surface and the use of tubes of smaller diameter and greater length, it has been found readily possible to arrange the surface for a single water pass, hold the water velocity through the tubes up to approximately 8 fps, and still not have the total circulating water exceed the amount which, for maximum load conditions, will correspond to a temperature rise of from 12 to 15 F, which in general has been found to result in a proper economic balance between pumping power, investment in circulating water tunnels and piping, and the steam consumption of the main turbines. The use of condenser shells of relatively smaller diameter with tubes up to 30 ft in length has, for the larger turbines, very much facilitated the design of the foundations which support the machines and has, in addition, resulted in condensers having appreciably lower cost for the shells, the tube sheets, and support plates.

For two-pass condensers the difficult problem is to insure that each tube will absorb its proportional amount of heat. For single-pass condensers the problem of longitudinal distribution of steam becomes difficult, particularly where very long tubes are used. Means have been devised, however, to insure that a proportionately larger amount of steam will be delivered to the cold end of the single-pass condenser.

Another interesting design trend is that toward the use of tubes rolled into the tube sheets at both ends. This feature reduces leakage and also tends to reduce the investment. A large percentage of all condenser installations in the United States have cast-iron shells and water boxes and brass tube sheets. The pressure to reduce investment has resulted in some of the newer installations being made with welded steel shells and steel tube sheets. It seems probable that this trend will continue at least as relating to the use of welded steel shells.

Deterioration of Condensing Equipment

The problem of slowing up the rate of deterioration of condensing equipment is a most difficult one, especially as the sources of condenser circulating water are becoming increasingly contaminated with sewage and industrial waste. It is fairly general practice to use Admiralty condenser tubes of No. 18 B.w.g. thickness for all installations with salt or brackish water, and Admiralty tubing is being used in preference to Muntz for perhaps 50 per cent of the newer installations using fresh circulating water from rivers or lakes. Corrosion of varying degrees of rapidity is encountered mainly at the inlet end of the tubes. This inlet corrosion is particularly acute for installations using salt water having a large amount of entrained air. The installation in the inlet water box of zinc, mild steel, or aluminum plates, supported by bronze bolts screwed into the tube sheet, has in many cases appreciably slowed up the corrosion. In a few cases provision has been made for the positive circulation of direct current between plates in the inlet water box and the tubes, and the results have appeared to be beneficial. The life of Admiralty tubing with salt water varies from three to seven years and probably averages nearer the lower figure. Aluminum bronze, aluminum brass, arsenical copper, and "30%-nickel–70%-copper" tubes are being experimented with and show promise of longer life where water conditions are bad. High content of entrained air in the circulating water causes rapid graphitization of cast-iron water boxes.

The use of brass and semi-steel water-box rings is being considered. There is a trend toward the use of slower-speed circulating pumps, and this, together with the installation of high-grade bronze impellers and the limiting of the suction head by setting the pumps low, is having a beneficial effect in reducing the maintenance on circulating pumps.

Chemical Treatment of Condenser Circulating Water

During the last two years provision has been made in at least 12 generating stations for the chlorination of the condenser circulating water. Chlorination is used to combat two distinct adverse conditions not usually coexistent:

1 The formation of slime on the wall of the tube, with rapid decrease of vacuum.

2 The growth of mussels on the walls of the intake and discharge circulating-water tunnels. This results in decreased water flow and fouling of the inlet tube sheet of the condenser as the mussels are dislodged and carried into the condenser.

Both of these problems are most acute in the summertime and add to the difficulties of maintaining even reasonably good vacuums with the warm circulating water. It has been demonstrated that by dosing the circulating water with chlorine for a few minutes at intervals ranging from 90 to 180 min, the forma-

tion of slime can be inhibited.[10] Conditions were favorable for the formation of slime from the circulating water taken from the Illinois River for the Powerton Station at Pekin, Ill. Fig. 7, contrasting a year's operation without chlorination of condenser circulating water with another year's operation during which circulating water was chlorinated for the condenser of a 55,000-kw turbine in this station, shows that with chlorination the performance of the condenser can be made to approach very closely that corresponding to a 100 per cent clean condition. Prior to the installation of chlorinating equipment the condenser tubes were scoured twice a day during the summer months by the injection of sand into the water box, and rubber plugs were shot through the condenser tubes whenever the unit was removed from service. These methods of cleaning the condenser have now been largely discontinued.[11] The proportion of chlorine required to be fed in at the screen well to accomplish this purpose is exceedingly small. While available data on mussel extermination are less decisive, results which have been reported are encouraging.

The chemical treatment of condenser circulating water bids fair to eliminate almost entirely the two filthiest jobs in the power station, the cleaning of the condensers and of the tunnels, and in addition it holds promise of very much reducing the outage time for the main turbines incident to performing this work. It is possible that, as additional experience is gained, appreciable savings will be effected in new stations by designing the circulating-water tunnels for higher water velocities on the basis of the knowledge that the tunnels can be maintained in a clean condition by the intermittent chlorination of the circulating water.

Condenser Auxiliaries

Steam-jet air-removal pumps are being used almost universally. Their compactness and low cost and the ability to absorb conveniently the latent heat of the steam used into the feedwater heating cycle have commended them in spite of their relative inefficiency when contrasted to slow-speed, motor-driven, rotative dry vacuum pumps.

Condensate pumps are usually designed to pump the condensate through at least one low-pressure extraction heater into an elevated surge tank or a combined deaerator, extraction heater, and surge tank. In most modern installations both condensate pump glands are under the pressure of the discharge from the first stage in order to avoid air leakage into the condensate. It is almost universal practice to install two condensate pumps for each turbine, one being held in reserve. Opinion is divided as to whether variable-speed drive should be provided for the condensate pumps.

The trend is toward the use of relatively slow-speed horizontal circulating pumps having efficiencies up to 85 per cent. It is fairly general practice to install two circulating pumps for each main turbine, their combined capacity being used for normal operation, but the capacity of one pump being sufficient for maximum load operation with some sacrifice in vacuum. Opinion is divided as to the necessity of variable-speed drive for the circulating pumps.

It appears that the trend with modern installations is or will be toward the maintained circulation, except for very light loads coincident with injection-water temperatures below 40 F, of an amount of circulating water which will correspond to water velocities through the tubes of from 7 to 8 fps.

[10] The original development work was done in the Crawford Avenue Station in Chicago, the Kearny Station in New Jersey, and the Northport Station on Long Island.

[11] For a more complete description, refer to article entitled "Chlorine Treatment of River Water at Powerton Station," by K. E. Stoll, Results Engineer, Super-Power Company of Illinois, *Power*, February 2, 1932.

Cooling Towers and Spray Ponds

There are probably not more than 15 generating stations of over 10,000 kw capacity in the United States that are served by cooling towers or spray ponds. The largest of these stations is the 82,500-kw station of the Dallas Power & Light Company.

Special Problems for Ohio River Plants

Very special problems are encountered in the design of the condenser installations for some of the turbine plants built on the Ohio River, the Mississippi River, and the Missouri River. In certain cases the rise in river-water level from the low flow condition to a flood condition is as much as 65 ft. The condensers and their auxiliaries in these stations are placed near the bottom of a well the floor of which is close to the extreme low-water level. In the case of the Columbia Station below Cincinnati, the floor

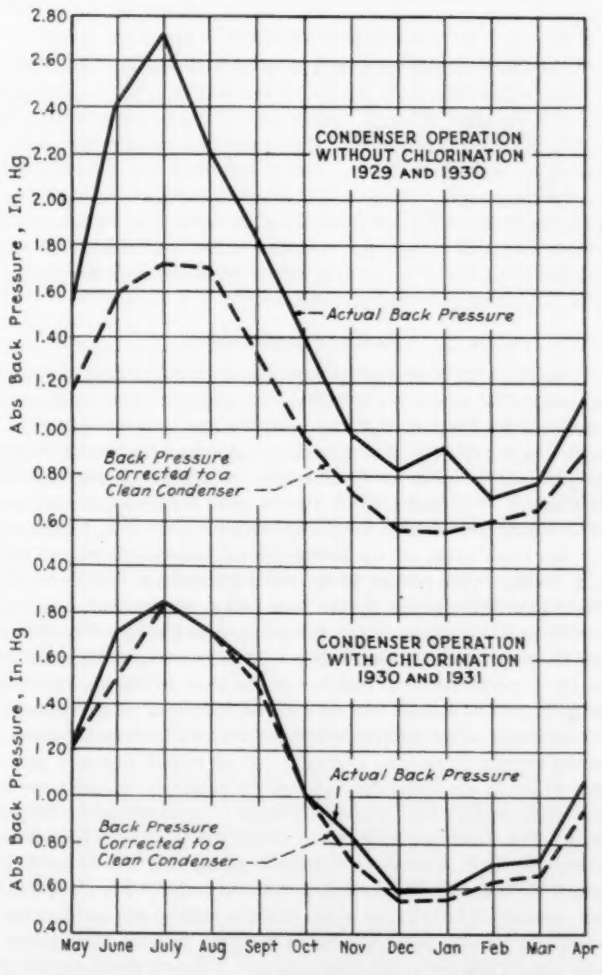


Fig. 7 Back Pressure at Exhaust of 55,000-Kw Turbine in Powerton Station
(With and without chlorination of condenser circulating water.)

of the condenser well is 86 ft below the top of the turbine platform, and the well has a diameter of 76 ft, being large enough to take the condensers and condenser auxiliaries for two 63,000-kw turbines. The very long revolving screens required for these river plants present special problems of design. In some cases special provision is made for removing debris from the space in front of the bar screens by means of a clamshell bucket. The most interesting installations of this type are the West End Station and the

Columbia Station at Cincinnati, the Northeast Station at Kansas City, Mo., and the Cahokia Station at East St. Louis, Ill.

Feedwater Heaters

The multi-stage heating of feedwater by steam extracted from the main turbines is mainly accomplished in closed feedwater heaters, which are placed as close to the turbine as possible. Vertical heaters are quite generally used in the recent installations because of their space economy. In a number of stations one open heater of the deaerating type, having a considerable amount of water-storage capacity, is placed so as to deliver water to the suction of the feed pumps. The closed feedwater heaters are designed to operate on from 3 to 10 F terminal difference, and the open feedwater heater is designed for less than 1 F terminal difference. It is general practice to cascade the drips from one heater into the next lower heater and pump the combined drips back into the feed line at either one or two points.

Evaporators

In the larger condensing stations of modern design, the make-up to the boilers runs from 0.8 to 1.5 per cent, and this make-up is usually supplied through evaporators. Fresh water is usually fed to the evaporators, and in some cases this water is first passed through a water softener in order to minimize the scale troubles in the evaporator. It is rather general practice to deliver low-pressure steam extracted from the main turbine to a single-effect evaporator and to deliver the vapor either into the next lower feedwater heater or to an evaporator condenser located just beyond the next lower extraction heater.

Boiler-Feed Pumps

Two or more boiler-feed pumps are usually grouped with and considered as a part of the turbine installation. In a number of cases the feed pumps have been selected so that one pump has sufficient capacity to handle the feedwater for a turbine of 50,000 kw and in a few cases for turbines of even larger capacity. There is a school of thought which favors more conservative practice in connection with this most important auxiliary, as evidenced by the installation of six motor-driven boiler-feed pumps and four steam-driven pumps to serve the 208,000-kw turbine in the State Line Station and by the installation of nine motor-driven pumps and two steam-driven pumps to serve the two 105,000-kw turbines in the Powerton Station. It is rather general practice to group moderate-sized turbines of less than 40,000 kw capacity in pairs and to install four boiler-feed pumps of equal capacity for each pair of turbines, one motor-driven and one steam-driven pump serving as reserve capacity. It is rather common practice to drive the boiler-feed pumps by means of variable-speed induction motors having a speed range of approximately 20 per cent. For steam pressures up to 650 lb per sq in., the feedwater passes through only the condensate pump and the boiler-feed pump on its way from the condenser to the boiler. For an operating pressure of 1200 lb per sq in., there is usually one and in some cases two booster pumps installed in series between the condensate pumps and the boiler-feed pumps. A recent development of interest is the use of motor-driven boiler-feed pumps of the plunger type to serve the two new 1200-lb-pressure turbines in the State Line Station.

Arrangement of Equipment

Most of the turbine plants fall into three classes, as regards the arrangement of equipment:

1 Turbines placed lengthwise in the turbine room with horizontal condensers directly beneath the turbine exhaust, the condenser tubes being at right angles to the turbine shaft. Turbines are arranged in pairs, with the throttles of No. 1 and No. 2 adjacent to each other and a common space provided for withdrawing the fields of No. 2 and No. 3 machines. Only a narrow platform is provided around the main turbine, and the turbine-room crane has direct access to the condenser water boxes and the circulating pumps. Space is provided on the condenser-room floor for the landing of parts when the turbines are dismantled.

2 Turbines are placed crosswise, and horizontal condensers are placed directly beneath the turbine exhaust, with the condenser tubes at right angles to the turbine shaft. Turbines are arranged in pairs, with a common space for each pair used for the withdrawal of condenser tubes and the installation of the condenser circulating pumps. Somewhat larger platforms are placed around the turbines, but in most cases openings are provided so that the crane has access to the condenser water boxes and the circulating pumps. Space is provided on the condenser-room floor for the landing of parts when the turbines are dismantled.

3 Where a high premium is placed on ground area, the turbines are placed crosswise in the turbine room and horizontal condensers are used with their tubes parallel to the turbine shaft. The space underneath the main generators is used for the withdrawal of condenser tubes and for the circulating pumps. The distance from center line to center line of adjacent turbines is cut down to a minimum. A continuous floor is placed at the top elevation of the turbine platform, and during dismantling turbine parts are landed in the space at the end of the generator. In order to provide space for the withdrawal of the condenser tubes, it is necessary to locate the generator air coolers, generator air ducts, and the external fans above the turbine platform. For such installations the external fans for the generators have been mounted on the two sides of the generator, at the end of the generator, or on top of the generator.

The trend of boiler-plant design has been toward more and more capacity per square foot of ground area, and the present tendency, at least for turbine units of 50,000 kw capacity and smaller, is to install one boiler per turbine. With the crosswise arrangement in the turbine room and a single row of boilers in the boiler room, the space requirements for boilers and turbines substantially match each other. It is believed that, even for very large turbine units and compact layouts used where ground area is at a premium, it will still be possible with a crosswise turbine-plant layout to match boiler capacity against turbine capacity by installing two boilers per turbine, one on either side of a central firing aisle, which will be parallel to the turbine room.

Centralized Control

In some of the newer stations of moderate capacity up to 100,000 kw, provision has been made for centralizing the control for two turbine-generators, together with their related boilers, and the electrical control for outgoing transmission lines at a point in the auxiliary bay between the boilers and the turbines. The wall between the turbines and the boilers is omitted so that the operator may be able to see the principal pieces of equipment from the control station.

Turbine-Generator Foundations

Both structural steel and reinforced concrete have been used for turbine supports. For the larger turbines it is usually necessary to use foundation supports of greater height in order to provide space for the condenser and the piping. This makes the problems of design more difficult.

Structural-steel supports have been most generally used where very compact layouts have made it necessary to conserve as much space as possible for the installation of the condenser and piping. This is particularly true where the condenser is installed with the tubes parallel to the turbine shaft.

Perhaps the most important factor contributing to the success-

ful performance of turbine foundations, whether of steel or reinforced concrete, is the stiffness of the individual members and the rigidity of the support as a whole. While increased mass in the support aids of itself in resisting tendencies to vibrate, and therefore the more massive reinforced concrete supports are inherently advantageous, it has been found that by limiting the deflections of individual members to low values and providing adequate bracing, satisfactory supports can be constructed of either steel or concrete. With steel supports mass and stiffness are provided by filling between steel girders with concrete wherever practicable. Generally the supports are entirely independent from the building framing. In a number of New York stations, however, the floors around the units have been framed from building columns directly to steel turbine supports with satisfactory results. This procedure conserves space below the turbine-operating platform by eliminating auxiliary floor-supporting columns or deep girders. With this construction the floor framing is usually not considered as bracing the turbine support, which is designed as though entirely independent, and special care is taken to make the support itself rigid to insure that vibration will not be transmitted to the building frame.

There have perhaps been more cases of troublesome vibration in resonance with the turbine in structural-steel supports than for those constructed of reinforced concrete. Most cases of troublesome vibration can be traced to a lack of stiffness in the support rather than to the material of which it was constructed, and usually it is found to be due to unbalance of the turbine-generator rather than to inadequacy of the support itself.

General Acceptance of Motor Drives for Turbine-Plant Auxiliaries

There has been a very definite trend toward the use of motor drives for the turbine-plant auxiliaries, the one major exception being the 280,000-kw East River Station of the New York Edison Company, in which case it was considered that extreme measures were justified for the insurance of reliability. In most cases 2300-v induction motors have been used for the drive of the condensate pumps, the circulating pumps, and the boiler-feed pumps. The incentive of simpler, less expensive, and more compact control equipment of the air-break type has dictated the use of 440-v and 550-v motors in several of the recent installations. Direct-current motors have been used in the stations of the Detroit Edison Company.

Protection of Power Supply for Auxiliaries

A 10-sec interruption in power supply for the circulating pumps or a 60-sec interruption for the condensate pumps and boiler-feed pumps is apt to cause an outage of turbines or a curtailment of their output. It becomes extremely important, therefore, to safeguard the continuity of power supply to these auxiliaries.

Five solutions of this problem have been quite generally used:

1 The power supply for the essential auxiliaries is taken through one or more banks of transformers from the main-station bus. This solution is particularly applicable where the station is interconnected with several other stations and there is fair assurance that at least one of the several sources of power supply will always be available. Facilities must be provided for quickly switching the station auxiliaries to any one of these sources of power supply, and it is to be noted, especially ni connection with the circulating pumps, that but little time is available to make a changeover. It is rather common practice to divide the essential auxiliaries into two groups, each deriving its power supply from a separate source, thus insuring that the loss of one source of power supply will shut down only one of the two circulating pumps for each machine.

2 The essential auxiliaries for each turbine are supplied with power from a bank of transformers connected directly to the generator leads. Where generator reactors or step-up transformers are installed in the generator leads, the voltage at which power is supplied to the essential auxiliaries still holds up fairly well, even though a short circuit occurs on an outgoing line, on the main-station bus, or in one of the main generators.

3 The essential auxiliaries for each turbine receive their power supply from a smaller generator directly coupled to the main generator. These smaller generators for auxiliary power supply are not operated in parallel, and the auxiliary generator for each turbine and its group of essential auxiliaries constitute a system that is independent of troubles on the main-station bus and independent of troubles with the auxiliaries of any of the other machines. In order to provide for starting and the possibility that any one of the auxiliary generators may be temporarily crippled, provision is made for an alternate source of power supply from the transformers that supply power to the non-essential auxiliaries or from a house turbine.

4 The power supply for the essential auxiliaries is taken from an auxiliary generator, which is driven either by means of a condensing turbine or by a turbine which supplies steam for feedwater heating. In the latter case, it is necessary to interconnect the auxiliary generator with the main-station bus in order to provide for interchange of power. This solution of using a separate auxiliary generator driven by means of a condensing turbine has the operating advantages of the auxiliary shaft generator and the further advantage of being free from troubles incident to a drop in system frequency. It involves, however, a rather considerable investment and at least some sacrifice in overall station efficiency.

5 A combination of the first and the fourth solutions may be obtained by installing a turbine-driven auxiliary generator, which is available to pick up the auxiliary load quickly in the event of trouble on the main-station bus bars. In some cases these auxiliary generators spin normally as synchronous condensers with no steam passing through the auxiliary turbines, and in other cases they stand normally at rest, but are of such design that they can be started and the load be applied within 15 sec. These auxiliary turbines usually exhaust to atmosphere.

The problem of protecting the power supply for the essential auxiliaries in the boiler plant is of similar nature and is directly related to turbine-plant problems. It has, however, certain special aspects, the discussion of which is not within the scope of this paper.

It is probable that at least 60 per cent of the large turbine-generators installed have direct-connected exciters.

Reliability of Turbine-Generators

For the last ten years an intensive and critical study has been carried on by the Prime Movers Committee of the National Electric Light Association, having to do with the extent, the causes, and the means of prevention of outages of turbine-generators.[12]

The results of this study over the 10-year period are summarized in Table 2. The most interesting things to note in this table are that the total average outage for the turbine-generator has decreased from 15.3 per cent of the total hours in 1922 to 8.37 per cent in 1931 and that there has been a substantial reduction in the outage hours for the turbine and for the generator, but only slight indication as yet of a reduction in the outage hours incident to condenser trouble.

[12] For the details of this study, which has been comprehensive, refer to the reports of the Prime Movers Committee, National Electric Light Association, for 1922 and 1923 and the Serial Reports of the Turbine Subcommittee of the Prime Movers Committee for 1924, 1927, 1929, 1930, 1931, and 1932.

TABLE 2 AVERAGE PERFORMANCE OF UNITS 20,000 KW CAPACITY AND LARGER

	1914–1921	1922	1923	1925	1926	1927	1928	1929	1930	1931
Turbine units reviewed	78	74	87	191	153	186	207	276	324	334
Service demand factor, %	77.80	74.80	78.40	68.80	70.55	71.11	72.09	69.43	65.58	60.40
Service demand availability factor, %	85.50	71.00	92.30	94.60	92.16	94.62	96.01	96.27	96.01	96.50
Unit capacity factor, %	43.80	48.10	51.60	44.40	49.20	46.13	48.15	40.57	43.35	38.42
Unit output factor, %	66.00	70.60	71.30	68.20	64.44	66.89	69.44	60.71	68.86	63.65
Service hours factor, %	66.50	68.10	72.40	65.10	67.06	67.14	68.58	66.84	62.96	57.93
Turbine outage factor, %	6.93	7.90	7.33	7.29	6.28	6.36	5.73	5.02	4.44	4.20
Generator outage factor, %	2.56	3.10	1.74	1.85	1.70	1.68	1.13	1.47	1.52	0.95
Condenser outage factor, %	1.20	2.50	3.18	3.51	2.65	2.75	2.77	2.61	2.51	2.16
Other causes outage factor, %	0.66	1.80	0.66	0.98	0.57	1.06	0.57	0.72	0.70	1.06
Total outage factor, %	11.35	15.30	12.91	13.63	11.20	11.85	10.22	9.82	9.17	8.37
Reserve hours factor, %	22.05	16.60	14.69	21.25	21.74	21.04	21.22	23.34	27.87	33.70

NOTE: "Service demand factor" is the ratio of the demand hours to the period hours, the demand being either because of need for capacity or for best system economy. "Service demand availability factor" is the ratio of the service hours to the demand hours. "Unit capacity factor" is the ratio of the kw-hr generated to the product of the unit rating and the period hours. "Unit output factor" is the ratio of the kw-hr generated to the product of the unit rating and the service hours. "Service hours factor" is the ratio of the service hours to the period hours. The several "outage factors" and "Reserve hours factor" are the ratio of the hours in question to the total period hours.

As the records of turbine-generator outages have been analyzed year by year, predominant weaknesses have been located and steps have been taken to eliminate them. The manufacturing companies have carried on extensive research work, and by the use of better materials and the better application of these materials, have eliminated causes of trouble. Almost every case of field trouble has been the subject of critical study by the engineers of the manufacturing companies. The operating engineers have studied the problem from the standpoint of developing better operating methods, detecting trouble in its early stages and avoiding its further development by preventive maintenance work, and by carefully scheduling a large percentage of the outages of the turbine-generator unit for times when the unit is not needed for capacity or for best system efficiency.

The development of machines that involve new design problems has almost always been accompanied by increased outages, and as field experience has been gained the outages have been materially reduced. In 1925 the turbine-outage factor for the seven turbines, operating at a pressure of approximately 550 lb per sq in. gage, was 14.75 per cent. This factor was reduced to 5.6 per cent in 1928, 4.76 per cent in 1929, and 4.32 per cent in 1930. The turbine-outage factor for turbines of 50,000 kw capacity and higher in 1928 was 8.05 per cent. This factor was reduced to 7.04 per cent in 1929 and to 5.17 per cent in 1930. The turbine-outage factor for 1930 for ten turbines operating at a pressure of approximately 1200 lb was 10.68 per cent. It is to be expected that the outages in connection with the 1200-lb turbines will be progressively reduced as additional field experience is gained.

The most encouraging aspect of the turbine record for 1930 was that the record for the 16 machines which saw their first year of operation in 1930 shows a total outage, chargeable to all causes, of 5.06 per cent, which is in contrast to an average outage of 9.17 per cent for the 324 machines reported upon. For every other year during which the records have been analyzed, the outages for the new machines having their first year of operation have always been appreciably higher than the average by reason of the troubles encountered during the early months of operation. This highly satisfactory record for 1930 gives indication that the manufacturers are now producing machines that may reasonably be expected to operate satisfactorily from the day they go into service.

The effective work that the operating engineers have done is evidenced by the fact that they were able to schedule 72 per cent of the total outages for the turbines during 1930 for such times as operation of the machines was not required.

Remaining Causes of Turbine-Generator Outages

Certain important factors remain that will continue to cause outages of turbine-generator units, as follows:

1 Breaking of turbine blades after they have become weakened through vibration. We believe that as a result of research work being done by the manufacturers the extent of this trouble is decreasing year by year.

2 Erosion of low-pressure blading. Although minimized to some extent by the use of better blade materials and the bleeding out of some water with steam extracted for the low-pressure heater, this trouble continues to be a serious one, and we must face periodic replacement of low-pressure blading.

3 Deposits of solid matter on turbine blades. For large turbines operating on high load factor, the loss in capacity may be as much as 6 per cent in three months from this cause. One solution is to wash the turbine by the injection of water sprayed into the steam lead while the turbine is in operation, but the fundamentally correct solution is a reduction, by use of corrective measures, of the amount of moisture and the solids carried over from the boiler with the steam.

4 Condenser-tube corrosion. The use of the best materials available for condenser tubes and of methods for inhibiting corrosion gives promise of reducing outages chargeable to this cause.

5 Condenser cleaning. This problem has become more acute year by year because of the increasing contamination of condenser circulating water. It will not be surprising, however, if as a result of the chlorination of condenser circulating water the condenser outage factor is reduced to as low as 1.5 per cent of the hours in the year.

6 Generator outage. It is believed that the extensive use of the closed system of ventilation for generators from 1923 on has been very largely responsible for the reduction in generator outages. There is now a trend toward a slightly increased outage chargeable to the generator, and this no doubt is due to the use of larger machines. This is evidenced by the fact that the generator outage factor for machines of 50,000 kw capacity and higher for the year 1928 was 0.78 per cent, for 1929, 1.57 per cent, and for the year 1930, 2 per cent.

Trend of Station Performance

There was included as a part of the 1925 report of the Power Generation Committee of the American Institute of Electrical Engineers a curve showing the trend of performances of typical stations of 60,000 kw capacity and higher, plotted against dates of initial operation of the stations. In Fig. 8 we have reproduced this same curve and extended it to 1932 and have also shown the way in which improvements in the boiler plant and turbine plant have contributed to reduce the overall fuel consumption per kw-hr of net station send-out. The improvements in turbine plant performance have been almost entirely the result of a reduction in the amount of heat rejected to the condenser circulating water.

The curves in Fig. 8 show not the best performance that has been obtained, but rather the average performance of typical stations. The following accomplishments are indicative of the

performance that is being obtained in the most efficient stations operating at high load factors:

The 400-lb Long Beach No. 3 Station, operating on a station capacity factor of 66 per cent, had an average fuel rate during 1930 of 13,075 Btu per kw-hr of net send-out.

The 1200-lb Gilbert Station, operating on a load factor of 65 per cent, had an average fuel rate for six months of 12,536 Btu[13] per kw-hr of net send-out.

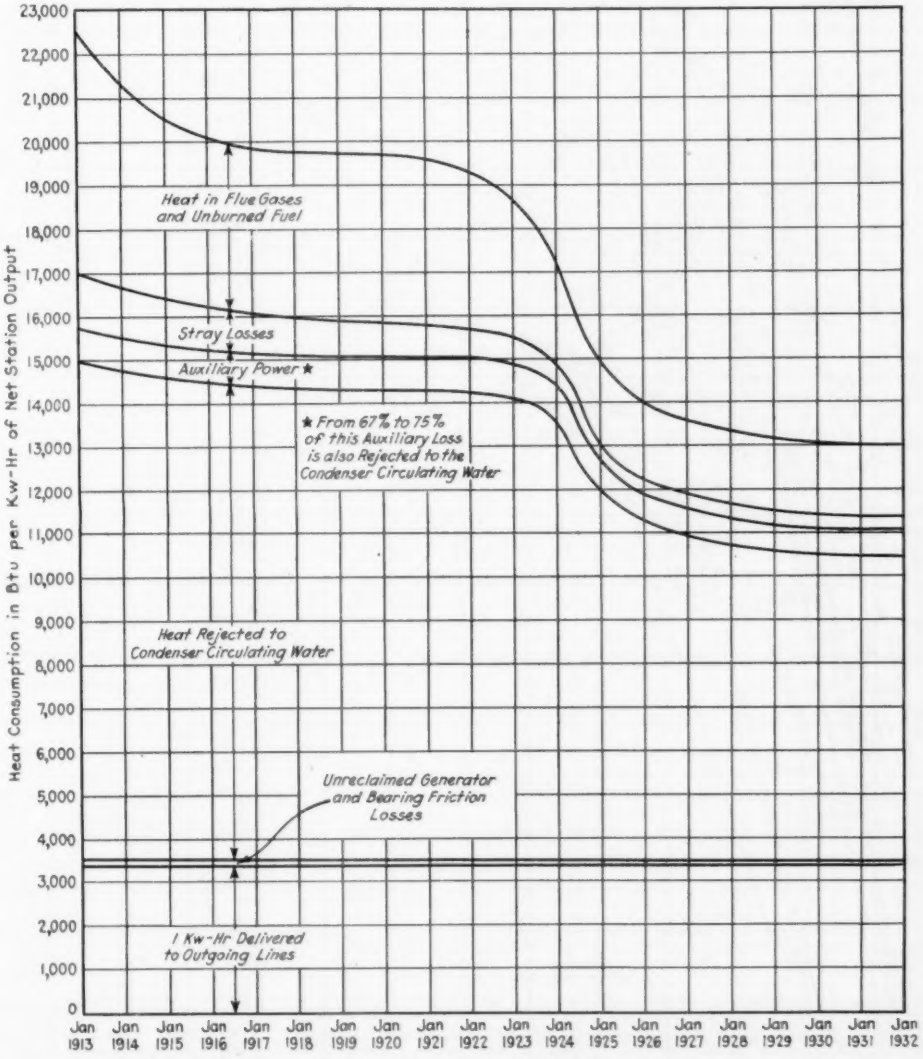


FIG. 8 PERFORMANCE OF TYPICAL STATIONS OF 60,000 KW CAPACITY AND HIGHER
(Plotted against dates of initial operation.)

The Powerton Station, operated at a steam pressure of 600 lb with reheat during 1931, with an average capacity factor of 63.8 per cent, had an average fuel rate of 12,680 Btu per kw-hr send-out, this result being accomplished with low-grade coal which averaged 10,213 Btu per lb.

The 1200-lb Deepwater Station, operating on a load factor of 81.1 per cent during 1931, had an average fuel rate of 12,050 Btu per kw-hr of net send-out.

WATER-RATE TESTS ON TURBINE-GENERATORS

A more specific review of turbine-generator performance is given in Table 3, which summarizes the results of carefully conducted water-rate tests made on 34 machines ranging from 6000 kw to 165,000 kw in capacity. Although not entirely satisfactory for the purpose, the overall Rankine cycle efficiency ratio (the efficiency of converting heat available in the steam into electrical energy delivered to the generator leads) has been found to serve most effectively as a basis of comparison for turbines operating under varying conditions as relating to steam pressure and temperatures at the throttle, back pressure at the turbine exhaust, extraction for feedwater heating, and the use of the reheating cycle.

The results of all the tests have been carefully reviewed and the average overall Rankine cycle efficiency ratios calculated on the basis of the Keenan steam tables, published in 1930 by The American Society of Mechanical Engineers. Figs. 9 and 10 show the overall Rankine cycle efficiency ratios plotted against net generator outputs. It is to be emphasized that a direct comparison cannot be made in Figs. 9 and 10 between two machines operating under widely different conditions and that this applies particularly as relating to back pressure at the turbine exhaust. Test No. 30 serves to illustrate this point, the Rankine cycle efficiency ratio for full primary-valve opening dropping off from

[13] This fuel rate is based on coal purchased at the mine and covers the difference between mine weights and plant use, which is estimated to amount to as much as 2 to 3 per cent.

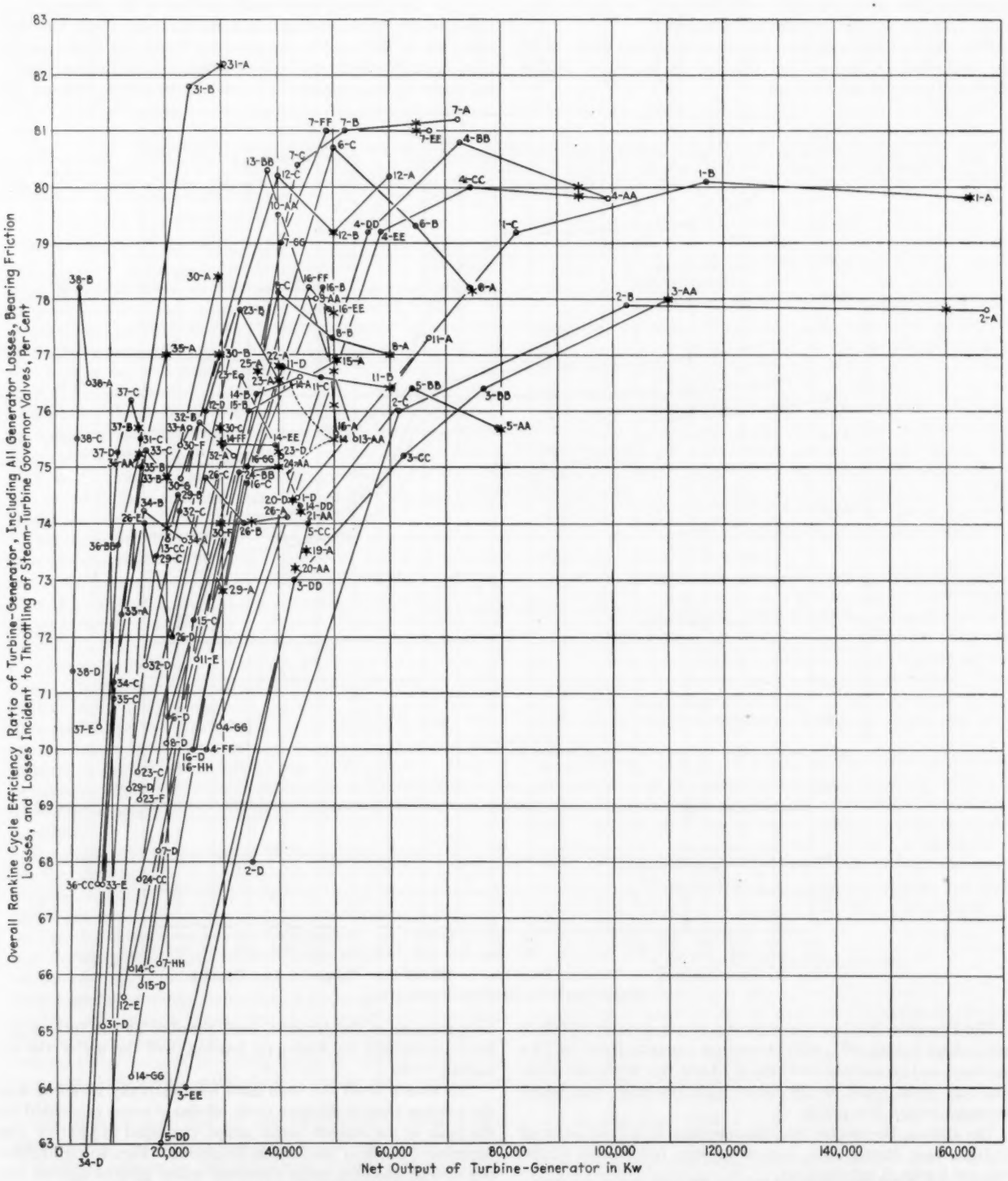


FIG. 9 TEST PERFORMANCE OF 34 TURBINE-GENERATORS RANGING IN CAPACITY FROM 6000 KW TO 165,000 KW

(Stars indicate normal rated capacity. Double letters indicate extraction tests and single letters non-extraction tests. Fine lines connect tests made at several different loads on same machine. Detail results of tests are summarized in Table 3. Also refer to Fig. 10 for large-scale graph of range from 0 to 50,000 kw.)

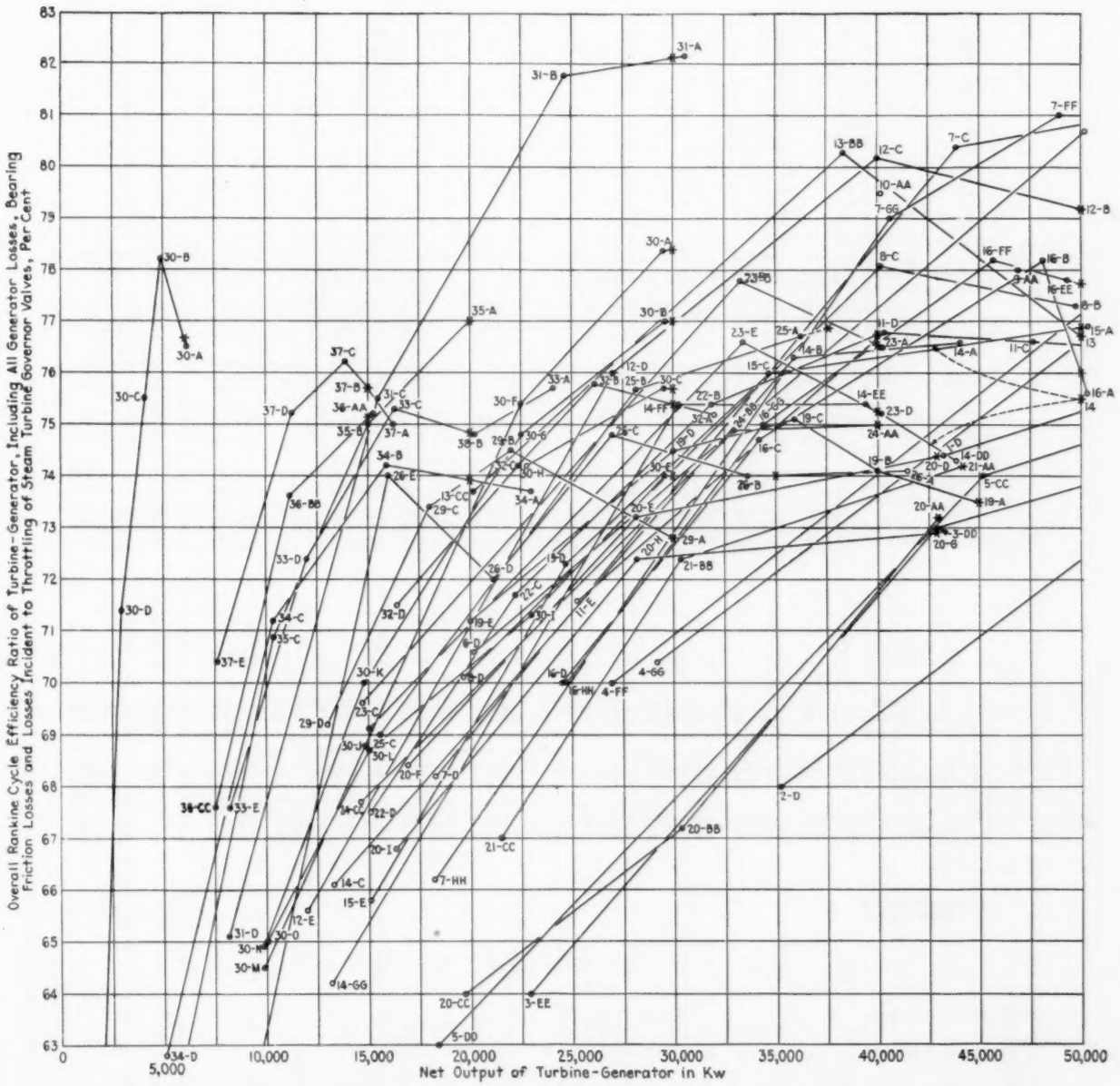


FIG. 10 TEST PERFORMANCE OF 34 TURBINE-GENERATORS RANGING IN CAPACITY FROM 6000 TO 165,000 KW.

(Stars indicate normal rated capacity. Double letters indicate extraction tests and single letters non-extraction tests. Fine lines connect tests made at several different loads on same machine. Detail results of tests are summarized in Table 3. Also refer to Fig. 9 for small-scale graph of range from 0 to 170,000 kw, which shows all of the test points.)

78.4 per cent for a back pressure of 1.505 in. Hg to 77 per cent for a back pressure of 0.987 in. Hg and to 74 per cent for a back pressure of 0.572 in. Hg. Test No. 30 was made on a turbine having a relatively low leaving loss. Tests made on a machine having a higher leaving loss would show an even more rapid decrease in Rankine cycle efficiency ratio with decrease in back pressure.

The results for each test must be interpreted in the light of the operating conditions as given in Table 3.

It is of interest to note that the Rankine cycle efficiency ratios for normal rated loads, as shown in Fig. 9, all fall within the range from 72.8 per cent to 82.2 per cent, both the maximum and minimum values occurring for 30,000-kw turbines. Maximum load tests Nos. 1-A, 2-A, 7-A, and 31-A, all made on a non-extraction basis, are fairly comparable, the back pressures ranging from 2.21 in. Hg to 2.48 in. Hg. Maximum-load tests Nos. 4-AA, 6-A, 8-A, 12-A, 16-EE, 30-A, and 38-A all show relatively good performances at somewhat lower back pressures ranging from 1.0 in. Hg to 1.58 in. Hg, and would certainly have shown somewhat higher Rankine cycle efficiency ratios had the tests been made with a back pressure of 2.25 in. Hg.

A study of Figs. 9 and 10 shows the very limited extent to which the use of very large turbines has contributed to the reduction of unit fuel consumption. The increased difficulty of fitting the larger machines to the daily load curves tends to increase the hours of light-load operation. Notwithstanding the rather marked effect which the use of from three to five governor valves has had in improving the partial-load performances of the larger turbines, their light-load performances still fall below the maximum-load performance of even the moderately efficient turbines of one-third the capacity.

No tests on turbine-generators operating on the reheating cycle were available for inclusion. It is probable that at least

TABLE 3 RESULTS OF TESTS ON LARGE TURBINE-GENERATORS INSTALLED IN UNITED STATES

	Characteristics of Turbine Installation									Test Conditions										
										Steam at throttle										
Test No.	Date installed	Normal rating, kw	Number of cylinders and speed in rpm	Type and number of stages	Make	Admission valves	Extraction points	Feed temp. for maximum load, F		Abs. pressure, lb	Temperature, F	Abs. back pressure at exhaust, in. Hg	Extraction or non-extraction	Generator net output, kw	Number of admission valves open (approx.)	Throttle water rate, lb per kw-hr of net output	Final feed temperature, F	Overall heat rate, Btu per kw-hr net output	Average overall Rankine cycle eff. ratio, %	Reference to detail report of test
1-A	1929	165,000	2—1800	cc	Westinghouse	4	2	265		282	663	2.21	Non-extr.	164,623	4.0	10.28	104.0*/	13,120	79.8	N.E.L.A. Pub. No. 151, p. 46
1-B	"	"	"	"	"	"	"	"		272	706	1.45	"	117,100	3.0	9.54	90.0*	12,545	80.1	
1-C	"	"	"	"	"	"	"	"		280	707	1.26	"	82,994	2.0	9.44	85.0*	12,450	79.2	
1-D	"	"	"	"	"	"	"	"		280	721	0.78	"	43,261	1.0	9.51	68.0*	12,780	74.4	
2-A	1928	160,000	2—1800–1200	cc	Brown Boveri	3	2	265		282	655	2.26	Non-extr.	167,737	3.0	10.65	105.0*/	13,550	77.8	N.E.L.A. Pub. No. 151, p. 42
2-B	"	"	"	"	"	"	"	"		279	633	1.35	"	102,576	2.0	10.13	88.0*	12,940	77.9	
2-C	"	"	"	"	"	"	"	"		280	658	0.94	"	61,876	1.0	10.13	75.0*	12,890	76.0	
2-D	"	"	"	"	"	"	"	"		278	600	0.87	"	35,150	0.6	11.31	72.0*	14,430	68.0	
3-AA	1928	110,000	2—1800	cc—30	Westinghouse	5	2	274.4		413	704	1.75	Extr.	110,020	5.0	10.29	274.4	11,520	78.0	N.E.L.A. Pub. No. 151, p. 37
3-BB	"	"	"	"	"	"	"	"		419	698	1.24	"	76,930	4.0	10.09	253.9	11,450	76.4	
3-CC	"	"	"	"	"	"	"	"		420	697	1.13	"	62,590	3.0	10.06	243.4	11,520	75.2	
3-DD	"	"	"	"	"	"	"	"		421	708	1.08	"	42,890	2.0	10.08	225.2	11,810	73.0	
3-EE	"	"	"	"	"	"	"	"		423	683	1.08	"	22,880	1.0	11.53	199.8	13,640	64.0	
4-AA	1928	94,000	2—1500	tc—21	Genl. Electric	3	4	393.2		415.1	713	1.33	Extr.	99,404	3.0	10.6	393.2	10,620	79.8	N.E.L.A. Pub. No. 078, p. 61
4-BB	"	"	"	"	"	"	"	"		414.7	688	1.582	"	72,706	2.0	10.63	365.7	10,800	80.8	
4-CC	"	"	"	"	"	"	"	"		415.4	687	1.015	"	74,519	2.0	10.36	366.0	10,520	80.0	
4-DD	"	"	"	"	"	"	"	"		415.7	704	1.651	"	56,322	1.0	10.52	346.3	10,990	79.2	
4-EE	"	"	"	"	"	"	"	"		420.2	704.6	1.052	"	58,653	1.0	10.1	345.4	10,560	79.2	
4-FF	"	"	"	"	"	"	"	"		419.7	713.8	1.493	"	26,950	0.5	11.24	301.1	12,300	70.0	
4-GG	"	"	"	"	"	"	"	"		418.2	717.8	0.745	"	29,122	0.5	10.52	300.5	11,580	70.4	
5-AA	1926	80,000	2—1800	cc—22	Westinghouse	3	4	392.2		400.8	679.4	2.3	Extr.	79,260	3.0	12.24	392.2	12,040	75.7	N.E.L.A. Pub. No. 151, p. 35
5-BB	"	"	"	"	"	"	"	"		406.3	688.4	1.86	"	63,960	2.0	11.49	370.5	11,630	76.4	
5-CC	"	"	"	"	"	"	"	"		412.5	691.8	1.58	"	45,230	1.0	11.26	340.7	11,770	74.0	
5-DD	"	"	"	"	"	"	"	"		413.6	689.4	1.35	"	18,360	0.4	12.57	288.7	13,800	63.0	
6-A	1929	75,000	1—1800	s—17	Genl. Electric	3	2	317		332.0	718	1.393	Non-extr.	74,200	3.0	9.45	84.0	12,500	78.2	N.E.L.A. Pub. No. 078, p. 57
6-B	"	"	"	"	"	"	"	"		334.0	706	1.177	"	64,860	2.0	9.13	81.0	12,050	79.3	
6-C	"	"	"	"	"	"	"	"		333.0	716	1.139	"	50,240	1.0	8.98	83.0	11,970	80.7	
6-D	"	"	"	"	"	"	"	"		341.0	722	1.022	"	20,160	0.4	10.10	79.0	13,550	70.6	
7-A	1927	65,000	1—1800	s—17	Genl. Electric	3	3	308		365.3	736.8	2.232	Non-extr.	72,476	2.9	9.39	99.5	12,370	81.2	N.E.L.A. Pub. No. 078, p. 55
7-B	"	"	"	"	"	"	"	"		369.7	707.7	1.886	"	52,437	2.0	9.37	97.0	12,210	81.0	
7-C	"	"	"	"	"	"	"	"		368.2	749.6	1.524	"	43,853	1.0	9.0	85.2	12,050	80.4	
7-D	"	"	"	"	"	"	"	"		374.1	748.5	1.494	"	18,237	0.4	10.53	88.0	14,050	68.2	
7-EE	"	"	"	"	"	"	"	"		367.9	722.1	1.761	Extr.	67,494	3.0	10.24	308.0	11,103	81.0	
7-FF	"	"	"	"	"	"	"	"		370.9	700.6	1.565	"	48,930	2.0	10.02	283.9	10,990	81.0	
7-GG	"	"	"	"	"	"	"	"		372.8	733.7	1.409	"	40,607	1.0	9.9	271.7	11,138	79.0	
7-HH	"	"	"	"	"	"	"	"		379.1	739.2	1.553	"	18,174	0.4	11.57	213.7	13,626	66.2	
8-A	1926	60,000	1—1500	s	Genl. Electric	2	3	340		366.0	728	1.29	Non-extr.	59,850	1.8	9.36	74.8	12,520	77.0	N.E L.A. Pub. No. 078, p. 54
8-B	"	"	"	"	"	"	"	"		369.0	729	1.05	"	49,750	1.4	9.12	68.1	12,250	77.3	
8-C	"	"	"	"	"	"	"	"		372.0	719	1.05	"	40,150	1.0	9.06	62.4	12,170	78.1	
8-D	"	"	"	"	"	"	"	"		371.0	694	1.02	"	19,650	0.5	10.24	53.1	13,700	70.1	
9-AA	1928	"	1—1800	"	Westinghouse	3	4	350		299.2	705	0.82	Two-stage extr.	46,900	2.0	9.435	194.0	11,380	78.0	.....
10-AA	1930	"	"	"	Allis-Chalmers	"	"	"		292.7	708	0.79	Two-stage extr.	40,200	1.0	9.26	186.0	11,010	79.5	.....
11-A	1918	60,000	3—1800	cc.	Westinghouse	4	None	Condensate temp.		216.2	547.3	2.67	Non-extr.	67,168	4.0	12.175	111.0*/	14,760	77.3	A.S.M.E. Trans., Vol. 44, p. 377
11-B	"	"	"	"	"	"	"	"		218.7	545.1	1.91	"	60,456	3.0	11.73	99.0*	14,350	76.4	
11-C	"	"	"	"	"	"	"	"		220.4	523.9	1.97	"	47,647	2.0	11.91	100.0*	14,430	76.6	
11-D	"	"	"	"	"	"	"	"		219.4	528.2	1.73	"	40,332	1.0	11.69	95.0*	14,350	76.8	
11-E	"	"	"	"	"	"	"	"		219.0	516.6	1.55	"	25,274	0.6	12.4	90.0*	15,100	71.6	

TABLE 3 RESULTS OF TESTS ON LARGE TURBINE-GENERATORS INSTALLED IN UNITED STATES (*Continued*)

	Characteristics of Turbine Installation								Test Conditions									
Test No.	Date installed	Normal rating, kw	Number of cylinders and speed in rpm	Type and number of stages	Make	Admission valves	Extraction points	Feed temp. for maximum load, F	Steam at throttle: Abs. pressure, lb	Steam at throttle: Temperature, F	Absolute back pressure at exhaust, in. Hg	Extraction or non-extraction	Generator net output, kw	Number of admission valves open (approx.)	Throttle water rate, lb per kw-hr of net output	Final feed temperature, F	Overall heat rate, Btu per kw-hr net output	Average overall Rankine cycle eff. ratio, %	Reference to detail report of test
12-A	1926	50,000	2—1800	tc—20	Genl. Electric	3	4	335	390.0	675	1.0	Non-extr.	60,000	...	9.0	79.0*/	11,710	80.2	N.E.L.A. Pub. No. 267-78, p. 41
12-B	"	"	"	"	"	"	"	"	390.0	675	1.0	"	50,000	...	9.12	79.0*	11,860	79.2	
12-C	"	"	"	"	"	"	"	"	390.0	675	1.0	"	40,000	...	9.0	78.0*	11,720	80.2	
12-D	"	"	"	"	"	"	"	"	390.0	675	1.0	"	27,000	...	9.5	77.0*	12,390	76.0	
12-E	"	"	"	"	"	"	"	"	390.0	675	1.0	"	12,000	...	11.0	75.0*	14,370	65.6	
13-AA	1928	50,000	1—1800	s	Genl. Electric	3	4	347	322.9	693.8	1.001	Extr.	53,943	3.0	10.94	347.0	11,462	75.5	N.E.L.A. Pub. No. 289-98, p. 48
13-BB	"	"	"	"	"	"	"	"	324.3	698.0	1.044	"	38,368	2.0	10.23	318.3	10,842	80.3	
13-CC	"	"	"	"	"	"	"	"	332.4	720.4	0.996	"	20,145	0.7	10.33	281.0	11,859	73.7	
13-DD	"	"	"	"	"	"	"	"	330.3	701.5	1.939	"	38,617	2.0	10.71	321.6	11,400	80.2	
13-EE	"	"	"	"	"	"	"	"	335.1	718.4	2.004	"	20,102	0.7	11.12	283.0	12,468	73.7	
14-A	1924	50,000	2—1200	tc	Westinghouse	3	2	300	288.7	625	1.30	Non-extr.	44,079	2.0	10.29	86.0*/	13,100	76.6	N.E.L.A. Pub. No. 289-98, p. 39
14-B	"	"	"	"	"	"	"	"	291.4	632.8	1.0	"	35,995	1.0	9.993	77.0*	12,860	76.3	
14-C	"	"	"	"	"	"	"	"	295.0	615.3	0.64	"	13,339	0.4	11.128	62.0*	14,370	66.1	
14-DD	"	"	"	"	"	"	"	"	284.5	609.7	1.1	Extr.	43,842	2.5	11.97	293.9	12,340	74.3	
14-EE	"	"	"	"	"	"	"	"	285.8	605.5	1.1	"	39,427	2.0	11.68	285.8	12,210	75.4	
14-FF	"	"	"	"	"	"	"	"	287.5	606	0.86	"	30,261	0.9	11.1	266.8	11,990	75.4	
14-GG	"	"	"	"	"	"	"	"	291.2	579.3	0.4	"	13,184	0.4	12.41	228.5	13,830	64.2	
15-A	1924	50,000	1—1200	s—21	Genl. Electric	2	2	300	280.2	627.8	1.709	Non-extr.	50,390	1.6	10.62	89.3	13,510	76.9	N.E.L.A. Pub. No. 289-98, p. 62
15-B	"	"	"	"	"	"	"	"	282.5	612.6	1.098	"	34,680	0.95	10.26	79.2	13,070	76.0	
15-C	"	"	"	"	"	"	"	"	283.9	614.4	0.871	"	24,730	0.7	10.53	73.7	13,490	72.3	
15-D	"	"	"	"	"	"	"	"	284.9	577.5	0.73	"	15,090	0.5	11.59	68.4	14,680	65.8	
16-A	1924	50,000	1—1200	s—21	Genl. Electric	2	3	233	391.6	683	1.02	Non-extr.	50,300	2.0	9.52	81.0	12,400	75.6	N.E.L.A. Pub. No. 078, p. 46
16-B	"	"	"	"	"	"	"	"	392.6	681	1.04	"	48,100	1.0	9.24	82.0	12,030	78.2	
16-C	"	"	"	"	"	"	"	"	395.6	678	1.03	"	34,200	0.7	9.65	81.0	12,550	74.7	
16-D	"	"	"	"	"	"	"	"	399.6	670	0.99	"	24,500	0.5	10.11	80.0	13,120	70.0	
16-EE	"	"	"	"	"	"	"	"	389.6	672	0.98	Extr.	49,300	2.0	9.78	233.9	11,160	77.8	
16-FF	"	"	"	"	"	"	"	"	394.6	673	1.02	"	45,700	1.0	9.72	230.0	11,100	78.2	
16-GG	"	"	"	"	"	"	"	"	393.6	685	1.03	"	34,400	0.75	9.99	218.0	11,610	75.0	
16-HH	"	"	"	"	"	"	"	"	400.6	683	0.92	"	24,600	0.5	10.49	201.0	12,380	70.0	
19-A	1920	45,000	1—1200	s—21	Genl. Electric	2	None	Condensate temp.	214.7	587.9	1.0	Non-extr.	45,000	...	10.91	78.0*/	13,840	73.5	N.E.L.A. Pub. No. 25-103, p. 21
19-B	"	"	"	"	"	"	"	"	214.7	587.9	1.0	"	40,000	...	10.82	77.0*	13,720	74.1	
19-C	"	"	"	"	"	"	"	"	214.7	587.9	1.0	"	36,000	1.0	10.68	76.0*	13,550	75.1	
19-D	"	"	"	"	"	"	"	"	214.7	587.9	1.0	"	30,000	0.8	10.77	75.0*	13,670	74.5	
19-E	"	"	"	"	"	"	"	"	214.7	587.9	1.0	"	20,000	0.6	11.26	73.0*	14,320	71.2	
		kva																	
20-AA	1926	43,750	1—1800	s—[b]	Westinghouse	3	4	364	368.0	688	1.09	Extr.	43,050	2.5	11.6	363.9	12,100	73.2	N.E.L.A. Pub. No. 289-98, p. 42
20-BB	"	"	"	"	"	"	"	"	375.0	699	0.77	"	30,310	0.9	11.5	333.9	12,640	67.2	
20-CC	"	"	"	"	"	"	"	"	375.0	679	1.2	"	19,700	0.6	12.31	313.3	14,050	64.0	
20-D	"	"	"	"	"	"	"	"	340.0	676	1.48	Non-extr.	42,890	2.0	10.23	85.6	13,270	74.4	N.E.L.A. Pub. No. 289-98, p. 48
20-E	"	"	"	"	"	"	"	"	341.0	675	1.5	"	28,160	0.9	10.4	88.6	13,470	73.2	
20-F	"	"	"	"	"	"	"	"	340.0	675	1.62	"	16,920	0.5	11.26	90.8	14,560	68.4	
20-G	"	"	"	"	"	"	"	"	342.0	694	1.03	"	42,840	1.9	9.91	72.4	13,100	73.0	
20-H	"	"	"	"	"	"	"	"	342.0	678	1.0	"	28,190	0.9	10.08	75.7	13,200	72.4	
20-I	"	"	"	"	"	"	"	"	337.0	668	1.02	"	16,330	0.5	10.91	78.0	14,200	66.8	
		kva																	
21-AA	1927	52,500	1—1800	s—[b]	Westinghouse	3	4	351	367.7	712	1.86	Extr.	44,200	2.2	11.46	350.9	12,070	74.2	
21-BB	[c]	"	"	"	"	"	"	"	367.7	712	1.49	"	30,410	0.9	11.28	330.1	12,270	72.4	N.E.L.A. Pub. No. 289-98, p. 43
21-CC		"	"	"	"	"	"	"	368.6	713	1.36	"	21,510	0.6	11.75	314.0	13,300	67.0	
		kw																	
22-A	1925	40,000	1—1800	s	Westinghouse	3	4	355	307.6	697.8	1.635	Non-extr.	40,345	2.0	10.0	94.0	13,040	76.8	N.E.L.A. Pub. No. 267-78, p. 22
22-B	"	"	"	"	"	"	"	"	306.5	685.7	1.439	"	31,850	1.0	10.1	89.7	13,150	75.4	
22-C	"	"	"	"	"	"	"	"	316.9	695.1	1.513	"	22,197	0.7	10.604	90.5	13,860	71.7	
22-D	"	"	"	"	"	"	"	"	315.7	691.8	1.549	"	15,096	0.5	11.331	90.4	14,750	67.5	
22-E	"	"	"	"	"	"	"	"	303.2	684.5	1.938	"	39,007	2.0	10.334	101.4	13,310	76.6	
22-F	"	"	"	"	"	"	"	"	310.8	676.0	1.969	"	30,188	1.0	10.554	99.7	13,600	75.2	

TABLE 3 RESULTS OF TESTS ON LARGE TURBINE-GENERATORS INSTALLED IN UNITED STATES (*Continued*)

	Characteristics of Turbine Installation									Test Conditions									
Test No.	Date installed	Normal rating, kw	Number of cylinders and speed in rpm	Type and number of stages	Make	Admission valves	Extraction points	Feed temp. for maximum load, F		Steam at throttle: Abs. pressure, lb	Steam at throttle: Temperature, F	Absolute back pressure at exhaust, in. Hg	Extraction or non-extraction	Generator net output, kw	Number of admission valves open (approx.)	Throttle water rate, lb per kw-hr of net output	Final feed temperature, F	Overall heat rate, Btu per kw-hr net output	Average overall Rankine cycle eff. ratio, %	Reference to detail report of test
23-A	1928[a]	40,000	1—1800	s—17	Genl. Electric	2	3 or 4	...		340.0	706	1.5	Non-extr.	40,330	2.0	9.79	86.9	12,860	76.5	N.E.L.A. Pub. No. 078, p. 42
23-B	"	"	"	"	"	"	"	...		338.0	702	1.49	"	33,290	1.0	9.64	80.3	12,710	77.8	
23-C	"	"	"	"	"	"	"	...		337.0	709	1.51	"	14,670	0.5	10.76	58.8	14,450	69.6	
23-D	"	"	"	"	"	"	"	...		343.0	716	1.16	"	40,260	2.0	9.61	78.0	12,750	75.2	
23-E	"	"	"	"	"	"	"	...		336.0	710	1.05	"	33,420	1.0	9.41	75.2	12,490	76.6	
23-F	"	"	"	"	"	"	"	...		337.0	716	0.97	"	15,070	0.5	10.28	55.3	13,900	69.1	
24-AA	1928[a]	40,000	1—1800	s—17	Genl. Electric	2	4	353		363.0	710	1.0	Extr.	40,040	2.0	10.9	353.0[f]	11,340	75.0	N.E.L.A. Pub. No. 078, p. 39
24-BB	"	"	"	"	"	"	"	"		375.0	702	0.79	"	32,980	1.0	10.57	335.0[f]	11,120	74.9	
24-CC	"	"	"	"	"	"	"	"		373.0	683	0.68	"	14,690	0.5	10.9	287.0[f]	12,610	67.7	
25-A	1924	37,800	1—1500	s—20	Genl. Electric	2	3	300		376.5	718.4	1.165	Non-extr.	36,300[g]	1.5	9.33	84.4	12,340	76.7	N.E.L.A. Pub. No. 25–103, p. 22
25-B	"	"	"	"	"	"	"	"		379.4	718	0.911	"	28,120[g]	1.0	9.216	77.2	12,240	75.7	
25-C	"	"	"	"	"	"	"	"		381.2	713.7	0.703	"	15,555[g]	0.6	9.876	72.4	13,150	69.0	
26-A	1924	35,000	1—1800	s—25[b]	Westinghouse	3	4	365		311.9	688	1.17	Non-extr.	41,495	2.8	10.01	83.0	13,110	74.1	N.E.L.A. Pub. No. 25–103, p. 26
26-B	"	"	"	"	"	"	"	"		316.4	678	0.93	"	33,584	2.0	9.88	76.0	12,950	74.0	
26-C	"	"	"	"	"	"	"	"		315.4	681	0.94	"	27,018	1.0	9.74	76.0	12,780	74.8	
26-D	"	"	"	"	"	"	"	"		318.2	680	0.85	"	21,137	0.8	10.025	72.0	13,200	72.0	
26-E	"	"	"	"	"	"	"	"		320.8	682	0.92	"	15,941	0.6	10.56	74.0	13,870	74.0	
26-F	"	"	"	"	"	"	"	"		318.6	683	1.6	"	9,071	0.4	12.5	90.0	16,240	61.7	
29-A	1920	30,000	1—1800	s—17	Genl. Electric	2	None	Condensate temp.		214.7	587.9	1.0	Non-extr.	30,000	...	11.02	78.0*[f]	13,990	72.8	N.E.L.A. Pub. No. 25–103, p. 23
29-B	"	"	"	"	"	"	"	"		"	"	"	"	22,000	1.0	10.76	77.0*	13,800	74.5	
29-C	"	"	"	"	"	"	"	"		"	"	"	"	18,000	0.8	10.92	75.0*	13,870	73.4	
29-D	"	"	"	"	"	"	"	"		"	"	"	"	13,000	0.6	11.6	73.0[b]	14,860	69.2	
30-A	1923	30,000	1—1800	s—17	Genl. Electric	2	1	Not known		323.2	691.5	1.505	Non-extr.	29,487	1.0	9.72	90.0	12,660	78.4	N.E.L.A. Pub. No. 25–103, p. 6
30-B	"	"	"	"	"	"	"	"		323.7	678.5	0.987	"	29,653	1.0	9.52	78.7	12,450	77.0	
30-C	"	"	"	"	"	"	"	"		320.2	684.8	0.745	"	29,477	1.0	9.38	71.4	12,360	75.7	
30-D	"	"	"	"	"	"	"	"		320.1	693.8	0.61	"	29,651	1.0	9.33	63.9	12,420	74.2	
30-E	"	"	"	"	"	"	"	"		322.0	682.6	0.572	"	29,609	1.0	9.4	61.7	12,450	74.0	
30-F	"	"	"	"	"	"	"	"		324.6	699.1	1.485	"	22,534	0.75	10.0	87.6	13,100	75.4	
30-G	"	"	"	"	"	"	"	"		324.3	692.9	1.005	"	22,541	0.75	9.72	78.2	12,800	74.8	
30-H	"	"	"	"	"	"	"	"		323.9	703.9	0.749	"	22,804	0.75	9.45	71.2	12,550	74.2	
30-I	"	"	"	"	"	"	"	"		324.7	693.9	0.463	"	22,995	0.75	9.45	58.1	12,640	71.3	
30-J	"	"	"	"	"	"	"	"		327.0	702.5	1.516	"	14,822	0.5	10.97	83.4	14,450	68.8	
30-K	"	"	"	"	"	"	"	"		328.0	685.5	0.987	"	14,834	0.5	10.4	74.0	13,700	70.0	
30-L	"	"	"	"	"	"	"	"		327.4	700.2	0.449	"	14,609	0.5	9.76	56.3	13,100	68.7	
30-M	"	"	"	"	"	"	"	"		325.9	672.2	1.493	"	9,845	0.35	11.9	80.6	15,500	64.5	
30-N	"	"	"	"	"	"	"	"		325.9	686.8	0.993	"	9,911	0.35	11.23	71.3	14,700	64.9	
30-O	"	"	"	"	"	"	"	"		327.9	677.6	0.4072	"	10,060	0.35	10.36	53.9	13,800	65.0	
31-A	1928	30,000	1—1800	s	Allis-Chalmers	2	3	240		345.6	702.7	2.48	Non-extr.	30,592	1.5	9.66	108.0	12,470	82.2	N.E.L.A. Pub. No. 289–98, p. 62
31-B	"	"	"	"	"	"	"	"		351.3	697.1	2.22	"	24,696	1.0	9.58	103.9	12,380	81.8	
31-C	"	"	"	"	"	"	"	"		345.4	678.0	1.875	"	15,484	0.7	10.31	98.3	13,270	75.5	
31-D	"	"	"	"	"	"	"	"		349.1	683.6	1.955	"	8,164	0.4	11.99	100.0	15,450	65.1	
32-A	1915	30,000	2—750–1500	cc	Westinghouse	2	None	Condensate temp.		217.1	515.4	1.377	Non-extr.	31,986	2.0	11.68	85.0*[f]	14,280	75.2	A.S.M.E. Trans., Vol. 38, p. 655
32-B	"	"	"	"	"	"	"	"		217.9	524.3	1.239	"	26,153	1.0	11.38	81.0*	14,010	75.8	
32-C	"	"	"	"	"	"	"	"		221.1	501.5	1.256	"	22,432	0.85	11.76	80.0*	14,340	74.2	
32-D	"	"	"	"	"	"	"	"		224.3	493.2	1.589	"	16,342	0.65	12.64	87.0[b]	15,250	71.5	
33-A	1924	20,000	1—1800	s—[b]	Westinghouse	3	3	300		292.5	733.6	1.067	Non-extr.	24,072	1.9	9.52	70.3	12,830	75.7	N.E.L.A. Pub. No. 25–103, p. 12
33-B	"	"	"	"	"	"	"	"		294.5	739.9	0.992	"	20,275	1.1	9.44	71.5	12,750	74.8	
33-C	"	"	"	"	"	"	"	"		290.9	750.1	1.012	"	16,339	0.9	9.43	74.0	12,760	75.3	
33-D	"	"	"	"	"	"	"	"		288.2	728.4	1.003	"	11,991	0.7	9.94	74.2	13,350	72.4	
33-E	"	"	"	"	"	"	"	"		288.7	741.5	0.999	"	8,286	0.55	10.55	73.3	14,230	67.6	
34-A	1924	20,000	1—1800	s—9	Genl. Electric	9	3	280		199.2	485.6	1.494	Non-extr.	23,005	...	12.4	91.0*[f]	14,900	73.7	N.E.L.A. Pub. No. 25–103, p. 24
34-B	"	"	"	"	"	"	"	"		200.9	489.5	1.489	"	15,892	...	12.26	90.0*	14,760	74.2	
34-C	"	"	"	"	"	"	"	"		200.7	490.5	1.512	"	10,296	...	12.8	88.0*	15,450	71.2	
34-D	"	"	"	"	"	"	"	"		203.2	499.6	1.518	"	5,154	...	14.8	86.0*	17,950	62.8	

TABLE 3 RESULTS OF TESTS ON LARGE TURBINE-GENERATORS INSTALLED IN UNITED STATES (*Continued*)

Test No.	Characteristics of Turbine Installation: Date installed	Normal rating, kw	Number of cylinders and speed in rpm	Type and number of stages[a]	Make	Admission valves	Extraction points	Feed temp. for maximum load, F	Test Conditions: Steam at throttle, Abs. pressure, lb	Steam at throttle, Temperature, F	Absolute back pressure at exhaust, in. $Hg	Extraction or non-extraction	Generator net output, kw	Number of admission valves open (approx.)	Throttle water rate, lb per kw-hr of net output	Final feed temperature, F	Overall heat rate, Btu per kw-hr net output	Average overall Rankine cycle eff. ratio, %	Reference to detail report of test
35-A	1928[a]	20,000	1—1800	8—14	Genl. Electric	2	3	340[a]	379.0	721	1.05	Non-extr.	20,050	...	9.18	89.0[a]	12,180	77.0	N.E.L.A. Pub. No. 078, p. 68
35-B	"	"	"	"	"	"	"	"	388.0	718	1.13	"	15,060	...	9.47	81.0[a]	12,540	75.0	
35-C	"	"	"	"	"	"	"	"	378.0	706	1.07	"	10,430	...	10.08	78.0[a]	13,320	70.9	
35-D	"	"	"	"	"	"	"	"	384.0	714	1.02	"	5,320	...	11.48	74.0[a]	15,250	61.6	
36-AA	1928	15,000	1—1500	8	Allis-Chalmers	2	3	342	415.7	725	2.61	Extr.	15,235	1.3	11.59	342.0	12,505	75.2	N.E.L.A. Pub. No. 289–98, p. 66
36-BB	"	"	"	"	"	"	"	"	417.0	725	2.08	"	11,158	0.9	11.29	320.0	12,387	73.6	
36-CC	"	"	"	"	"	"	"	"	416.3	725	1.66	"	7,463	0.6	11.84	298.5	13,263	67.6	
37-A	1924	15,000	1—1800	8	Allis-Chalmers	...	...	...	269.5	621.7	1.542	Non-extr.	16,264	...	10.85	92.0	13,750	75.0	N.E.L.A. Pub. No. 25–103, p. 34
37-B	"	"	"	"	"	...	...	...	271.3	619.8	1.493	"	15,005	...	10.7	91.0	13,550	75.7	
37-C	"	"	"	"	"	...	...	...	271.9	618.0	1.41	"	13,860	...	10.53	88.0	13,360	76.2	
37-D	"	"	"	"	"	...	...	...	272.9	616.7	1.275	"	11,267	...	10.57	84.0	13,460	75.2	
37-E	"	"	"	"	"	...	...	...	274.1	604.1	1.115	"	7,582	...	11.2	78.0	14,260	70.4	
38-A	1928	6,000	1—3600	8	Allis-Chalmers	2	4	Not known	253.2	548.7	1.58	Non-extr.	6,156	1.4	11.23	92.6	13,800	76.5	N.E.L.A. Pub. No. 289–98, p. 68
38-B	"	"	"	"	"	"	"	"	251.1	537.7	1.55	"	4,902	1.0	11.02	89.4	13,550	78.2	
38-C	"	"	"	"	"	"	"	"	252.6	530.8	1.4	"	4,065	0.85	11.32	87.6	13,850	75.5	
38-D	"	"	"	"	"	"	"	"	244.2	549.8	1.92	"	3,024	0.65	12.35	98.6	15,100	71.4	
38-E	"	"	"	"	"	"	"	"	254.6	525.2	1.19	"	1,176	0.35	15.42	81.0	18,900	54.4	

[a] About. [b] Baumann multi-exhaust. [c] Except for larger generator, this unit is same as 43,750-kva unit reported upon. [d] Types of turbine construction are indicated as follows: s, single-cylinder; tc, tandem-compound; cc, cross-compound. [e] Approximate. [f] In these cases (indicated by *), the condensate temperature was not recorded, and a value has been assumed as related to the exhaust-steam temperature and the load. [g] Performance of 37,800-kw turbine is based on gross generation, which includes approximately 120 kw of excitation load.

some of the turbines operating with steam delivered to the throttle at 550 lb per sq in. gage and 725 F and reheat of the steam by flue gas to 725 F would show on test an overall Rankine cycle efficiency ratio of from 83 to 84 per cent.

The extraction of steam for the heating of feedwater tends to improve the overall Rankine cycle efficiency ratio. The accompanying trend toward the development of appreciably higher capacities from existing turbine frames resulting in higher leaving losses has largely canceled the potential improvement in Rankine cycle efficiency ratio by reason of extraction of steam for feedwater heating. The higher blade and disk friction losses and the greater blade-tip leakage losses with high-pressure steam tend to a reduction in Rankine cycle efficiency ratio, but this effect has been offset by the trend toward higher steam temperatures, with the resulting reduction in friction losses in the high-pressure section of the turbine by reason of reduced steam density and a very marked reduction in friction losses in the low-pressure blading by reason of reduced moisture content in the steam.

It should be appreciated that high overall Rankine cycle efficiency ratio is not an end in itself, but simply one of the means toward obtaining the desirable result of a low unit fuel consumption. It is relatively easy to obtain a high Rankine cycle efficiency ratio with a poor vacuum at the turbine exhaust, but the poor vacuum makes it difficult to obtain a low unit fuel consumption.

If a high Rankine cycle efficiency ratio is obtained as the result of operating the turbine at less than its maximum economic capacity or by design features which increase the time required for warming up and applying load to the turbine or which increase outage time and reduce turbine reliability, then it is easily possible that the debits will more than offset the credits.

Notes are included in the Appendix which will serve to answer most of the questions that may arise in regard to the turbine-water-rate tests and the methods used in analyzing the results of these tests.

Mercury-Vapor Installations

A 10,000-kw mercury-vapor turbine is installed in the South Meadow Station at Hartford, Conn., and two 20,000-kw mercury-vapor turbines are now being built. One of these turbines will be installed in the Kearny Station and the other in a new power station being built at Schenectady. Operation at the South Meadow Station during the 12-month period from February 4, 1930, to February 4, 1931, was very satisfactory. Since the latter date this unit has been out of service to permit of retubing the boiler and at the same time to correct an inherent defect in the design of the cores for the boiler tubes. The indications are that these mercury-vapor turbines, operating together with the steam-turbine plants which absorb the steam from the condensers for the mercury-vapor turbines, will generate energy for somewhat less than 10,000 Btu per kw-hr of net station send-out.

Standardization

As yet there has been little indication in American turbine-plant practice of any desire for or real attempt at standardization. The most interesting steps along this line have been the installation by the Detroit Edison Company of nine duplicate units of 50,000 kw capacity and the installation of six duplicate units of 40,000 kw capacity by companies associated with the American Gas & Electric Company.

It would appear that in the years immediately ahead the greatest promise for the reduction in cost of energy on the station bus lies in some measure of standardization, particularly as relating to steam conditions and the operating cycle, and in simplification of design which will result in a substantial reduction in investment.

Appendix

NOTES ON WATER-RATE TESTS ON TURBINE-GENERATORS

WITH a few exceptions the water-rate tests which are summarized in Table 3 were published in detail in the reports of the Turbine Subcommittee of the Prime Movers Committee, National Electric Light Association, for the years 1925, 1927, 1929, 1930, and 1931. There have been included, for comparison with these tests made in recent years, summaries of tests Nos. 11 and 32 made in 1916 and 1921 on the 30,000-kw and 60,000-kw cross-compound turbines in the 74th Street Station of the Interborough Rapid Transit Company. All of these tests were conducted by the operating companies.

For tests Nos. 11, 26, 31, and 35 the condensate was measured by volume in carefully calibrated tanks. For tests Nos. 16 and 25 the steam flow to the turbine was measured by flow meters, and for test No. 16 the steam flow meter was calibrated by means of a weighed water test. For test No. 4 the 94,000-kw turbine with its boilers was isolated and the feedwater going to the boilers was measured by means of venturi meters. For tests Nos. 12 and 37 methods of measurement are not reported. For the other 25 tests, among which are included the tests on a 110,000-kw turbine, a 160,000-kw turbine, and a 165,000-kw turbine, the condensate from the main condensers was weighed.

For tests Nos. 3 and 5 the condensate from the extraction heaters was measured by volume in carefully calibrated tanks. For the other extraction tests the steam delivered to each of the feedwater heaters was computed from a knowledge of the weight of water passing through the heater, the temperatures of the water at the inlet and outlet from the heater, the pressure of the steam as extracted from the turbine, the temperature of the heater drips, and the heat content released by drips cascaded into the heater under consideration from other heaters. For the purpose of these computations in most cases the heat content of the steam extracted from the turbine was estimated from expansion curves furnished by the manufacturers for each different load at which a test was made. The quality of the extracted steam in the superheat zone was checked by the measurement of temperature at the extraction point.

Each test was from one to three hours' duration, and sufficient time was allowed between tests to permit of stable conditions being established after a change in load, before starting the next test.

In practically all cases the water as actually weighed was corrected for changes in level in the condenser hot well, the surge tank, and the deaerator; the condenser leakage was checked and a correction was made if necessary; and corrections were made so as to exclude the low-pressure gland-seal leakage water and the condensate from the inter- and after-condensers of the steam-jet air-removal pumps from the condensate as weighed and charged to the turbine. The indications are that in no case was the steam for the steam-jet air-removal pumps charged to the turbine and that in every case the steam going to the high-pressure packing was charged to the turbine. In some cases the leak-off from the high-pressure packing was expanded through the low-pressure stages of the turbine. In certain cases this leak-off steam was exhausted to atmosphere or to the condenser, and in some of these cases, as for example test No. 30, a correction was made for the power that would have been generated if this steam had been expanded through the low-pressure stages of the turbine.

With the exception of tests Nos. 12, 19, and 29, actual test values of steam consumption are recorded in Table 3. For these three tests the actual test values have been corrected to standard throttle steam conditions, and for test No. 12 the tested steam consumption has been corrected to a back-pressure of 1 in. Hg at the turbine exhaust.

With one exception, the excitation current and the power for driving the generator fans are deducted from gross generation to arrive at the net output on which the performance is based. The performance for test No. 25 is based on gross generation, which includes excitation. In no case was the power for driving the condenser auxiliaries deducted in arriving at net output.

The average overall Rankine cycle efficiency ratio has been calculated in every case on the basis of an equivalent of 3412 Btu per kw-hr of net output at the generator leads, the steam pressure and temperature at the inlet to the turbine throttle, the steam pressure at the turbine exhaust, the steam pressures at the several extraction openings from the turbine, the assumption that all steam (both that extracted from the several stages of the turbine and that which passes to the condenser) is expanded on the same average overall Rankine efficiency ratio, and the use of the Keenan steam tables. The assumption of a constant efficiency for all stages of the turbine is known to be incorrect, but the extent to which this incorrect assumption affects the end results is negligible. In general, it is true for most turbines that the efficiency of the high-pressure stages is relatively low, the efficiency of the intermediate-pressure stages very high, and the efficiency of the low-pressure stages in the moisture zone somewhat lower. In general, it is true, at least for the larger turbines, that the steam extracted for feedwater heating is expanded on a higher average overall Rankine cycle efficiency ratio than the steam which goes to the condenser. As an exception the Rankine cycle efficiency ratio for test No. 36 has been calculated on the basis of the pressures at which extracted steam is delivered to the feedwater heaters, the pressure drop through the piping being charged to the turbine.

The Btu per kw-hr of net output for non-extraction tests, as given in Table 3, is based on the water rate, the heat content of the steam corresponding to the pressure and temperature ahead of the turbine throttle, and the heat content of the condensate as it leaves the condenser hot well, the heat contents being based on the Keenan steam tables. In cases where the condensate temperature was not recorded, a value has been assumed as related to the exhaust-steam temperature and the load, and in these cases the value of the condensate temperature recorded in Table 3 has been distinguished by an asterisk (*). It is to be appreciated that the values of Btu per kw-hr given in Table 3 for non-extraction tests are of somewhat academic interest. In many cases the turbines, though tested on a non-extraction basis, are actually operated normally on an extraction cycle, with an appreciably lower overall heat rate. In other cases the turbines in some of the older stations are normally operated on a non-extraction basis, but steam for a single stage of feedwater heating is expanded through moderately efficient turbines, which results in a somewhat reduced overall unit heat rate for the turbine plant.

The heat rate in Btu per kw-hr of net output would, for an extraction test in which the weight of water returned to the boilers is the same as the weight of steam delivered to the turbine throttle, be calculated in a very simple manner on the basis of the throttle water rate and the heat contents of steam at the throttle and feedwater at the outlet of the high-pressure heater. In the case of the partial-load test, particularly, it is quite often true that the weight of water returned to the boilers from the high-pressure heater is not the same as the weight of steam delivered to the turbine throttle during the period of the test. Correction must be made for the greater or lesser amount of heat that is delivered to the turbine throttle for use in heating feedwater, and correction must also be made in such cases for heat in the form of hot water stored in or withdrawn from the surge tank and deaerator during the period of the test. Another complication entered into the extraction tests Nos. 20 and 21; a small portion of the steam used for heating the feedwater was taken from auxiliary turbines. In order to make corrections for conditions such as those described, one must have a very thorough knowledge of all the details of the test. It was, for this reason, considered wise to accept and record in Table 3 the heat rates from the original test reports for all extraction tests, even though it was apparent that in a number of cases they were based on either Marks and Davis' or Goodenough's steam tables. It is probable that in many cases the heat rates would have been from 0.5 to 0.8 per cent higher than the values given in Table 3 if they had been calculated on the basis of the Keenan steam tables. It is to be clearly understood that though the heat rates recorded in Table 3 for the extraction tests are taken from the original test reports, the Rankine cycle efficiency ratios recorded have in every case been calculated on the basis of the Keenan steam tables. It is true in a number of cases that minor errors may have entered in, due to the fact that the quantities of extracted steam have been calculated on the basis of the older steam tables. It is believed, however, that these errors are not important as affecting the final results.

In connection with a number of turbine-generators on which extraction tests were run, provision was made for reclaiming in the condensate the heat losses from the generator air coolers, the turbine oil coolers, and the inter- and after-condensers for the steam-jet air pumps. This heat is accepted as a gift and goes to reduce somewhat the overall heat rate.

The designation in Table 3 of "Number of admission valves open" must be considered as rough. In most cases an even number, such as 2, may be taken as a fairly clear indication that the primary and secondary valves are wide open and that the tertiary valve is on the verge of opening. In a number of cases tests were made with the valves blocked in a fixed position.

In some cases the back pressure at the turbine exhaust was held constant for all loads by bleeding air into the air-pump suction for partial-load operation. In most cases, however, the tests were made with the lowest back pressure available and there was a progressive reduction in back pressure as the load decreased.

Discussion

A. G. CHRISTIE.[14] The paper presents such complete data that little is left for discussion. The purpose of adding a stage of reheating is to decrease the per cent of moisture at exhaust and thereby lessen blade erosion, which is frequently serious. A recent study made under the writer's direction indicates that the moisture content of the steam at the last row of blades is generally 1 per cent greater than usually assumed and may be even worse with turbines having high leaving losses. Data are at hand based upon careful tests indicating a comparatively small withdrawal of moisture at bleeders. A study of steam flow by means of polarized light now in progress at Johns Hopkins University indicates that the droplets formed by condensation are extremely small and do not appear to grow appreciably in size as condensation proceeds. It does not seem probable that these will be separated out in the moving blades by centrifugal force, and hence they can be removed only to a comparatively small extent at bleeder points.

On the third page the authors indicate that 200,000 kw will probably prove the largest economic size of turbine. It is always dangerous to predict turbine sizes. The writer recalls that 20 years ago one of the leading power engineers stated that no conceivable justification could be made in the future for a turbine exceeding 25,000 kw capacity.

The authors present good arguments for the more extended use of multi-cylinder turbines. When the advantages are better understood, there will undoubtedly be more two-cylinder 3600-rpm units installed in sizes up to 40,000 kw.

On the fourth page the discussion of the influence and importance of leaving losses on single-cylinder turbines is well presented. Engineers must become more familiar with the effect of leaving loss upon turbine economy and more particularly upon the economic capacity of the casing.

One criticism might be offered of the authors' use of "per cent" to express leaving losses. This refers to a certain per cent of the total adiabatic heat drop, which may vary by 100 or more Btu under different steam conditions. Also, when a stage of reheating is used, to what heat drop is the leaving loss referred? In the writer's opinion engineers should adopt the practice of

[14] Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

expressing leaving losses in Btu per pound of steam to exhaust, which signifies a definite value in every case and also indicates a fixed leaving velocity from the blades.

The paragraph on the fifth page covering turbine speeds omits reference to the 15,000-kw, 3600-rpm single-cylinder Parsons turbine that has been in operation for several years at the Municipal Power Plant, Regina, Sask. This was designed for 400 lb gage, 750 F, and 29 in. vacuum. Two years ago C. A. Parsons & Co. submitted proposals for the Municipal Plant, Edmonton, Alta., for single-cylinder 3600-rpm turbines of 20,000 and 25,000 kw. capacity.

On the sixth page reference is made to the nine duplicate units in the Detroit Edison system. While this duplication has certain advantages, no attempt is made to apply the economic gains in turbine construction and operation since the first of these units was designed nearly a decade ago.

The authors stress the decrease in heat to the condenser per kilowatt output which results partly from increased heat drop and partly from bleeding. The problem before engineers is to develop an economic use for great quantities of heat at comparatively low temperatures and further reduce this loss to condensers. The Germans are using some of this steam to heat greenhouses. A study of Fig. 8 indicates that further hope for decreased heat consumption lies in either the reduction of the loss of heat in flue gases or in the heat to the condenser. The reduction of the latter offers opportunities for much thought.

Various phases of condenser design and operation are still undergoing rapid development. Treatment of circulating water will be an important factor in the future design of many power plants. Regarding variable-speed circulating pumps, the writer has found in most cases that the operators seldom avail themselves of the speed changes, but operate at full speed. Hence in later stations constant-speed motors have been used.

There seems little practical justification for a solid wall between boiler and turbine room in a modern plant. Its omission lessens first cost. Boiler rooms are now maintained practically as clean as turbine rooms. The dust in the turbine room does little damage, with totally inclosed generators and turbine governor gear.

The writer collected performance data on 16 modern American stations for a paper prepared for the International Electrical Congress, Paris. The figures on these stations check very closely the data given by the authors on the thirteenth page, except in the case of Deepwater Station, where the performance supplied to the writer for Dec. 1, 1930, to Dec. 1, 1931, was 10,850 Btu per net kw send-out.

The authors have presented valuable data upon water-rate tests of turbo-generators. The data plotted in Figs. 9 and 10 show considerable scattering of values. This is to be expected from the facts of the case. In a paper on "Trends in Steam Turbine Development," delivered by the writer at the Fourth National Meeting of the A.S.M.E. Fuels Division at Chicago in 1931, the following statement appeared:

"Efficiency in turbines is largely a matter of price. Each manufacturer offers units which will give about the same efficiency and will sell at about the same price as those of his competitors. Turbines of higher efficiency than these commercial units can be built if one is prepared to pay the higher costs of these improved designs. Load conditions, early supercession by more modern units, changing steam conditions, low fuel costs, and other factors make it uneconomical to purchase a turbine of the highest efficiency. Nevertheless, the performance of steam turbines is being steadily bettered, and this improvement will continue."

Granted that this statement is true, one can easily understand the variations in performance noted by the authors. Their conclusion that the use of very large turbines has contributed to a very limited extent to the reduction of fuel consumption is apparently justified by the data available. However, an analysis of these large units will indicate that their capacity over most efficient load is high, which with other factors accounts for their moderate performance. There is nothing to prevent the design of units of large size for higher efficiency than at present except the matter of price.

The authors have stated turbine performance in terms of Rankine cycle efficiency ratio. The derivation of this figure for bleeder and reheating turbines presents many difficulties. It would seem best to use Keenan's "availability" as a reference base, as he outlined in his paper at the last annual meeting.[15]

This paper repays careful analysis and study. It brings American practice up-to-date and indicates trends. The profession is indebted to the authors for their care in analyzing and assembling the material at hand.

T. E. PURCELL.[16] The keynote in American steam-plant practice has been reliability and economy, but sometimes it is amazing to realize what is happening to the simplicity of design which is so important, particularly during those emergencies which arise during operation. The modern tendency seems to provide such a multiplicity of controls and regulating devices and complex arrangements of piping and equipment as to tax the utmost skill of the operator.

The trend toward higher pressures has brought into existence several schools of thought as to the most suitable operating steam pressure. We should look with regret upon such discord, for it reflects the failure of the designers to agree when evaluating the factors that determine the design pressure. Some degree of standardization in the relative arrangement of equipment in the cycle, in the methods of arriving at the various design factors, and in the methods of computing the heat cycle should tend to unite the schools and place the problem on a sound economic basis. The standardization of the 400, 600, or 1200 lb pressure of course cannot be expected, because fuel cost varies and has such a profound influence on the design. Yet it seems we could standardize on the method of determining the design element.

The selection of initial steam conditions is important also, because it in some measure determines the extent of trouble that will be experienced with moisture in the lower stages of the turbine. There is no doubt as to the seriousness of the moisture problem even at values appreciably less than the 12.5 per cent mentioned in the paper. While blade materials more resistant to moisture bombardment are being developed, it appears that the most satisfactory ultimate solution will involve the elimination of moisture as it is formed rather than the development of materials resisting its attack. The corrosion problem in idle machines is also an important one which may some day be solved by circulating heated gases through the turbines when out of service.

The combined trend toward the increase in the size of steam turbines and toward the use of higher steam temperatures has very materially increased the starting time of modern turbine-generators. The means suggested in the paper to facilitate quick starting, i.e., low temperatures and perhaps multi-cylinder design, do not appear to be the most suitable in practice. Quicker starting should be obtained preferably by developing designs that permit a rapid readjustment in temperature without thermal distortions, or by maintaining the turbine at approximately operating temperatures when out of service for short periods. We have, at the Colfax Power Station, two single-cylinder machines

[15] "A Steam Chart for Second-Law Analysis," by J. H. Keenan, presented at the Steam Table Session of the Annual Meeting, New York, N. Y., November 30 to December 4, 1931, of The American Society of Mechanical Engineers; reprinted in *Mechanical Engineering*, March, 1932, p. 195.

[16] Duquesne Light Company, Pittsburgh, Pa.

that are very sensitive to thermal distortion. Originally, the time required to start one of these machines varied between 2 to 3 hours. Since slow-turning devices have been provided, the starting time after short outages has been reduced to about 45 min.

The acceptance of the 5 to 6 per cent leaving loss for large single-cylinder machines when operating at high loads is causing erosion and vibration troubles in the condenser. While vibration troubles due to the impingement of the high-velocity exhaust steam can be corrected by changes in the design of the condenser, it is nevertheless a factor that needs to be considered simultaneously with the acceptance of high leaving losses.

The actual turbine-generator capacities, rather than the rated capacities, are used on our system when determining the machines to be operated to adequately protect the load. However, we have not as yet found it convenient or very desirable to depend upon the reserve capacity available in the turbine by shutting off steam to the high-pressure bleeder heaters. The additional generator capacity available by operating the generators at high power factors would not be considered very desirable for our system, as we find it necessary from the standpoint of system stability to limit the power factor on the generators to a maximum value of 90 per cent.

It appears that operating engineers have established even a better record than that revealed by the fact that 72 per cent of their turbine-generator maintenance during 1930 was done when operation of the machines was not required. The 72 per cent apparently includes a penalty for work done when the machines should have operated for best system economy. The major concern of the operators, however, is to schedule their maintenance work on the machines when they are not needed for capacity, important as the thermal performance of the system may be. A detailed analysis of conditions on our system disclosed that, except for short periods during forced outage, the maintenance work is so planned that it never reduces the system capacity below that needed to amply protect the load.

Interchangeability of turbine-generator parts, as for example turbine spindles, to reduce outages of the unit, is rather difficult to apply in practice, although on paper it may appear to have merit.

E. L. Robinson.[17] The authors have arrived at the conclusion that the use of large-capacity steam turbines results in a reduction of fuel consumption to only a very limited extent. The implication is clearly against the installation of large units. As a matter of fact, the Rankine cycle efficiency plays only a minor part in the reduction of heat rates, the steam conditions and the type of cycle employed being considerably more important. Furthermore, the trend toward large units has been greatly influenced by the reduced fixed charges associated with large units. This point has not been particularly emphasized by the authors.

Throughout the paper the importance of leaving loss in determining the economical size of a steam turbine is well brought out. Leaving loss is commonly computed from the total weight of steam flowing to the condenser, supposing condenser pressure to exist at the last wheel annulus. A more careful analysis shows leaving loss to be made up of a number of items, among which are the following:

1 Annulus velocity energy, less a credit for particles of condensed moisture which move with very low velocity.

2 Edge eddy losses resulting from excess velocity on account of bucket-edge thickness.

3 Pressure drop through the exhaust hood itself from the wheel to the exhaust flange of the turbine, less a credit for the corresponding reduced velocity loss at the wheel due to the smaller specific volume of the steam at that location.

4 Whirl loss due to the reaction component of velocity at other loads than that corresponding to minimum leaving loss.

5 Losses due to non-uniform velocity distribution.

Items 4 and 5 are particularly important at low-volume flows with a last-stage bucket designed for high-volume flows. Furthermore, the leaving loss may be expressed (*a*) as a per cent loss of potential heat drop in comparison with the adiabatic heat drop theoretically available or (*b*) by a smaller figure expressing the loss as a percentage of the energy at the switchboard, including the energy contributed in the high-pressure stages by extraction steam.

The authors have pointed out the danger of making comparisons without taking account of leaving losses. The writer would add that it is also misleading to make comparisons based on estimates of leaving loss computed in a variety of ways.

J. N. Landis.[18] On January 22 and May 4 of this year, respectively, the Nos. 7 and 8 units carried load for the first time. Each unit is a General Electric 160,000-kw single-generator tandem-compound machine operating at 400 lb per sq in., 730 F temperature.

Both machines are smooth-running units and have given no operating difficulties with the exception of minor oil leaks and sticky governor valves. These troubles were quickly eliminated.

A weighed-water acceptance performance test has been conducted on the No. 7 unit when operating on a bleeding cycle. The results were satisfactory, the unit bettering its performance guarantee by approximately one-half of 1 per cent throughout the load range. A check test on No. 8 turbine shows it to be very slightly better than No. 7. The heat-rate curve of the unit between 80,000 and 160,000 kw has a variation of less than 1.3 per cent of the minimum heat rate.

The two condensers that serve the units have 101,000 sq ft of condensing surface each and are the largest condensers in the world. No. 7 condenser is by Worthington and No. 8 is by Ingersoll-Rand. The two units are operating satisfactorily, and preliminary figures now available from acceptance tests show that both units perform essentially in accordance with guarantees. The No. 7 main condensate pumps have given a little trouble with thrust bearings, which has been eliminated. The No. 8 condenser gave trouble at the start with high salinity caused by the ferrules backing out of the tube sheets. It is now believed that the trouble was caused by tube vibration in local spots. The tubing in those locations has been given extra staying and the remedy seems to have been successful.

E. H. Tenney.[19] The summary of steam-turbine practice is obviously a sound interpretation of typical rather than exceptional turbine developments. Since there has been at least a temporary slowing up of station economy improvement and also because of the present necessity for limiting capital expenditures, it seems likely that many of the older plants will be rehabilitated to bring them up to standards of capacity and economy. This situation may lead to a program of station rebuilding such as was witnessed a decade and a half ago, and for that reason a few comments on the question of turbine-room rehabilitation may be of interest.

In Table 1 the authors refer to a 75,000-kw turbo-generator at the Cahokia Station which in 1929 replaced a 35,000-kw unit

[17] Turbine Engineering Department, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

[18] Mechanical Engineer, Brooklyn Edison Company, Brooklyn, N. Y. Assoc-Mem. A.S.M.E.

[19] Chief Engineer of Power Plants, Union Electric Light and Power Company, St. Louis, Mo. Mem. A.S.M.E.

installed in 1923. In this example of rehabilitation, advantage was taken of two developments in order to bring an older section of this plant up to date in economy and particularly in the utilization of space for greater capacity. The boilers at Cahokia were originally designed in 1923 for 12,000 Btu heat liberation per cu ft of furnace per hour. Subsequent developments in pulverized-fuel firing have made it possible to achieve heat liberations upward of 25,000 Btu. Simultaneous increase in the capacity of single-cylinder turbine units and increased effectiveness of condenser surface (as pointed out by the authors) made it possible to utilize the greater boiler steaming capacity by moving, to a standby plant, a 35,000-kw unit and installing in the same space and on the same lower foundations a 75,000-kw unit of 18 per cent better fuel economy, and at an expenditure of approximately one-third of what the additional capacity would cost in a new plant. Such correlated improvement of boiler-furnace performance and turbine capacity applies at many plants and presents a definite possibility for low-cost increase of system capacity.

JOHN H. SMITH.[20] Theoretically the process of condensation is one of unqualified simplicity. The thermal and physical laws relating thereto have long since been established and countenance no trace of the mysteries nor the attending deficiencies incident to the multiplicity of structural complications which are introduced periodically as contributions toward further development of the commercial condenser. It is but a logical deduction, therefore, to expect improved condenser performance to remain forever a function of the approach, offered by the practical condenser, toward the ideals of simplicity upon which the fundamental laws were founded.

The condensation of exhaust steam effected in a vessel sealed against atmospheric pressure is made possible by extracting the latent heat of evaporation therefrom, and during the change from the vapor state to that of a liquid, the process is accompanied by a great reduction of volume and a corresponding reduction of temperature and pressure. The greater the amount of heat extracted, the greater will be the amount condensed and the lower will be the final temperature and pressure. Obviously, all vapor will never be condensed until the pressure is reduced to zero and the temperature to absolute zero.

Thus it becomes obvious that non-condensable vapor removal from any region within a condensing chamber other than that of lowest temperature surface, and therefore lowest potential pressure, results not only in a greater ratio of water vapor per pound of air removed therefrom, but also increased air and vapor pressure, increased terminal difference, and lower vacuum at the turbine exhaust.

It is axiomatic that thermal head induces steam flow to the region of lowest temperature surface, and that pressure drop resulting therefrom limits steam flow up the point of equilibrium.

Since equilibrium will be established in the vapor circuit of any condenser, it follows that all resistance other than that resulting from direct contact between condensing surface and steam should be minimized in order to promote natural flow to the region of highest thermal head in far greater quantities than would otherwise be possible. And with non-condensable gas removal correctly provided for at the exit terminal of the streams of flow, so as not to impede their natural tendency, the condenser can be said to be operating at its natural pressure drop.

As to external air coolers, they simply extend complicated structure to another unit subject to additional air leakage, and combine with all other forms of resistance to increase turbulence, eddies, and pressure drop in the effort to control steam flow. Their principal application appears to lie in the field of deficient installations in which non-condensable vapors are incorrectly withdrawn from the condenser; that is, where condensable water vapor finds the air-pump suction without having passed completely through the tube bundle to the region of greatest condensing capacity or lowest temperature surface. A condenser design of the simplicity necessary to favor the related laws of physics, thus contributing to natural flow in establishing equilibrium, renders it quite impossible to fully utilize the thermal head, and therefore requires no such appendage as an external air cooler.

Obviously the parallel relation between the several duplicate elements insures against the sharp reduction of flow areas in the natural path of vapor flow characteristic of the converging-lane principle now in common use. A comparison of the two principles is indicated in Figs. 11 and 12.

It is apparent that the converging-lane principle, indicated in Fig. 11, is based on the erroneous fundamental assumption of equal condensation per square foot of surface, hence the area reduction between planes, and obviously therefore on the basis of vertical steam flow through the condenser. The sectioned portion between condensing elements illustrates reducing steam quantity and corresponding free-space reduction. The writer, among the early advocates of the converging-lane principle, and the inventor of many existing condensers employing it, has long since come to realize that vertical steam flow through a condenser is not desirable, nor is it possible of attainment except by the introduction of complications to structure which further increase the throttling effect resulting from such area reduction along the paths of natural flow. Thus would lines of flow approach the vertical a material distance into the condenser, as indicated by lines x_1, x_2, etc.

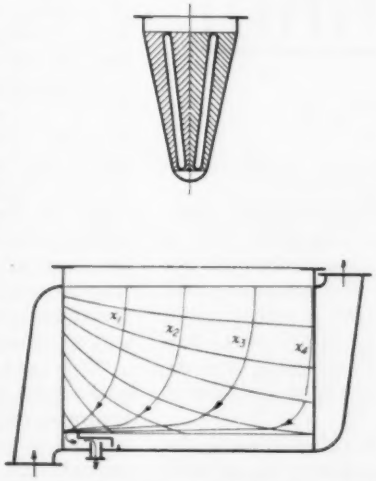


FIG. 11 CONVERGING-LANE PRINCIPLE

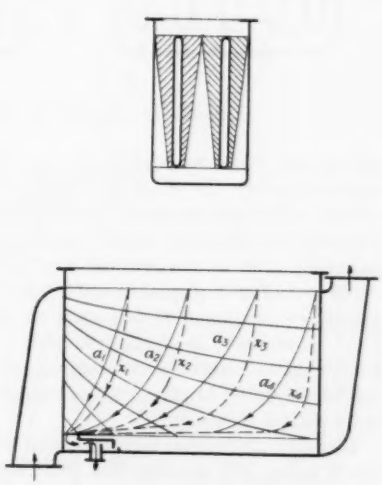


FIG. 12 PARALLEL-LANE PRINCIPLE

Full recognition of variations in heat head, and therefore variations in condensation per square foot, and also that natural pressure drop will balance the urge of steam flow to satisfy heat head far short of the potential capacity in regions of the exit terminal, has prompted the writer to provide not only the same areas characteristic of Fig. 11, but in addition thereto, the unshaded areas indicated in Fig. 12, thereby contributing to natural flow in increased amount. It is apparent that flow lines such as x_1, x_2, x_3, and x_4, characteristic of the converging-lane principle, would assume a more direct path of travel such as indicated by the full

[20] Alco Products, Inc., New York, N. Y.

lines a_1, a_2, a_3, and a_4 shown in Fig. 12, by virtue of greater flow area. This contributes to reduction of the time element and greater condensation per square foot of surface at a given pressure drop in the region of highest heat head.

Thus, it is believed, the most efficient utilization of available space provided for the installation of a condenser has only been approached, and as the economical demand for further surface increases per cubic foot of shell displacement continues, this fact becomes even more pronounced.

Neglecting adverse directional effect and high velocity of exhaust steam, it is obvious that the initially contacted surface being the most active, renders the more remote units of surface

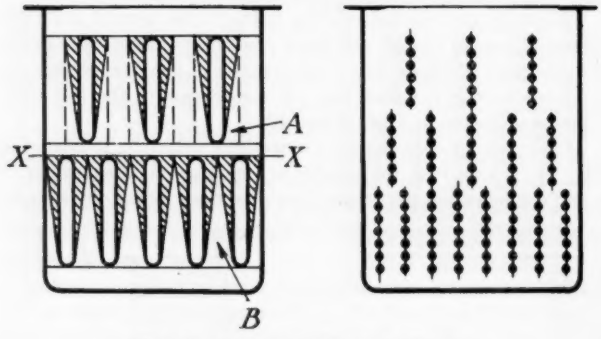


FIG. 13 INCREASED-SURFACE DESIGN

less active, thus demanding the greatest care, in distributing surface through the available volumetric space, in order to approach a balance between positional advantage and positional disadvantage of the units of surface comprising the total. This is especially true as the demand for further surface increase per cubic foot of shell displacement continues, thus demanding even greater consideration of flow areas in the path of natural vapor flow. Obviously, as steam is condensed in the top group, flow areas can be reduced accordingly in the lower group, and the logical procedure in effecting this reduction is by the addition of more surface elements.

Therefore, in further increasing surface in a given space by the use of tubes, it is evident that strict application of the parallel-lane principle throughout the tube bundle provides flow areas such as are required to promote natural flow in quantities greater than would obtain through use of the converging-lane principle. Such an arrangement is indicated in Fig. 13. Not only is a better balance between positional advantage and disadvantage of the tubes obtained, but of special importance is the assistance offered by this arrangement toward overcoming certain physical handicaps. The minimum amount of surface placed in the immediate region of the steam inlet, and therefore providing maximum free area, which can be interpreted as increasing per pound of steam in a given tube group as steam condenses, obviously provides for maximum dissipation of adverse directional effect of high-velocity steam in the sparsely populated upper group, thereby assisting in promotion of natural flow as steam passes to the lower tube groups of progressively increasing surface per cubic foot of space. Thus is the effect of tube erosion from this source minimized; and, too, since the minimum amount of condensate has the maximum distance to fall, and the maximum amount has a minimum distance to fall through the tube bundle, maintenance of free area and shell width at the bottom minimizes the detrimental effect of condensate and also reduces impedance to natural flow as compared to condensers employing the converging-lane principle, shell-width reduction, and surface disposition in the reverse order.

While it is generally recognized that, in order to improve condenser performance, steam loading on the cold end of a condenser must be increased and that on the less efficient hot end must be decreased, such as is provided for in a natural and simple manner by use of the parallel-lane principle, the use of tubes of substantial length introduces tube supports as a necessary evil. For example, it is reasonable to assume that if the tube arrangement is such as to impede natural flow of steam, the addition of tube supports will certainly add further media of prevention, thereby approaching full realization of undesirable vertical flow as the number of tube supports is increased.

Obviously, the directional effect of high-velocity steam becomes a more serious detriment to condenser performance as the condenser is sectionalized into well-defined compartments by the use of solid tube supports. Recognition of this fact has led to various methods of treating tube supports in an attempt to relieve overloaded hot sections of steam trapped therein. For example, in a previous invention the writer has cut tube supports into sections so as to provide staggered relation in various tube groups. Believed to be a step in the right direction, approximate vertical flow obtains, however, throughout the depth of any section of support plate, and in no way assists the problem of dissipating high steam velocity in the same limited sections of the upper or outer group, and contributes little toward promoting natural flow. Another method employed in an attempt to overcome these difficulties is to provide large so-called equalizing holes in the tube supports well within the tube bundle. Concentration of relief in a single large hole, or in a few smaller holes, in no way promotes balanced natural flow over the extent of the tube bundle, and certainly provides little if any relief in the section or sections between such holes and the main steam-inlet opening.

Only in the event of providing relief openings in balanced relation with every tube in the tube bundle, thus providing equal flow areas between all tube rows, is it possible to eliminate all traces of sectionalization, and thereby promote balanced natural flow throughout the tube bundle, to the maximum extent possible within the limits of practical space allowance.

While some of the existing methods of treating tube supports tend only to slightly relieve pressure in the hotter sections of the condenser, other designers have sought, through revival of various mechanical means, to control the flow of steam in predetermined amounts, and irrespective of its initial direction of flow, into the various compartments formed by tube supports. Not only have the thermal laws relating to vapor flow in a heat exchanger been wholly overlooked, evidence of which is found in the general disregard of flow areas in the natural path of travel, but by resorting to mechanical expedients in an attempt to control and redirect the flow of high-velocity steam, rather than occasion dissipation thereof within the tube bundle, it is believed that engineers have violated important physical laws as well, with the result that not only is condenser efficiency impaired, but the prime mover is prevented from driving the full benefit of the vacuum that is produced in the condenser tube bundle. For example, if it is assumed that the rate of condensation can be controlled in the various compartments by increasing resistance to flow through each succeeding hotter compartment, it can be accomplished only at the expense of turbulence and eddy losses incident to the necessity of any excess of steam reversing its initial direction of flow against the impact resulting from turbine exhaust velocities in the nature of 500 fps and above. Furthermore, the added steam load at the periphery of the cold end reflects in increased pressure drop and increased vapor content per pound of air, which of necessity must be removed by the air pump at a higher vacuum. This obvious mark of inefficiency, together with the violent turbulence and eddy losses throughout the vapor circuit resulting from attempts to control the flow of steam by introducing such artificial resistance media as throttle valves, ports, orifices, air baffles,

deflector plates, external air coolers, etc., presents a combination of factors which coincide with the effect of tube supports and an erroneous fundamental assumption relating to steam flow in defiance of the basic thermal and physical laws involved in the process of condensation.

As the name implies, air baffles are but another form of resistance to the free flow of vapors along natural lines toward regions of lowest temperature and pressure within the tube bundle. Moreover, their very location limits the effectiveness of the condensing surface they are intended to isolate, and their presence at all positively prevents any possibility of approaching balanced natural steam flow through the tube bundle at any transverse plane. Thus, the apparent wide variations in effectiveness of tubes on any horizontal plane, together with the increase of back pressure resulting from additional turbulence and eddies due to the many abrupt directional changes in the path of steam flow in gaining access to the isolated tubes and the common vapor header, render so-called air baffles not only useless, but a serious detriment to good condenser performance.

The ideal condenser consists of an infinite number of infinitely small surface elements operating in parallel, and while introduction of the second and third dimensions as applied to surface extension in a practical condenser introduces series operation as applied to a major portion of the surface elements, a logical procedure in approaching fundamentals would be to obtain as many duplicate elements transverse of the tube bundle operating in parallel as is physically possible. Mere symmetry about the vertical center line of a condenser is not sufficient to promote a balance of steam flow, but symmetry about the center lines of a multiplicity of duplicate elements affords an approach thereto. And this symmetry cannot be disturbed by introducing air baffles into the cross-section, in any attempt to approach the ideal.

It is highly desirable in any case, and especially so in the case of high-pressure steam stations, to expand the tubes into both tube sheets in order to eliminate leakage of cooling water into the condensing chamber. Moreover, where packing and ferrules are used, tube-end erosion and corrosion caused by eddies and gas liberation resulting from disturbance to stream-line flow, incident to the physical relationship between ferrules and tube ends, often result in excessive tube replacements. To minimize the effect of these maintenance difficulties, expanding tubes into both tube sheets has become a more common practice, and with it further complications to structure designed to compensate for the difference in expansion between ferrous metal shells and nonferrous metal tubes. Just as simplicity of structure is practiced as an essential procedure in giving full recognition to laws relating to steam flow, so does the writer believe the problem of compensating for expansion differentials to be equally simplified.

For example, tubes initially bowed for drainage purposes and rigidly secured at both ends are virtually self-supporting. Yet, while readily susceptible to further bowing, there is no assurance, under temperature changes, that the tubes will bow as a group in the direction prescribed by the initial bow unless so guided; hence, the principal value that can be attached to tube supports, which for this purpose can be free to move upward. Therefore, with virtually all metal not required to maintain a constant relation between tubes removed from tube supports, the weight of such tube guides is materially less than the existing forms of tube support of equal thickness and for a given number of tubes supported. And as such, this method of compensating for expansion differentials is a natural sequence to the fundamental requirements of structural simplicity.

Nor is the problem of approaching ideal conditions confined to the vapor circuit alone, for without a similar approach in the hydraulic circuit both objectives remain in a remote sphere. And in consideration of this fact, sufficient justification against mechanical manipulation of the vapor circuit is to be found, for the potential efficiency of any tube is a function of the heat-absorbing capacity of the water traversing it, which in turn is a function of water quantity; while the actual efficiency of the tube then becomes a function of the facility with which steam can contact the successively colder increments of its length.

The fundamentals of the problem of surface condenser design, therefore, appear to be clearly evident. First, the tubes of a condenser constitute the number of elements that are intended to operate in parallel, and if similarly treated, all would operate equally and in parallel. Obviously, the series function in terms of progressive condensation would be equal for all tubes, and the physical ideals of the vapor circuit would be fully provided. Second, equal distribution or circulation of water through all tubes would complete perfect balance of flow in both circuits. Obviously, neither condition can be met in practice, but a practical approach thereto is to be found in a condenser that provides physical balance, at least, between all tubes on any horizontal line across the tube bundle.

I. E. Moultrop.[21] There are now installed in the United States 2,800,000 kw of turbine-generator capacity operating on

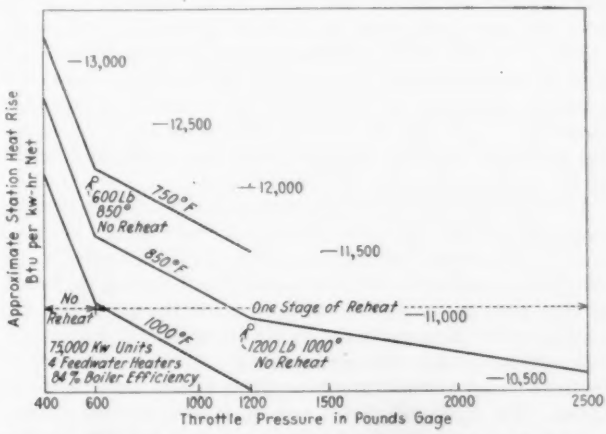


Fig. 14 Comparison of Approximate Station Heat Rates at Constant Temperature and Varying Steam Pressure

the regenerative-reheat cycle, and for approximately 80 per cent of this capacity flue-gas reheaters are used.

The May, 1932, Prime Movers Committee Report on Higher Steam Pressures and Temperatures gives some interesting data on 22 1200-lb boilers ranging in size from 150,000 to 500,000 lb per hour. This report shows that the availability of these boilers is substantially the same as for boilers operating at lower pressures, and the availability of the 1200-lb boilers installed in Milwaukee is comparable with the best performance listed in the 1931 Report of the Power Division of the A.S.M.E.

It seems to be the consensus of opinion of those who have installed and operated reheat boilers that they are just as reliable as those without reheaters.

Fig. 14 shows the approximate station heat rates at constant temperature and varying steam pressure. As one can readily see, the 600-lb, 850-deg non-reheat station will show approximately 5 per cent better station heat rate than the 400-lb, 850-deg station. But it is also a fact that the 1200-lb, 850-deg reheat station shows approximately 9 per cent better station heat rate than the 600-lb, 850-deg. non-reheat station.

The operating companies that have built 1200-lb reheat stations report that there is practically no difference in unit cost of

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construction as compared with 400-lb non-reheat stations. In view of these facts, there seems to be no justification for the statement that the 650-lb, 825-deg station is the most attractive one.

Another important consideration in the selection of the most economical pressure and temperature is the fact that the design and manufacture of high-pressure generating equipment is in its infancy. As time goes on, designs will be improved and manufacturing methods will be changed so that we will be able to install the more economical equipment for considerably less per unit of capacity than we can today. It is necessary, however, that we go through a period of development. If we can promote this development without actual financial loss, by the use of equipment that is economically justified at its present price, we aid in the progress of the art and will reap even greater profits in the future. If we refuse to install the equipment which our engineering studies indicate is the most economical, we are seriously hindering the progress of our industry.

The paper points out that larger units reduce the unit cost of construction, but that this is partially offset by the investment in spare capacity. We should also remember that large units reduce operating labor and maintenance, and up to a rather large size improve the station heat rate. It is wondered if, in the long run, large units really do increase the investment in spare capacity. If the largest unit has a capacity of from 20 to 25 per cent of the system peak, it is believed that it actually results in a lower cost including spare capacity than if the individual units have a capacity of only 10 per cent of the system peak. The average additional capacity required due to the larger units is not great and the unit cost is considerably less.

It is very encouraging that the multi-cylinder units are showing even greater availability than the single-cylinder units. We need units of high availability which can be put on the line quickly. If the system load curves are going to continue to grow in the future as they have in the past, with sharper and sharper peaks, the need for units which can be put on the line quickly and which can pick up load quickly will become acute. It will be very wasteful to be forced to load turbines as slowly as it is necessary to do today. The multiple-cylinder turbine is probably the answer.

In its proper sphere, standardization is very desirable, but the writer questions the advisability of attempting to go very far in the standardization of station design for some time to come. The design of station equipment is going through such a radical period of development that too much standardization is likely to slow up development. If a large number of units of the same size and design are installed for a rapidly growing system, the unit cost of construction will be considerably higher and the slight advantage of a number of duplicate units will be more than offset.

If we are going to be able to take advantage of the new ideas in steam-generator design, such as the series boiler, it is necessary that we find some way to reduce condenser leakage. The condenser with tubes expanded into the tube sheets seems to offer a practical solution.

If chlorination of the circulating water will permit continuous condenser performance equal to new tube conditions, it will not only result in better station economy, but it will permit a further reduction in condenser size and a worth-while reduction in the unit cost of construction and maintenance. While the improvement shown for Powerton Station could probably never be realized by tidewater stations, nevertheless the possibilities are very much worth while.

Electrically driven auxiliaries have been adopted quite generally, but unless more satisfactory variable-speed motors and controls are produced, we can expect to see a swing back to turbine-driven auxiliaries for some of the important auxiliaries. For auxiliaries requiring large-capacity variable-speed prime movers, the turbine drive has many advantages that may even justify a slightly higher station heat rate.

M. K. Drewry.[22] The authors devote only a single statement to the new subject of turning gears. This apparatus has proved to be of considerable importance to the successful operation of two recently installed 60,000-kw units. Daily shutdown and starting of these two machines, of different manufacture, is impossible without turning gears. The first machine was

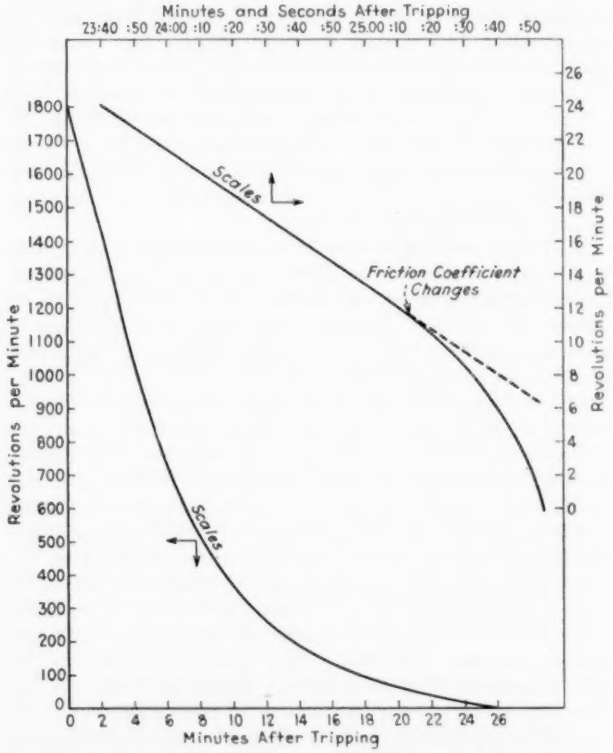


Fig. 15 Deceleration Curves of a 30,000-Kw Impulse Turbine (Zero field current. Normal operating oil temperatures.)

started only once after a short idle period without previous use of the turning gear. It was not brought into service until noon, though started early enough for forenoon loading. The second machine was installed initially without a turning gear. When started after several days' outage, no trouble was experienced, but it positively could not be used for overnight outage service. On several occasions it could not be started at all during the day that its use was desired. Turning gears on both machines have made their operation entirely satisfactory.

The spindles of both turbines are of large diameter, being of the reaction type. When at rest, considerable difference in metal temperature must occur between the top and bottom of the spindle because of its large diameter. Very slight temperature difference causes sufficient deformation to result in severe vibration when starting, and the deformation cannot be removed in a few hours of turning.

An eccentricity of 0.015 in. near the shaft seals would occur without use of the turning gears after shutdown. Two days' cooling would eliminate this bowing, as would 6 hours of turning at 25 rpm. The machines are always kept turning for 8 hours after shutdown. Approximately 6-kw input is required for 25 rpm.

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Fig. 15 gives data showing the importance of reasonably high turning speeds. When a typical turbine slows below approximately 10 rpm, its speed drops very rapidly, showing that lubrication changes nature. The shaft speed must be high enough to maintain oil films, or oil under high pressure must be supplied below the journals.

Authors' Closure

The discussion presented has served in the main to supplement the paper. Our purpose in presenting this paper was to give the truest possible picture of steam-turbine-plant practice in the United States. It has seemed important to present, not only the facts, but also our interpretation of the facts. It is perhaps not too unexpected that several of the engineers who have discussed the paper should question our interpretation of the facts as presented. Operating experience and the trend of design during the years immediately ahead will perhaps serve as the most effective answer to the questions which have been raised.

[Note: Discussion of this paper, jointly with that of F. W. Gardner, by M. H. Adams, Gerald Stoney, H. M. Martin, and W. F. Ryan will be found following the discussion of that paper.]

FSP-55-3b

British Practice in Steam-Turbine Design

By F. W. GARDNER,[1] NEWCASTLE-ON-TYNE, ENGLAND

This paper reviews considerations affecting steam-turbine development in England. Turbines of the size and type required for the larger central stations are covered, although description of some of the smaller plants has been included. In discussing steam conditions, attention has been confined to general tendencies. Developments in turbine practice have been influenced by the plan to link up all the power stations by an interconnecting network of transmission lines. To cope with the increased demand, new large power stations will be constructed, which will have the advantage of high load factors and favorable conditions for base-load running. Illustrations are given of types of turbines and equipment.



THE aim of this paper is to present a brief review of considerations affecting steam-turbine development in Great Britain at the present time. The subject embraces such a wide field that it is only possible here to draw attention to a few of the salient features and to indicate very briefly their influence on the progress of design. Reference has been made to a few of the problems that arise, with the knowledge that these and many similar problems are familiar to American engineers, for it is felt that they will gain rather than lose interest on that account. For the most part the paper has reference to turbines of the size and type required for the larger central stations, but a brief description of some small plants has been included to illustrate the developments at the other end of the scale. A short note dealing with condensing plants has also been included.

The many interesting developments that have taken place recently on the marine side do not come within the scope of the paper, which has been confined to features relating to land practice.

In preparing the paper, the author has drawn chiefly upon the work of the firm with which he is connected, for it is naturally of this work that he has the most detailed knowledge and experience. At the same time he has endeavored throughout to treat the subject from a general standpoint, and he gratefully acknowledges the assistance received from friends in other companies, which has helped to make the paper representative of British practice.

[1] Chief Engineer, C. A. Parsons & Co., Ltd. (Turbine Department). Mr. Gardner was born near Liverpool, England, and was educated at the King's School, Chester; was graduated in Mechanical Sciences at Cambridge University. He joined the staff of C. A. Parsons & Co. at Newcastle-on-Tyne in 1912, leaving to join the forces at the outbreak of war in 1914. Held a commission as Engineer Officer in the Naval Airship Service, becoming a Major in the Royal Air Force on the transfer of the airship personnel to the Air Force in 1918. At the end of the war he returned to the Parsons Company, becoming head of the turbine-design staff a year or two later, and was recently appointed to his present position. Mr. Gardner visited the United States and Canada in connection with the firm's business in 1930.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

In discussing such matters as steam conditions, attention has been confined to general tendencies, and no reference has been made to experimental, specialized, or exceptional plants, though of course a number of installations in these categories exist.

INFLUENCE OF THE "GRID"

Developments in turbine practice in Great Britain during recent years have been influenced to a considerable extent by the introduction of the National Electricity Scheme, which provides for the eventual linking up of all the power stations in the country by an interconnecting network of transmission lines known as the "grid." It also provides for the erection and equipment of a number of new large central generating stations, for the extension of the larger and more efficient of the existing stations, and for the scrapping of a great number of small and inefficient stations.

At the period in 1920, when the Electricity Commissioners were appointed and given power to investigate the whole subject of electricity supply in Great Britain, the requirements both domestic and commercial were being dealt with by some 540 power stations, varying from quite small outputs up to 100,000 kw capacity. The largest individual units at that time were about 25,000 kw. These stations were variously owned by municipalities, public supply companies, or private companies, all working under statutory orders, but operating independently of each other. As a consequence of this there were wide variations in the systems of supply, each authority or company having fixed the conditions to suit the requirements of its own undertaking, with the result that there existed d-c systems at various voltages, a-c of one, two, and three phases, and frequencies ranging from 25 to 100 cycles. Each undertaking had to make its own provision for the standby and emergency plant to insure continuity of supply, though in certain areas, where conditions permitted, voluntary interconnections were made to enable some degree of interchange of supply to be arranged.

The Electricity Commissioners were given powers to control all additions to generating plant in existing power stations, and new power stations could be built only after their approval had been obtained. These powers applied only to public-service undertakings and did not cover private generating stations in factories or mills. The commissioners were also given powers to compel undertakers to install interconnecting lines to facilitate the interchange of supply. Already the number of stations in operation has been reduced to some 300, and provision is made for an eventual reduction to about half that number.

The investigations of the Electricity Commissioners and the subsequent report of the special committee under the chairmanship of Lord Weir in 1925, led to the passing of the Electricity Act of 1926, under which the Central Electricity Board was appointed to undertake the construction and operation of the grid power lines and the unification of frequency throughout the country at 50 cycles. A voltage of 132 kv has been adopted for the main transmission lines of the grid.

The central board has power to raise capital, out of which they meet the expenses incurred by the undertakers on change of frequency in their areas, where such change has been carried out under an instruction from the board. The board is paid by the Electricity Commissioners the sums required to meet interest and sinking-fund charges on the money raised for frequency-change purposes, and the Electricity Commissioners in turn make a

charge on all authorized undertakers on the basis of revenue received from the sale of electricity, other than that sold in bulk to authorized undertakers.

Developed under the long-sighted and organized policy of the commissioners, the National Electricity Scheme will undoubtedly encourage the use of electricity for all purposes, and as a result a continued increase in consumption is to be expected. To cope with the increased demand, as it arises, new large power stations will be constructed, which will have the advantage of high load factors and the conditions most favorable for base-load running. Although a considerable amount of new generating plant will thus be required to carry the load that will be available when the grid gets into full operation, the ultimate effects of the scheme will be, first, to encourage large units and eliminate small ones, and, second, to diminish the total kilowatts of new plant constructed, owing to the reduction in the amount of standby plant required. The manufacturing capacity of the turbine works in the country will therefore be much greater than will be required to meet the demand for new plant, as it is clear that much less labor, machine time, and works capacity will be employed in the building of one 50,000-kw unit than would be required for the production of, say, five 10,000-kw machines.

The demand for small units for factories and mills that generate their own power is very meager at the present time in consequence of the general trade depression by which so many industries are affected. The grid scheme is likely to reduce considerably the demand for such units in the future, as under it factories will be able to obtain very favorable rates for power supply, and electrification without capital expenditure on generating plant will become the most attractive proposition. Industrial plants that require large quantities of steam for process work will, however, be able to generate more cheaply with their own plant, and will in some cases be able to feed current into the grid.

Although the amount of water power available in Great Britain provides only a small percentage of the total requirements, hydroelectric installations are being constructed, which will take advantage of the grid lines for distribution into the general system.

Size and Type of Units

To equip the new central stations, large generating units are required—that is to say, large in comparison with the sizes previously found economic for the conditions existing in England. At the present time a number of units are in operation or under construction ranging in capacity from 50,000 to 75,000 kw. Where extensions to existing stations are carried out and where existing plant of non-standard frequency has to be replaced, units of from 15,000 to 40,000 kw capacity are usually required.

Practice in England is definitely in favor of single-line units, and all the plants now being installed are of this type. The size of unit in relation to the normal output of the station is so chosen that there does not arise the necessity for operating one section of a machine independently of another, a provision that usually has to be made where large cross-compound units are installed.

Multi-cylinder machines are general for both impulse and reaction turbines. The principle that a large number of stages with moderate steam speeds are essential for high efficiency has long been known, but has in the past been rather ignored in impulse-turbine design, where the application of partial admission encouraged the use of large wheels at the high-pressure end. Latterly, however, the manufacturers of impulse turbines have concentrated on many-stage machines. Fine diaphragm and blade clearances are as essential for efficiency as low steam velocities. This necessitates short, stiff rotors and imposes a limit on the number of wheels per cylinder. Consequently, to utilize efficiently the heat drop available with the higher steam conditions now in use, either two- or three-cylinder designs are nearly always employed.

The demand for high thermal efficiency has necessitated also the provision of large blade areas at the exhaust end to keep the leaving losses small. This has doubtless influenced the choice of rotational speeds, and up to the present single-line units of 40,000 kw and upward have all been designed for 1500 rpm. Several 30,000-kw plants at 3000 rpm with double-flow low-pressure elements are in operation, and as 3000-rpm generators can be built for much larger outputs, an extension of the size at this speed is to be expected because the manufacturing cost of the turbines is much reduced. Larger units have, in fact, been introduced on the Continent. A 3000-rpm turbine with three exhausts and the same tip speed as is used in current practice, say, 1050 fps, is capable of an output of 50,000 kw.

Present Trend in Steam Conditions

Steam Pressure. For the larger central-station plants now under construction a steam pressure of 600 to 650 lb per sq in. is being generally adopted. This applies whether reheating is employed or not, and indicates that there exists a remarkable uniformity of opinion among those responsible for the choice of steam conditions on the question of the most suitable pressure. Whereas in the past pressures have crept up gradually and by small stages, interest today is concentrated chiefly on two pressure levels at about 600 and 1200 lb per sq in. This is because the theoretical cycle efficiency is increasing fairly rapidly with the pressure up to 600 lb, but above this the efficiency curve becomes relatively flat, and the view is widely held that intermediate pressures possess to a considerable degree the limitations of the higher level, while the potential saving is too small to justify their adoption on the score of efficiency. They are found only in cases where special conditions have had to be taken into consideration, such as the power-output requirements in relation to the quantity of process steam for chemical or industrial purposes.

Among English engineers there appears to be a definite preference for the 600-lb level. Much importance is attached to securing the highest possible internal efficiency of each individual part of the plant, and this is regarded as the best method of obtaining a good overall performance, as it involves less capital expenditure than the adoption of the higher pressure. With the total temperature and other conditions kept the same, the maximum possible saving in fuel at 1200 lb, as compared to 600 lb, amounts to some 5 per cent, but this difference is easily lost unless the efficiency of the boilers, turbines, and auxiliary equipment is fully maintained.

Temperature. The generally accepted steam temperature is 800 to 850 F at the turbine stop valve. In the matter of temperature there is not really very much choice at the present time, since a fairly definite limit is set by the properties and behavior of the materials available. The strength of the usual materials of construction falls off as the temperature rises. In the past this has generally been met by designing the parts subjected to the full temperature with proportionately lower stresses so that the factors of safety have been maintained. Between 850 and 900 F, however, change in the physical properties becomes so rapid that it cannot be allowed for by design alone. Much attention is therefore being paid to the production of materials more suitable for higher temperatures and to the investigation of the properties of such materials. Designers would not, however, be justified in adopting a higher temperature for an important base-load station until a fuller knowledge is available as to the effect on these materials of prolonged exposure to high temperatures. Further, special materials are usually expensive, and even when their reliability is established the economic problem of their use has to be considered.

In connection with the problem of service at high temperatures,

interest is concentrated primarily on the creep characteristics of the material. The short-time yield, which is such an important factor in assessing the suitability of material at lower temperatures, ceases to possess much physical significance, because the amount of strain is influenced so much by the time interval during which the stress is applied. It is now generally realized that the creep limit cannot be defined as the stress below which the creep strain is zero, but that any stress will produce a slow plastic deformation. The designer has therefore to decide what rate of deformation is permissible for any particular application of his material.

The design of the high-temperature part of the boiler presents the most serious problems, but these do not come within the scope of this paper. As regards the turbines, it may be broadly stated that for cylinder castings and shaft forgings the stress must not exceed a value which will produce a creep of 10^{-6} in. per in. per hr, a rate which would produce a strain of 0.05 in. in a 5-ft length in 10 years. For turbine blades that in the high-temperature part are usually of small dimensions and are subjected to their maximum stress only on a small portion of their length, a much higher creep rate is permissible, say, 10^{-5} in. per in. per hr, and a similar figure may be taken as applicable to such items as valves, which in any case require readjustment from time to time. Valve stresses are, however, usually quite indeterminate, and design can be based only on experience.

Data on creep rates as low as 10^{-6} can be obtained only by very careful research extending over a considerable period of time. Though a great deal of information relating to creep has been accumulated, much of it applies to rates higher than 10^{-5}, and caution is necessary in drawing inferences from this, because materials showing good properties at the higher rates may not maintain their relative superiority at the lower rates. The good results obtained with steels containing from 0.5 to 1 per cent of molybdenum are leading to the employment of these alloys for an increasing number of uses, and they are now generally regarded as the most suitable materials for both castings and forgings that have to operate at temperatures above 800 F.

The development of these molybdenum alloys has had an important influence on the technique of steel production, and they have found many applications where high temperatures are not necessarily involved. It seems likely that in the future their use in turbine work will not be limited to cases involving temperatures above 800 F.

An additional and somewhat different problem arises in the case of bolts for the turbine and steam-pipe joints. It is due to the phenomenon of stress-temperature embrittlement. If, for instance, a nickel-steel bolt made from material with an Izod value of 50 ft-lb be subjected to stress at a temperature of 800 F for a few months and then tested, its Izod value will be found to have fallen to 3 or 4 ft-lb. This embrittlement may not perhaps be as serious as it sounds, especially as some uncertainty still seems to exist as to whether a corresponding change occurs in its ability to resist shock stresses at working temperature. The possibility of bolts in service being subjected at one time or another to shock stresses when either hot or cold cannot, however, be ignored. Such conditions might, for example, arise due to water hammer. In these circumstances embrittled bolts would certainly be a source of weakness.

The embrittlement effect seems to be chiefly associated with nickel alloys—it is characteristic of the nickel and nickel-chrome steels that in the past have been so widely used for bolts; but there appears to be some evidence that it can be prevented by the introduction of molybdenum. Opinion is divided as to whether molybdenum in a nickel alloy is effective in this respect. Some metallurgists have been led by their research to the conclusion that the embrittlement effect is inseparably connected with the use of nickel, and that alloys that do not contain nickel are much superior to those that do. The materials most favored at the present time for bolts for high-temperature service contain from $^1/_2$ to 1 per cent of molybdenum, about 1 per cent of chromium, nickel (if employed at all) about 2 per cent, and carbon varying from 0.3 to 0.5 per cent according to the tensile strength specified.

Reheating. A very substantial gain in the steam-cycle efficiency becomes available if intermediate resuperheating, generally called reheating, is adopted. So much attention has been paid to this subject in technical papers and so many theoretical data have been published that the potential advantages of reheating can hardly fail to be generally appreciated. Nevertheless, its merits have not up to the present received the recognition they would seem to deserve; at all events, they have not received the practical recognition of being generally adopted, even for large plants.

The new station now being erected at Dunston for the Newcastle-upon-Tyne Electric Supply Company is a notable exception in this respect and will be referred to more fully later.

While there would seem to be no doubt that reheating will become an essential part of the thermodynamic cycle in the future and is a logical corollary to the use of higher pressures, the progress of its development has been slow. The pioneer installations in this country have had to contend with many difficulties and adverse circumstances, some of them not directly connected with the reheating, and others easily corrected in the light of the experience gained. Consequently, operating results have not always appeared very convincing and have sometimes been misinterpreted by the skeptic.

The earliest reheating installations were put in by the Newcastle-upon-Tyne Electric Supply Company about 1918. In that year a reheater supplied with live steam at a pressure of 275 lb per sq in. was put into service on an 11,000-kw tandem turbine at Carville Power Station. In the following year at Blaydon Burn, a small waste-heat station operated by the same company, a separately fired reheater employing coke-oven gas was set to work and reheated the steam between the cylinders of a two-cylinder geared turbine to 400 F. Owing to the low temperature of the reheat the actual gain in economy in both these installations was small, but the experience gained with them paved the way to the adoption of flue-gas reheaters integral with the boiler in the North Tees "A" Station in 1920, and later in connection with a 40,000-kw unit at the Barking Station of the County of London Electric Supply Company. The latter was the first installation in which the reheating was carried right up to the initial temperature. The test results on these plants have demonstrated quite convincingly that the full theoretical reductions in turbine heat consumption are realized.

One of the objections to reheating frequently raised is the complication of the pipework to and from the reheater. Experience with existing installations has shown that the accommodation of the pipes and the necessary provision for their expansion is not really difficult. Operating results have proved that the problems peculiar to reheat—such, for example, as the regulation of the temperature and the provision of adequate emergency control—can be dealt with quite satisfactorily. Little or no difficulty has been experienced in the general operation of the plants.

It is of course essential that reheat units should operate as base-load machines and that they should run for long periods with a reasonably good load factor. Frequent starting and stopping involves the loss of much of the advantage of the reheat, owing to the time taken on each occasion to reach steady conditions. This consideration has perhaps had much to do with the tardy progress of reheat, for the ideal of base-load machines is apt to be elusive, and the adoption of reheating can be justified

on economic grounds only if the potential saving is realized.

So far as the turbine itself is concerned, continuous running presents no difficulty, but the boiler must be laid off for repairs to refractories or to grates. As with existing systems the turbine is tied in with its own reheater boiler, this involves either shutting down the turbine or running without reheating. In consequence, much importance attaches to the frequency of these boiler stoppages. For similar reasons it is essential that the condensing plant should operate for as long as possible without requiring tube cleaning, and the periods for boiler overhauls and condenser cleaning must correspond so that they can be carried out together. This condition may not be easy to attain, for it is equally important to maintain a reasonably good vacuum—a drop of 1 in. neutralizes half the gain due to the reheat. Much depends of course on the circulating-water conditions at the individual station. In cases where serious difficulty has been encountered from fouling of tubes due to the growth of algae, chlorination of the water has given very satisfactory results.

Reheating by means of live steam, which at one period became very much favored in the United States, has not received much support in England. The system is subject to the limitation that the gain, in comparison with the non-reheat system operating with the same total steam temperature, is inevitably small and insufficient to make it attractive. The view usually has been taken that, wherever there is a good case for reheating, boiler reheating up to the full temperature affords the sounder proposition.

So far as the author is aware, reheating by steam has been applied only to one recent machine. This is at the Valley Road Power Station of the Bradford Corporation. The turbine, a tandem unit of 30,000 kw, built by the English Electric Company, operates on a stop-valve pressure of 200 lb per sq in., but there is a boiler at a pressure of 1100 lb per sq in. supplying a high-pressure turbine which exhausts into the 200-lb steam range, and steam from the 1100-lb boiler is utilized for reheating.

Feed Heating. The maximum theoretical advantage that can be derived from feed heating by bled steam increases with the number of stages adopted; so also does the most economical value of the final feed temperature. For the stages beyond the fourth, however, the percentage is small, and practical conditions such as initial cost and additional complication usually limit the choice to this number. As in most installations an evaporator taking bled steam is incorporated, it is usual to supply the evaporator with steam from the same bleeding point on the turbine as one of the higher feed-heat stages, and to condense the evaporate in the next lower stage. The appropriate final temperature of the feed under these conditions is that corresponding to about 70 per cent of the saturation temperature at boiler pressure. As the feed temperature necessarily varies with the output, this value is normally designed for at economical load, so that the best conditions of feed heating may be maintained at maximum load and also for a reasonable range below economical load.

Feed heaters with an outlet temperature up to 220 F can generally be installed on the suction side of the feed pumps, which are arranged to suit this suction temperature. Above this temperature high-pressure heaters have to be employed on the discharge side of the feed pump and provided with an automatically controlled bypass, so that in the event of a tube failure the turbine is protected from flooding without the supply of feed to the boiler being interrupted.

The practice of draining each heater into the next lower stage and finally through a drain cooler to the condenser has the merit of avoiding the use of a small motor pump to return the condensed steam to the feed system at each stage, and is for this reason frequently preferred by operating engineers. The use of drain pumps does, however, give a slightly better economy. The difference may amount to more than half of 1 per cent, which is an important consideration on a large unit, but the greater part of this gain can be secured by utilizing a pump for the last stage only and draining the others in cascade.

A special deaeration plant is not generally found necessary, as it is possible, if the system is properly arranged, to keep the oxygen content sufficiently low by deaerating the make-up in the condenser. The high condensate temperature obtained with modern condensers helps to make them efficient deaerators.

Some Details of Design

Rotor Forgings. Of all the factors that influence the progress of turbine design, the most important is the production of sound steel. So much depends on the reliability of the forgings available that the turbine manufacturer finds himself compelled to take an active and leading part in the development of steel-works practice. The close cooperation of both interests has enabled rapid strides to be made in steel production during the last few years, and a remarkably high standard has been achieved in the forgings available today for large turbine and alternator rotors.

A rigorous control is exercised on the passage of the steel right through from the refining of the metal and the pouring to the final treatment of the rough-machined forging. All rotor and disk forgings used by the author's firm are manufactured in strict accordance with its own specification, which regulates the entire process so that the internal strains due to inequalities of temperature are kept down to an absolute minimum throughout. A new method of casting ingots, due to the late Sir Charles Parsons, was developed a year or two before his death. In this method the ingot is cast cylindrical and with a diameter about twice its height. The mold is constructed with a massive chill at the bottom so that the cooling proceeds from the bottom upward, the sides of the mold being made of firebrick preheated to 1000 C. Heat is applied to the top of the ingot to delay setting of the metal. This method, which is described more fully in a paper read before the Iron and Steel Institute (*Journal of Iron and Steel Institute*, 1929, Vol. 119, page 255), has proved extremely successful. The segregates instead of being forced in toward the center are driven up continuously as the ingot cools from the bottom, and are accumulated in the top layer, so that not only is the resulting steel much freer from harmful segregates and axial unsoundness, but also the discard is very much reduced.

All rotor forgings are of course bored, and where possible chambered, and are examined by means of a borescope. In addition, all exposed faces and journals are sulphur-printed as a check on segregates and ingot corner weakness. When all tests have been passed, the rotor is machined nearly to final dimensions, leaving only a small allowance for finishing, and then run up in a bombproof shelter to a speed of $1^1/_2$ times the working speed, corresponding to double stress. This test affords a final assurance that the material is perfectly sound and up to specification in every respect, although the precautions taken in its examination and physical tests make the chances of a failure on overspeed remote. Turbine disks are subjected individually to a similar overspeed test before mounting on the shaft, their dimensions being carefully gaged before and after the test.

Cylinder Construction. The design of high-pressure cylinders resolves itself mainly into a question of reducing to a minimum the strains and distortions which result from inequalities of temperature. With this end in view, a design has been adopted which aims at providing a rigid and symmetrical tubular core which will attain a sensibly uniform temperature across transverse sections, and will under all conditions dominate the shape of the structure. The steam inlets, feed-heating ports, and so forth are arranged as flexible belts which, where their natural expansion differs from that of the main core, can accommodate themselves without introducing appreciable constraints. Three views of

such a cylinder are shown in Fig. 1. It will be seen that the tubular core which carries the blading is made of very heavy-section metal. It is provided with a series of symmetrically disposed ports at each belt for the admission of steam. The walls of the belts themselves are made no thicker than is necessary to sustain the pressure and are shaped so as to give flexibility. The flanges are very deep and the bolts are kept close in, also the outer portion of the flange is parted at each bolt hole; thus the same effective thickness of metal is maintained all around, and stresses due to pull of the cooler part of the flange are avoided. The joint is strengthened by the provision of long studs through bosses in the cover close up to the inner core, these bosses being fitted with mild-steel liners in cases where the complexity of the casting makes this desirable.

When the steel portion of the cylinder is made in two sections with a transverse joint between them, this joint is held by heavy countersunk rivets and is welded all around on the inside. Rivets have the advantage that they can be pitched closer than bolts,

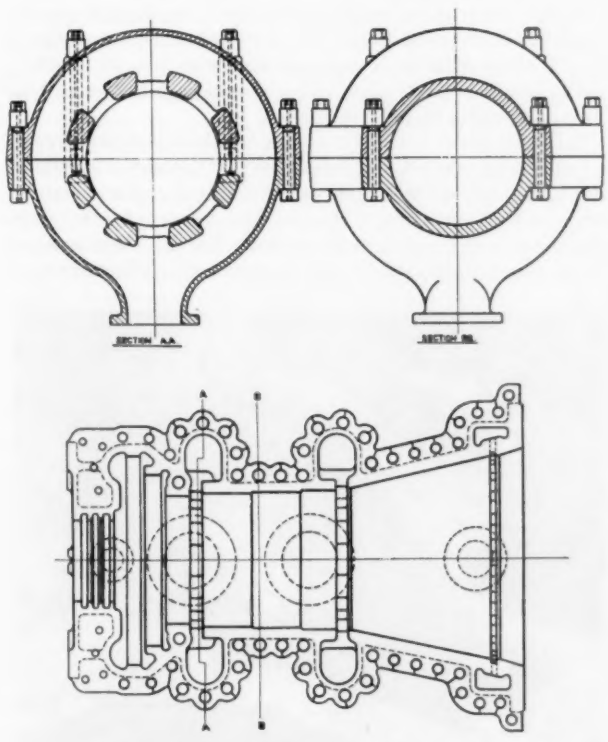


FIG. 1 ARRANGEMENT OF STEAM-INLET BELTS

and also as there are no projecting heads the bolts in the horizontal joint can be brought up closer to the vertical flange.

Steel cylinder castings are fully annealed at the foundry. After removing the headers, they are dressed and cleaned and then charged into an annealing furnace, brought up to 920 C, and afterward cooled down gradually in the furnace to atmospheric temperature. After machining and boring to $^1/_{16}$ in. off finished size, they are mounted on end on a bogey and put into a totally enclosed furnace, where they are brought up to 500 C, and again cooled down very slowly. This second treatment removes any internal stresses that may remain after the machining and that otherwise might be slowly released in service.

The practice of heating the cylinder-joint bolts to tighten them up has the advantage that clearance for a heavy spanner is no longer required—the bolts can be pitched closer together and they also can be tightened to an equal and determinate stress. The arrangement of electric heating shown in Fig. 2 has proved to be simple, speedy, and effective. It is important that the bolts should be heated up as rapidly as possible; otherwise too much heat will be conducted to the flange. For this reason a carbon rod that will absorb about 4 kw is much superior as the heating medium to any form of wire-wound element. Tightening of, say, a 4-in. bolt will take only from 15 to 20 min.

The electric heating is used for both closing in and for opening out the turbine, and since there is no pressure on the thread when

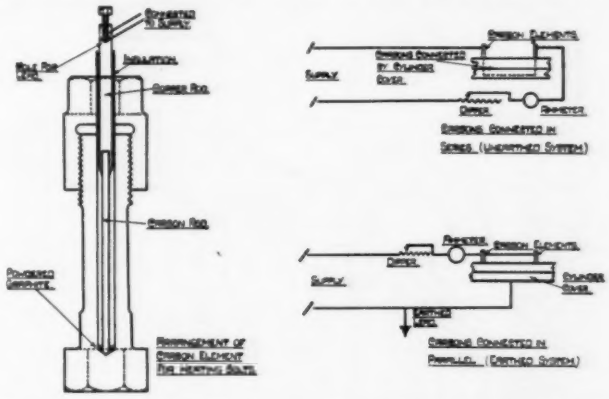


FIG. 2 ELECTRICAL METHOD OF TIGHTENING CYLINDER BOLTS

the nut is turned, it has been found very useful in reducing the seizing of nuts, which used to cause so much trouble when machines were being opened out after prolonged running on high-temperature steam. As a further precaution against seizing, it has been found advantageous to spray the threads of all bolts with aluminum. The aluminum appears to protect the surface of the steel from oxidation, and though an oxide film probably forms on the aluminum, this can be sheared comparatively easily when it is desired to slacken the bolt.

As an alternative to the electric method, gas heating is sometimes employed, but it is open to the objection that it is not possible to heat the bolt rapidly and uniformly over its whole length by gas. Furthermore, gas is not available in many power stations, and if portable cylinders of compressed gas have to be employed, the method becomes relatively costly.

Blade Materials. Blade corrosion, a few years ago one of the most frequent causes of turbine troubles, is no longer a source of anxiety to the operating engineer. This is due primarily to the development of the so-called stainless steels. The refinements introduced into the feedwater equipment of the power station of today have set a high standard for the purity of the feed, and it is seldom that a case is met with where the steam contains any appreciable quantity of impurities likely to cause serious corrosion of modern materials. For general purposes, the most useful grade of stainless steel is that known in the steel trade as stainless iron. It contains about 13 per cent of chromium and has a carbon content of 0.1 per cent. Stainless iron is to be preferred to the higher carbon stainless steels because it is freer from the danger of becoming brittle due to air hardening during manufacture. Its mechanical properties are such that it is suitable for the higher-stressed low-pressure blading, at least up to tip speeds of 1050 fps, and it gives equally good results at the high-temperature end of the turbine. For brazing, chromium steels require treatment different from carbon steel, chiefly because the chromium oxide requires a longer time for fluxing. Some difficulty was experienced when it was first used due to the failure of manganese-copper shrouds which began to disintegrate in service and were found to have been permanently affected by the high brazing temperature. The difficulty was eliminated by employ-

ing monel metal instead of manganese copper for the shroud material.

Stainless iron is not suitable in cases where salt is present in the steam, as it is not highly resistive to chlorides. Occasionally in industrial plants it is not possible at all times to keep the concentration of salt in the boilers low enough to prevent some chloride being carried over by the steam. In such cases the only satisfactory solution appears to be the use of more costly materials for the blading. Either austenitic steels or non-ferrous alloys such as monel metal may be used. It is, however, essential to

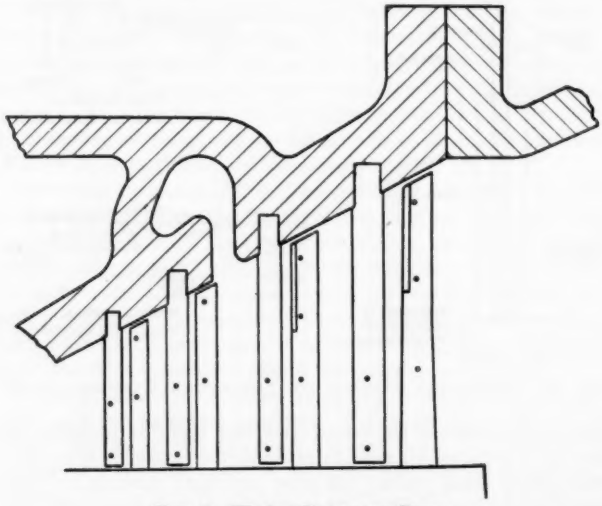


FIG. 3 WATER-DRAINAGE BELT

take all possible steps to avoid the presence of chlorides, as even if a resistive material is employed for the blading, the problem of corrosion of other parts, such as valve spindles, gland journals, and cylinder bores, still remains.

Erosion. Just as the difficulty of corrosion was being successfully mastered, a new and unforeseen blading problem began to present itself. As advances were made in peripheral speeds, a stage was reached at which the erosion of the exhaust-end blades by the impact of the water of condensation became rapid enough to suggest that it might prove to be the most serious factor limiting further progress.

Much confusion has resulted in the past from a failure to distinguish the difference between erosion and corrosion. Properly applied, the term "erosion" means wastage by a purely mechanical action—that is, by the disintegration of the surface layer of the blade into minute particles, without any chemical change whatever.

It is convenient to divide the methods employed to prevent erosion into two classes, according to whether they aim at the production of blades capable of resisting the action of the water or whether they depend on reducing the quantity of water striking the blades by withdrawing it from the turbine as it condenses. To meet the severe conditions obtaining at the present day in high-speed turbines, it is desirable to make use of both of these methods. By this means the longest life for very high-speed blades can be obtained. Were it possible to eliminate the water entirely, this would of course remove the cause of the erosion, and would therefore be the most direct solution. Unfortunately, however, there are fundamental reasons why the water cannot be entirely eliminated. Properly arranged, a simple system of drainage can be made to remove an appreciable quantity, but it does not appear to be possible even with the most elaborate arrangement to get rid of anything approaching the whole of the water.

It is reasonable to assume that the water that causes erosion of the last row is limited to that which condenses out and forms drops on the blades of the preceding guide row. The water condensed in the earlier stages will have been thrown out to the bore of the casing by centrifugal force and will for the most part be retained there, passing through to the exhaust as a layer on the cylinder. A small proportion of this water at the top will tend to leave the casing, as the effect of gravity is not quite negligible. This is probably responsible for the rounding off of the blade tips frequently observed.

A form of drainage belt arranged to trap as much as possible of the water accumulated on the casing is shown diagrammatically in Fig. 3. This belt is connected to a small feed heater, through which a portion of the condensate is passed in parallel with the ejector condensers. This facilitates withdrawal of the water by creating an appreciable outward flow of steam through the port. It also will be noted that the tips of the rotor blades in the preceding row fractionally overlap the metal of the casing. The layer of water flowing from the casing bore into the mouth of the belt acts as a seal to prevent steam leakage.

With a view to producing blades with the highest possible degree of resistance to erosion, the firm with which the author is associated has carried out extensive researches into the problem, and as this has led to some rather striking conclusions, a brief description will perhaps be of interest.

Test equipment was constructed in which the conditions of an actual turbine exhaust were reproduced. It consisted essentially of a bladed spindle that could be rotated in a casing containing steam at high vacuum and charged with a spray of water drops which traversed the path of the blades. The drive was arranged so that speeds up to 1100 fps could be obtained. The water spray

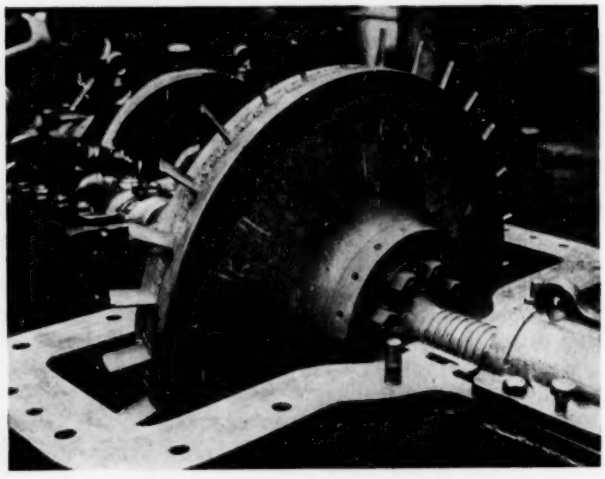


FIG. 4 EROSION-TEST EQUIPMENT

was produced by two oil-fuel burner nozzles that discharged the water in the form of a spinning cone of finely divided particles. The spray was arranged for the correct drop size and gave as fair a representation of turbine conditions as is obtainable, but the actual quantity of water producing erosion was increased so as greatly to accelerate the action.

A photograph of the apparatus showing the bladed rotor is reproduced in Fig. 4.

At the outset it was realized that, in general, the harder the material the higher would be its resistance to disintegration by a purely mechanical action, but it was not known definitely whether other factors might enter into the question that would make some materials more suitable than others by virtue of properties not directly related to hardness. The investigation showed con-

clusively that hardness is the essential property which determines the resistance to erosion by water impact. This property is most conveniently measured by a Brinell or equivalent test, and the comparative resistance of different materials may be predicted directly from their Brinell numbers.

It appears logical to suppose that ductility would influence the rate of disintegration by erosion as between two materials of equal hardness. This, however, can only be inferred and has no direct practical application, because among materials of the class, which by virtue of their extreme hardness are most effectively employed to resist erosion at high impact speeds, it is not possible to obtain any appreciable ductility.

In commercial practice the greatest degree of hardness can be obtained with the high-speed tool steels appropriately treated, and these have been shown to possess the highest resistance and have been found to be the most suitable for blade protection.

Another important fact that had not previously been appreciated was brought out by these experiments—namely, that water impact will cause erosion even though the yield point of the material is far above the calculated value of the resulting fluid pressure. The most satisfactory explanation for this is afforded by the supposition that the pressure is generated, not on a smooth surface, but at the bottom of a small pit in the metal. In this case tensile stress much greater than the pressure itself would be possible across the fibers surrounding the apex of the cavity. As a consequence of this, though theoretical considerations enable an expression to be found connecting the fluid pressure with the velocity of impact, no similar relation exists between the velocity and the stress produced in the metal. It is therefore not possible to deduce any useful relation between the yield point of a material and its resistances or to obtain experimental confirmation of the calculated value of the pressure. For the same reason it is not possible to find for any material a limiting or critical value for the impact velocity below which erosion cannot occur.

A typical example of the form of blade now used is shown in Fig. 5 to illustrate the manner in which the protection against erosion is applied. The particular example figured is employed for the last row in 3000-rpm machines with tip speeds in the neighborhood of 1000 fps. The blade itself is rolled integral with the root and is tapered in section from the root to a point halfway up the blade. From this point to the tip the profile is parallel and has the section shown in the figure. The ratio of the cross-sectional area at the root to that of the parallel upper portion is 3 to 1, a value which permits of an efficient design of blade and at the same time keeps the centrifugal stresses moderate. The blade profile is twisted so as to give the appropriate outlet angle at each individual point in its length. An independent erosion-resisting shield is attached to the upper portion of the inlet edge and on the convex side. As the blade and shield are separate, each can be made in the material best adapted for its purpose. The properties required in materials suitable for blades differ fundamentally from those required to resist erosion. A blade material must, for example, possess adequate ductility to preclude the possibility of failure from fatigue under the influence of any vibration stresses to which it may be subjected in operation. It also must be suitable in every way for the processes of blade manufacture. On the other hand, the essential property for resisting erosion is hardness, a property possessed in the greatest degree by materials of a class quite unsuited for making into blades, but which can quite conveniently be employed for shields.

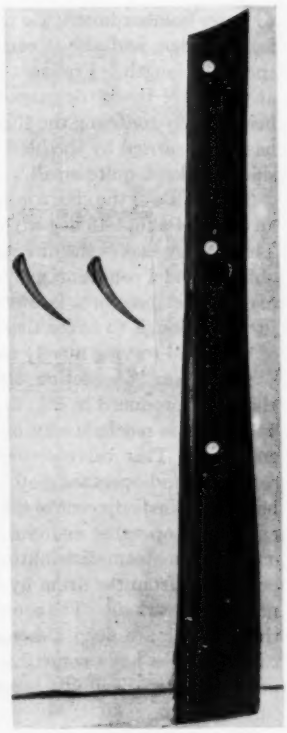


Fig. 5 Protected Rotor Blade

Another feature of the separate shield is the ease with which it can be removed and renewed. If in process of time the shields are worn down and replacement becomes necessary, the operation of renewing them is a very small matter in comparison with the alternative of replacing the complete row of blades. As the shield need be applied only over that area of the blade which is subjected

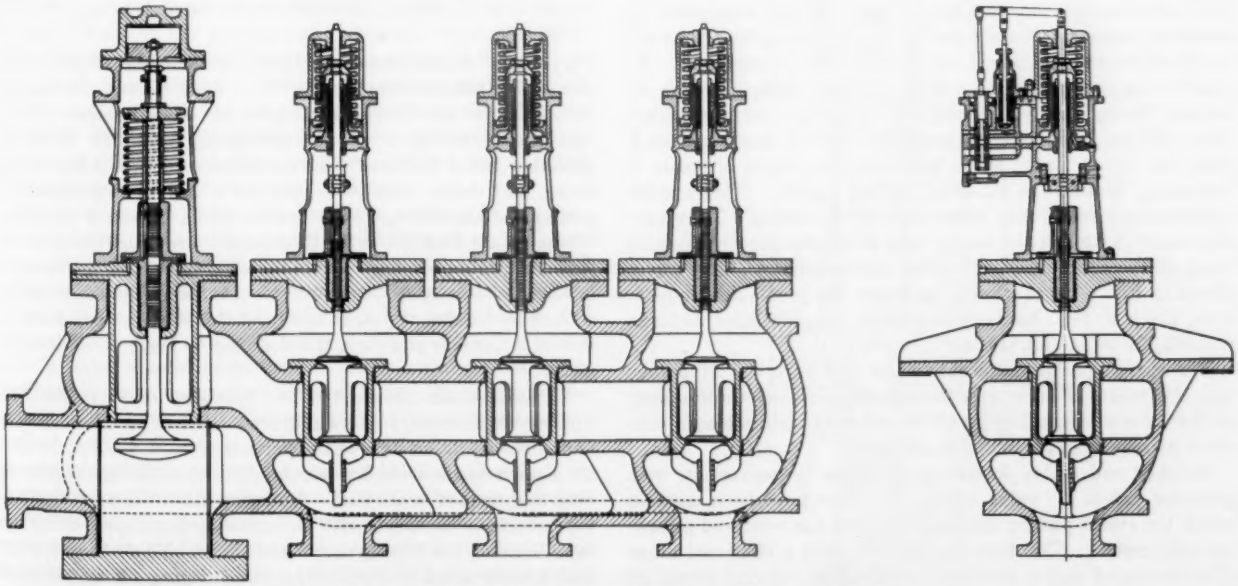


Fig. 6 High-Pressure Steam Chest and Governor Relay

to water bombardment, the protection can be concentrated on the leading edge, and also it can be confined to a limited portion of the blade length. Frequently the erosion is confined to a region at the tip of the blade extending for, say, one-third of the blade height. By confining the shield to this area, the extra load which has to be carried by the blade, due to the centrifugal force of the shield, is kept quite small.

The profile of the shield is beveled so as to have a flat face set at an angle of about 45 deg with the plane of rotation of the blades. This feature makes the direction of impact of the water and blade oblique, and a condition much less favorable to the building up of high impact pressures is therefore obtained. The bevel extends just far enough to cover the whole of the exposed area, the tail of the shield serving merely to blend it into the blade profile.

Valve Gear. A section through a large high-pressure steam chest is reproduced in Fig. 6. The first valve is the emergency valve and is mechanically operated from a centrifugal runaway governor. This valve serves as an emergency control only, a separate hand-operated stop valve for use in starting up the plant being provided adjacent to the steam chest. In addition, a motor-operated stop valve manipulated from the control room is fitted on the main steam-distributing drum. The turbine can therefore be isolated from the drum by two positively closing valves during periods of overhaul. The connections between the steam chest and the cylinder are deep U-bends of corrugated pipe. These corrugated pipes have come into very general use for a variety of purposes where flexibility is required, both in the boiler house and turbine room.

There are three governor valves, all of equal size, fed from the common central port of the chest and controlling the steam for 60, 80, and 100 per cent load, respectively. The emergency valve and the governor valves are all forged in one piece with their spindles. This makes them very robust and avoids the use of nuts or other fixings, exposed to the full temperature of the steam, which are likely to be a source of weakness. The spindle glands are plain bushes of pearlitic iron, with a fine clearance and provided with two leak-off ports for the leakage steam. Pearlitic iron has proved most successful for this purpose and gives no trouble through distortion, even on the highest temperature machines.

The design of the governor valves is a matter of some importance. The momentum of the steam passing through the valve at high velocity is very large, particularly where the steam pressure, and consequently the density, is high. If this momentum is altered by contact with some part of the valve away from the seat, considerable out-of-balance forces on the valve are produced. As these forces depend on the weight of steam passing and on its velocity through the valve seat, their magnitude varies considerably with the load, being in general greatest at about half-load. They fall off at higher loads, although the weight of steam is increasing, because the velocity is falling rapidly. These rather indeterminate forces may cause unstable governing. They may also cause chatter of the valves, and as a consequence excessive wear, and sometimes actual fracture of the spindles. The valves shown in Fig. 6 are of what is known as the profile double-plate type, and their form has been evolved so as to eliminate, as far as possible, forces of this nature.

The curved profiles adjacent to the seat have the advantage that the ratio of effective area to valve lift increases continuously as the valve opens, a feature which assists in maintaining a constant sensitivity of governing at all loads.

Another and rather interesting problem in connection with governor valves is "valve whistle." There have been cases in which the steam passing through the valve has produced a high-pitched screech. This has occurred only over a restricted range of valve travel and is produced by the high-velocity steam jet from the valve impinging on the edge of some part of the seating or chest. Although it has proved difficult to locate the exact point where the screech was produced, it has been effectively silenced by an alteration to the valve profile.

The arrangement of the governor relay is shown in Fig. 6. The small auxiliary piston shown between the power piston and its control plunger is operated by an independent oil system, the pressure in which is controlled by the speed governor. The regulation of the pressure in this auxiliary system is effected directly by the movement of the governor sleeve, so that the connection between the governor and the power relay is by an oil link and not by a lever train.

In the case of reheat units, control of the steam stored in the reheater is effected by the provision of valve chests at the inlet to the low-pressure turbine. The valves are controlled by a centrifugal governor independent of the main governor, and are arranged to be normally full open, but to respond very quickly to the governor if the turbine speed reaches 5 per cent above normal. This avoids the danger of the machine tripping out on overspeed due to the reheater capacity.

Turning Gear. The practice of providing large machines with motor-driven turning gears is now becoming general. The necessity for such a fitting arises from the fact that the cooling of a stationary machine after shutting down does not proceed evenly. As the heat stored in the hot end of the spindle gradually soaks through into the colder end, the average temperature of the

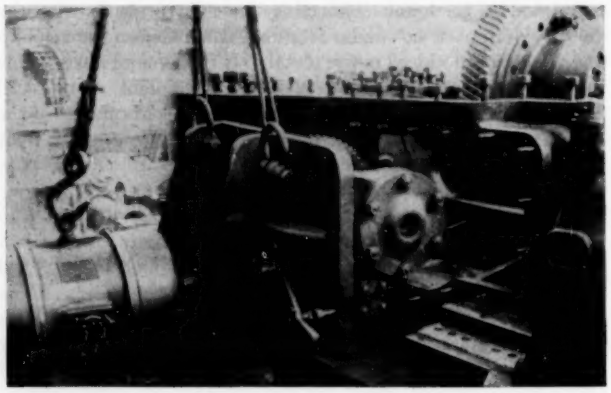


FIG. 7 TURNING-GEAR ASSEMBLY

top portion tends to rise above that of the bottom portion. The greater the temperature gradient initially and the more massive the shaft, the more pronounced does this effect become. Large shafts may thereby be bowed sufficiently to throw them appreciably out of balance. The distortion may persist for several hours, and during this period difficulty may be experienced in getting the machine up to speed without objectionable vibration. Owing to the time taken for the temperature distribution to adjust itself, vibration may persist for some time after the machine has been put on load. To avoid this it is necessary to control the cooling by turning the shaft slowly; if this is done a symmetrical flow of heat takes place and the shaft has no longer a tendency to distort.

To achieve this result it is not necessary to keep the shaft turning continuously; it is sufficient to turn it at intervals, perhaps once an hour, through a portion of a revolution, so that the same influences which produce a distortion during one interval tend to correct it in another. A turning gear of this intermittent type should be arranged with an extremely large speed-reduction ratio between the driving motor and the turbine shaft, as a dead-slow turbine speed is of advantage in producing a large torque to overcome the initial friction. If, however, the turning gear is of

the continuous type, the speed of the journals must be high enough to maintain the oil film in the bearings; i.e., it must be of the order of 1 fps, corresponding to, say, 15 rpm rolling speed for an average 1500-rpm machine. As the motor speed in this case would normally be 1500 rpm, the gear ratio is limited to about 100 to 1. To obtain a starting torque capable of overcoming the friction of a dry bearing, an unduly large motor would be required. It therefore becomes necessary to provide a high-pressure oil system connected to the bearings, which can be used to lift the journal off the bearing and to establish an oil film before the turning-gear motor is switched in. Oil pressures of 1000 to 1200 lb per sq in. are usual for this purpose, supplied from a small motor-driven, multi-throw plunger pump.

Apart from the turning of the shaft during a temporary shutdown, the turning gear has another application, of particular importance for machines operating with very high-temperature steam, and that is to simplify the process of starting up from cold. For this purpose the turning gear must be of the continuous type, which on this account is to be preferred to the intermittent type. It is essential when starting a cold machine to have the spindle rotating during the warming-up period; otherwise unequal heating of the spindle is almost unavoidable. If it is started from rest by steam, a very large amount of hot steam has to be passed suddenly into the turbine, and this involves subjecting the cold cylinder and valve chests to severe temperature shocks. If, however, the turning gear is utilized to rotate the spindle, only enough steam need be admitted to warm the plant up gradually and uniformly. In this case the procedure is, first, to raise some vacuum, say about 10 in., then to engage and start the turning gear and open the stop-valve bypass a small amount. As the vacuum rises, a point will probably be reached where the warming-up steam is sufficient to keep the spindle rolling. The turning gear will then automatically disengage, and the motor can be switched off. When the vacuum has reached, say, 25 in., running up to speed is commenced.

The location of the turning gear on the turbine depends on the arrangement of the machine. On single-cylinder units or tandems with double-flow low-pressure cylinders, the motor and gear have usually to be accommodated between the cheeks of the exhaust, and the drive is transmitted to the turbine shaft through a toothed wheel attached to the coupling. A very neat arrangement can be obtained when the unit includes a single-flow intermediate-pressure or low-pressure turbine, by fitting the gear below the main shaft in the intermediate pedestal, the motor being supported on a bracket at floor level, as shown in Fig. 7. In the example illustrated, the first reduction is a worm gear, but a helical gear is used for the second reduction, as it has the advantage of being independent of the axial adjustment of the spindle. The pinion slides on a long-pitched screw thread, and engagement is effected by a hand lever.

Shaft Alignment. There has been much variation in the practice of different makers with regard to the type of coupling employed between turbine and alternator rotors. The use of a solid flanged coupling is attractive, because it leads to a simplification, not only in the coupling itself, but also in the thrust blocks, and thereby effects some saving in length. There is today a movement in favor of solid couplings, but there is some hesitation in introducing them generally. While many installations are operating excellently with solid couplings, there have been others in which the experience has not been so satisfactory and where it has proved necessary to revert to couplings of the flexible type. There seems to be no reason to doubt their suitability provided the balance of the individual rotors is good and the alignment accurate. Such conditions should not be difficult to obtain—for example, between a low-pressure turbine and its alternator rotor; and here, therefore, the application of a solid coupling seems

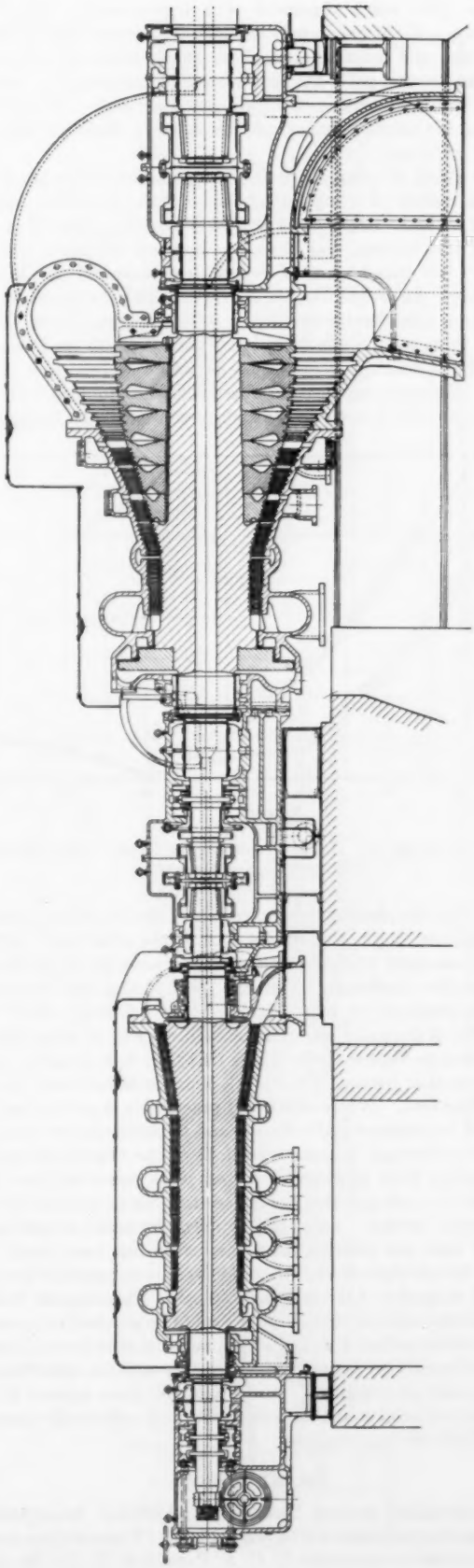


Fig. 8 50,000-Kw Tandem-Cylinder Turbine, 1500 Rpm (Parsons)

logical. The exact alignment of a high-pressure cylinder is, however, not always so easy to achieve, because the supports, pedestals, and concrete foundation blocks alike are subject to considerable changes of temperature due to the proximity of steam pipes, and their expansion affects the alignment. To meet the varying conditions arising from this cause, a flexible coupling is to be preferred.

The effects of unequal heating of foundation blocks are nowadays a matter of some importance. Many machines are arranged with the steam chest and its connecting pipes all on one side of the turbine, and the pipes between the chest and the cylinder are usually carried several feet down for the sake of flexibility. Consequently, the concrete block gets heated on this side for a considerable proportion of its height. With a high steam temperature this may have the effect of producing an appreciable differential expansion between the two sides of the block. Instances can be cited where vibration troubles have been experienced due directly to the high-pressure cylinder being lifted

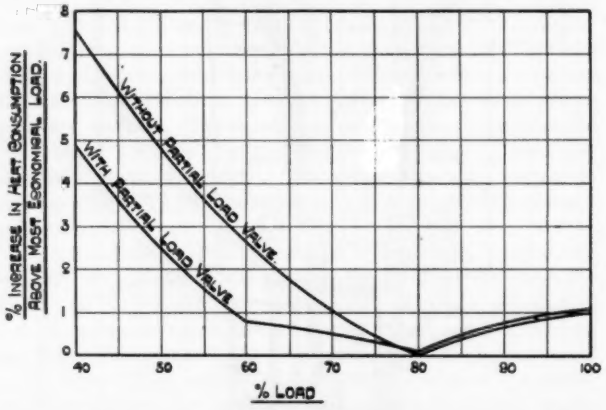


FIG. 9 CURVES OF HEAT CONSUMPTION WITH AND WITHOUT PARTIAL-LOAD VALVE

clear of the sole plate on the cooler side of the foundation, and so being left free to pivot about the foot on the other side. It has become necessary to take this factor into account in the design of foundation blocks and to arrange them so that such temperature gradients as are unavoidable do not adversely affect the alignment of the machine. The most direct way of doing this is of course so to dispose the hot parts that they do not radiate heat directly to that portion of the block which is immediately under the turbine foot. Where this is not practicable, a screen may be provided to intercept the radiation and to distribute the heat by conduction through the cooler portions of the foundation block. The simplest form of screen is a steel plate separated from the concrete by an air gap over the hot section, but in contact with it over the cool section. Cases have come under notice where harsh running that has resulted from this cause has been cured by setting the sole plate down on one side so as to compensate for the unequal expansion of the foundation block. This suggests that a very rational solution of the problem could be attained by mounting the steam end on a pad of elastic material which would compress sufficiently to maintain a reasonably uniform distribution of load under all conditions. Unfortunately, there appears to be no elastic material available with a structure sufficiently permanent to fulfil the requirements.

RECENT DESIGNS

A longitudinal section through a 50,000-kw two-cylinder 1500-rpm reaction turbine is shown in Fig. 8. Three of these units are now under construction by C. A. Parsons & Co. for the new station that is being erected at Dunston in connection with the conversion of the northeastern section of England to 50-cycle supply. The turbines are designed for 600 lb stop-valve pressure, 800 F temperature, and 29 in. vacuum, with intermediate reheating up to the initial temperature, and feed heating by bled steam to 350 F. The most economical point is at 40,000 kw load, or 80 per cent of the maximum continuous rating. With the operating conditions obtaining in England, this ratio of economical to maximum load seems to give the best overall results and has long been a standard that has not been departed from. The practice of building machines with large overload capacity has never been introduced in this country, though in point of fact by the use of partial-load valves the consumption at, say, 60 per cent can be brought down so near to the 80 per cent figure that a generally similar consumption characteristic is obtained down to this point.

The form of the heat-consumption curve for a machine with a partial-load valve is shown in Fig. 9, compared to that of a similar unit without the valve. This curve assumes that the condenser is designed for the 80 per cent load and that the circulating-water quantity is kept constant, so that the vacuum varies with the load. A rather curious condition arises in this connection in the case of a reheat machine, in that the steam consumption at 60 per cent load is actually lower than that at the most economical one, though the heat consumption is a little higher. This is of course due to the fact that at the lighter load more heat units are put in per pound of steam in the reheater.

The two-cylinder design with a single-flow low-pressure turbine enables the reheating to be arranged between the cylinders. This is a material advantage, because for this output at 1500 rpm the provision of reheating steam belts on the high-pressure turbine presents difficulties, and would involve some sacrifice of efficiency in this turbine. The tip speed at the exhaust end of the low-pressure turbine is very moderate, and ample exhaust area is obtained with both shaft and disks of such dimensions that very simple forgings can be employed for their manufacture.

The reheat pressure in this case is 115 lb per sq in. The glands of the high-pressure turbine are composite, a labyrinth being provided to break down the pressure, followed by a carbon-ring gland for sealing. The port between the two is connected to the low-pressure feed-heat belt. Water-sealed glands, so general in America, have never achieved the same measure of popularity in England for high-pressure turbines. This affords an interesting instance of divergence of practice in the two countries. Properly designed steam-sealed glands are undoubtedly remarkably free from mechanical troubles, but so long as they were provided with vent pipes they were apt to use them to excess This weakness has now been eliminated by the introduction of small condensers in place of the vent pipes. These may be of a very simple type, usually consisting of a coil of copper tube arranged to pass a small quantity of condensate from the extraction pump. The "visible discharge" from the gland is retained by allowing the condensed steam to drip into an open funnel.

In Fig. 10 is shown a three-cylinder turbine at 1500 rpm designed for conditions similar to the machines just described. This arrangement would be adapted for outputs of 75,000 to 100,000 kw. The steam is reheated between the h-p and i-p cylinders and is comparatively cool when it enters the double-flow low-pressure turbine, at which point the pressure is slightly below atmospheric.

Fig. 11 shows a section through a 75,000-kw three-cylinder tandem unit at 1500 rpm, by the British Thomson-Houston Company, at Rugby. Two of these units are being supplied for the new Barking Power Station of the County of London Electric Supply Company. The turbines are designed for a steam pressure of 600 lb per sq in. at the stop valve and a tem-

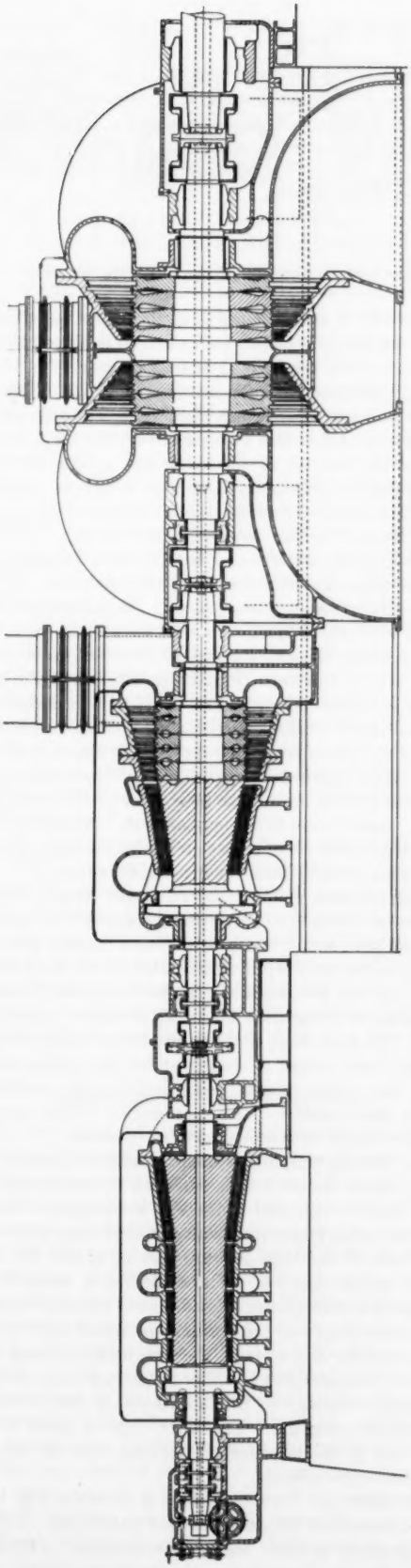


FIG. 10 75,000 TO 100,000-KW THREE-CYLINDER TURBINE, 1500 RPM (PARSONS)

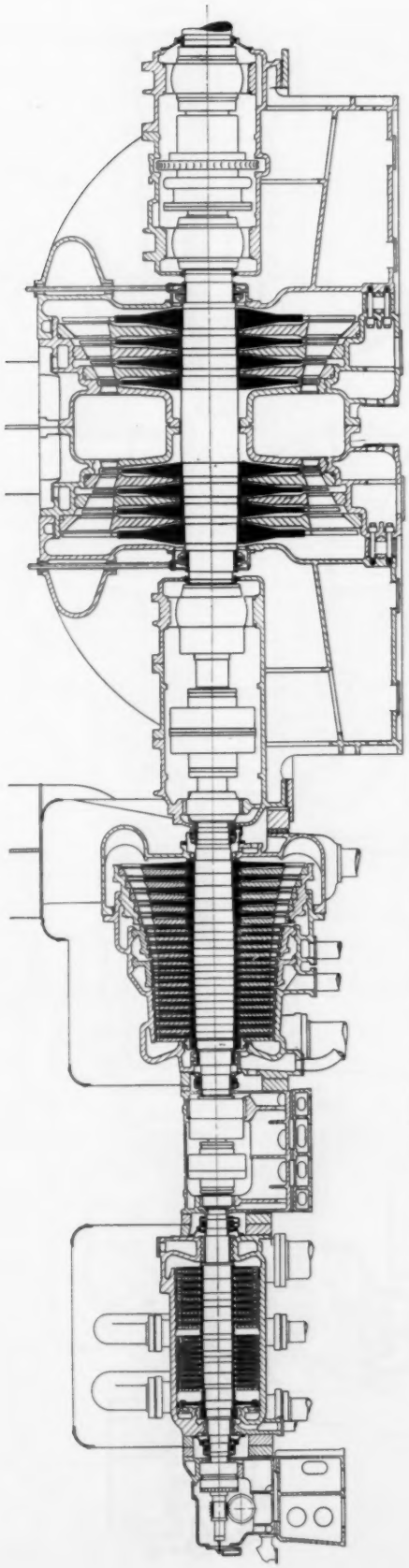


FIG. 11 75,000-KW THREE-CYLINDER-TANDEM TURBINE, 1500 RPM (BRITISH THOMSON-HOUSTON COMPANY)

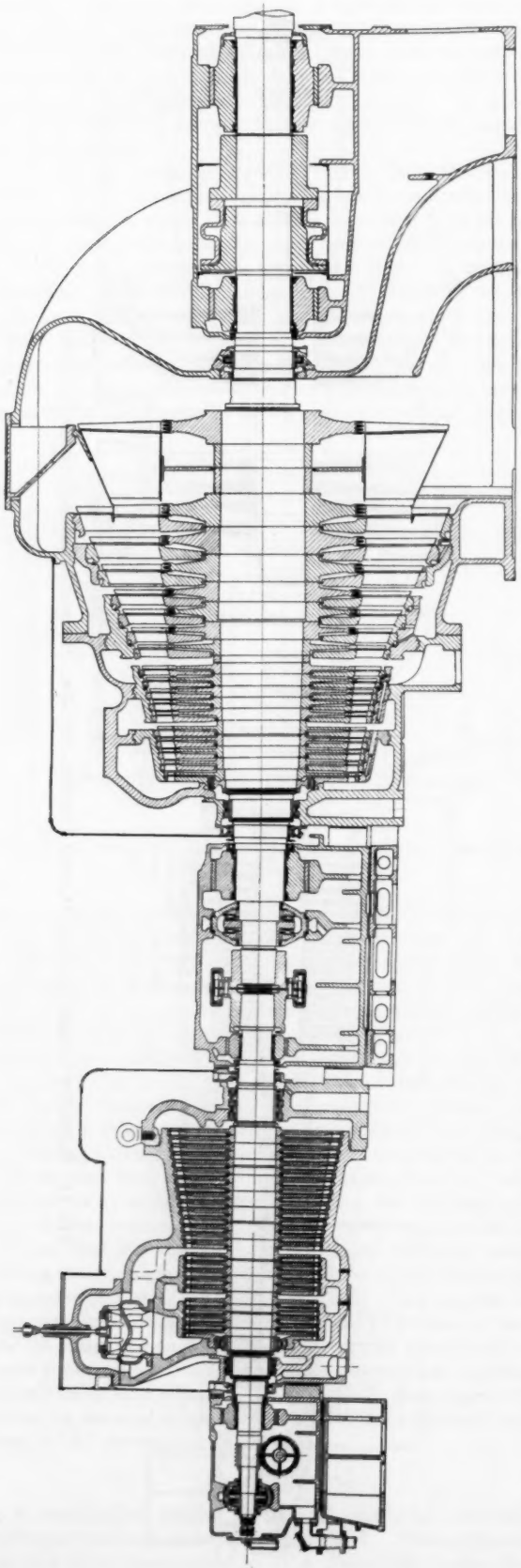


FIG. 12 51,250-KW TANDEM-CYLINDER TURBINE, 1500 RPM (METROPOLITAN VICKERS COMPANY)

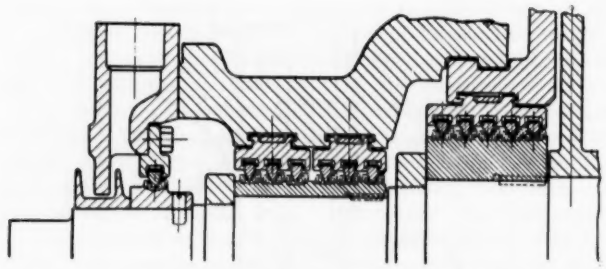


FIG. 13 LABYRINTH GLAND (METROPOLITAN VICKERS COMPANY)

perature of 800 F and vacuum of 29 in. The most economical load is 60,000 kw, and at this load the feed is heated to 340 F by steam tapped from the turbine at four places.

The high-pressure cylinder contains a Curtis wheel and 19 single-row wheels of 36 in. mean diameter. The two-row stage is used for partial loads and is bypassed at the most economical. For maximum load, 10 of the single-row stages are bypassed. The intermediate-pressure cylinder has 20 stages, and from its exhaust the steam is conveyed through overhead interconnecting pipes to the double-flow low-pressure cylinder. This cylinder has only four stages at each end, with a mean diameter of 116 in. at the last row. Flexible couplings are used, and each rotor is located axially by its own thrust block. In the case of these units the glands in all three cylinders are of water-seal type.

A two-cylinder tandem turbine by the Metropolitan Vickers Company for the Clarence Dock Power Station of the Liverpool Corporation is shown in Fig. 12. This unit is designed for an economical load of 41,250 kw, with a continuous maximum rating of 51,250 kw, and runs at 1500 rpm. The steam conditions are 400 lb per sq in. stop-valve pressure, 800 F temperature, and 29 in. vacuum, but the full output can be obtained with 340 lb per sq in. stop-valve pressure and 675 F temperature. Steam for feed heating is supplied from three points on the l-p turbine, and an additional tapping is taken from the h-p turbine exhaust.

The high-pressure cylinder has 25 stages ranging from 42 in. to 53 in. mean diameter. On overload the first five stages are bypassed, and provision is made to bypass a further group of five stages to enable the output to be obtained on the lower steam pressure. In the low-pressure cylinder there are 17 stages with mean diameters rising from 60 in. at the inlet to 96 in. at the exhaust. The final stage is of the Baumann multi-exhaust type. A built-up construction is employed for the high-pressure diaphragms, the nozzle blades being individually machined and riveted to steel centers. The diaphragms of the low-pressure cylinder are of cast iron with cast-in steel blades.

A Bibby flexible coupling is used to connect the high-pressure and low-pressure turbine rotors, while the generator coupling is of the semi-flexible type, and consists of a corrugated-steel sleeve. Each turbine rotor is provided with a Michell thrust block.

The glands of the high-pressure turbine are of the labyrinth type with radial contacts. These glands, a detailed view of which is given in Fig. 13, are formed of steel segment rings. Both the stationary rings and the housings in which they are carried are supported by flat springs, the arrangement being intended to permit of displacement of either individual rings or the entire group without damage to the labyrinths, in the event of accidental contact occurring. This same type of gland is fitted at the inlet end of the low-pressure turbine, but the exhaust end has a water-sealed gland.

A recent 3000-rpm Parsons turbine is shown in Fig. 14. This is a tandem machine with an output of 30,000 kw. The double-flow low-pressure cylinder has a mean diameter of 60 in. at the last row. Fig. 15 illustrates a three-exhaust design, with the

same blade annulus at the exhaust, to give an output of 50,000 kw at 3000 rpm. This is a three-cylinder turbine arranged for reheating.

The English Electric Company has built a number of three-cylinder tandem machines at 3000 rpm, one of which is illustrated in Fig. 16. This is a 30,000-kw unit designed for a stop-valve pressure of 200 lb per sq in., 700 F temperature, and 28.5 in. vacuum, with feed heating to 227 F.

The high-pressure turbine is of the impulse type, with twelve stages of 36 in. mean diameter. The low-pressure consists of two single-flow cylinders with reaction blading arranged to exhaust into the condenser through a common opening. The mean diameter of the last row is 60 in. The interconnecting pipes between the high-pressure turbine and the first low-pressure cylinder are carried overhead, while the longer pipes to the second low-pressure cylinder are arranged below the floor.

A disk construction is employed for the two low-pressure rotors, and they are connected by a rigid coupling, on either side of which a journal bearing is provided, and also a pivoted pad thrust housed in the bearing shell. The thrust collars are formed on the coupling flanges. As the rotors are opposed, their end thrusts balance and no dummies are required. Semi-flexible couplings of the toothed-ring type are used for the high-pressure turbine and for the generator. Carbon rings are used for all the glands with the exception of the inlet end of the high-pressure cylinder, where a labyrinth is fitted.

In the built-up diaphragms developed by the English Electric Company and used for the high-pressure stages, the nozzle blades are milled in steel blocks, which are shaped so as to form, when assembled, the outer ring of the diaphragm. Countersunk screws are used to unite the outer portions of the blocks. The inner portions have a tenon which is riveted in a groove in the steel diaphragm center, steam tightness being insured by a vee-weld on the discharge side. A detail of the construction of these diaphragms is given in Fig. 17.

A 60-cycle Parsons machine, a 15,000-kw unit at 3600 rpm, is arranged with a duplex exhaust, a construction that has been very successfully used in a number of cases to obtain a large output from a single-cylinder plant, particularly where a short overall length is essential. The steam is divided at a pressure of about 4 lb per sq in. abs, and a single duct of large capacity in the top half of the machine is employed to carry the steam over to the second set of low-pressure blading, the corresponding space in the bottom half forming the exhaust from the first set. (See Fig. 18.)

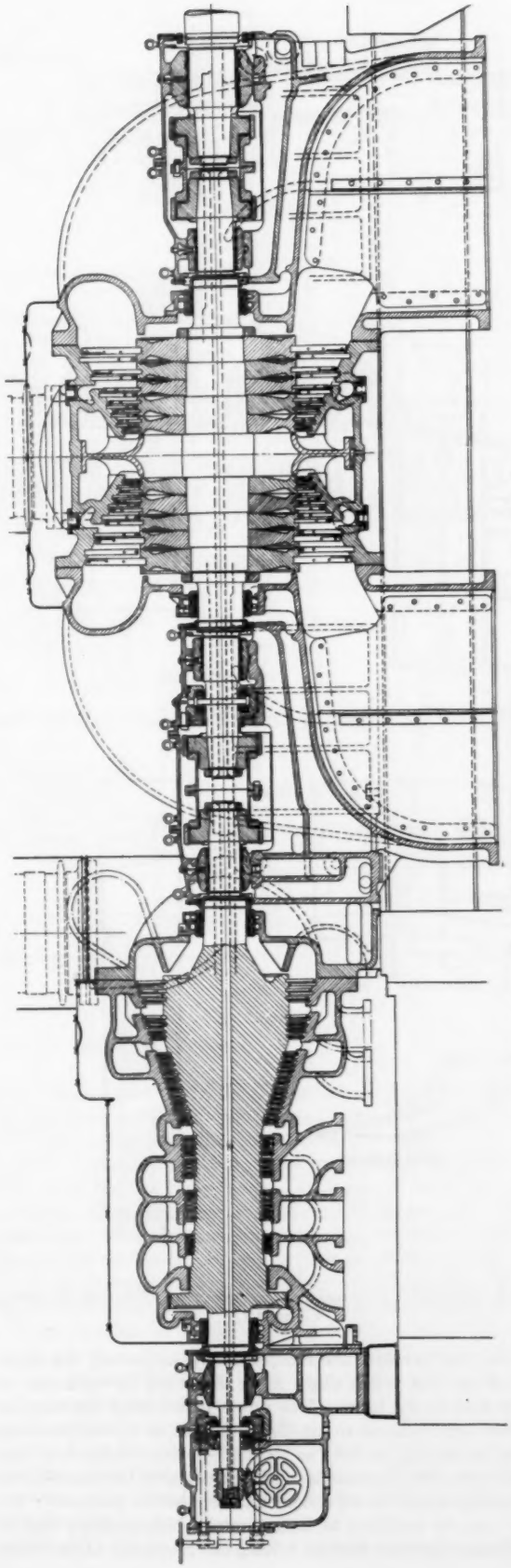


FIG. 14 30,000-KW TANDEM-CYLINDER TURBINE, 3000 RPM (PARSONS)

SOME TENDENCIES OF DEVELOPMENT

Double-Deck Compound Units. Cases sometimes arise in which the floor space occupied by a generating unit is a matter of extreme importance, and in which it is essential to adopt the type of unit which effects the greatest economy in this respect. Hitherto it has been possible in England to meet this requirement very effectively by the use of 3000 rpm, single-cylinder turbines with duplex exhausts. As the size of machines increases, however, it becomes necessary to consider compound units with one section disposed above the other, an arrangement that has already been adopted for a number of units in the United States.

A design of this type recently developed by the Parsons Company is shown in Fig. 19. The distinctive feature is that the high-pressure line is placed beneath the low-pressure line. There is ample space under the low-pressure turbine in which to dismantle the high-pressure turbine or, if required, to remove it complete, as the former is supported quite independently of the latter. Both turbines are thus readily accessible and the covers can be lifted without breaking the high-pressure steam-pipe joints.

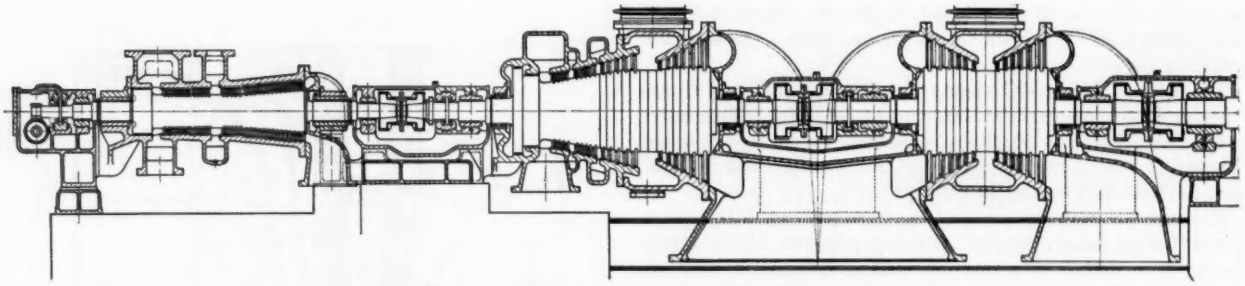


FIG. 15 50,000-KW THREE-CYLINDER TURBINE, 3000 RPM (PARSONS)

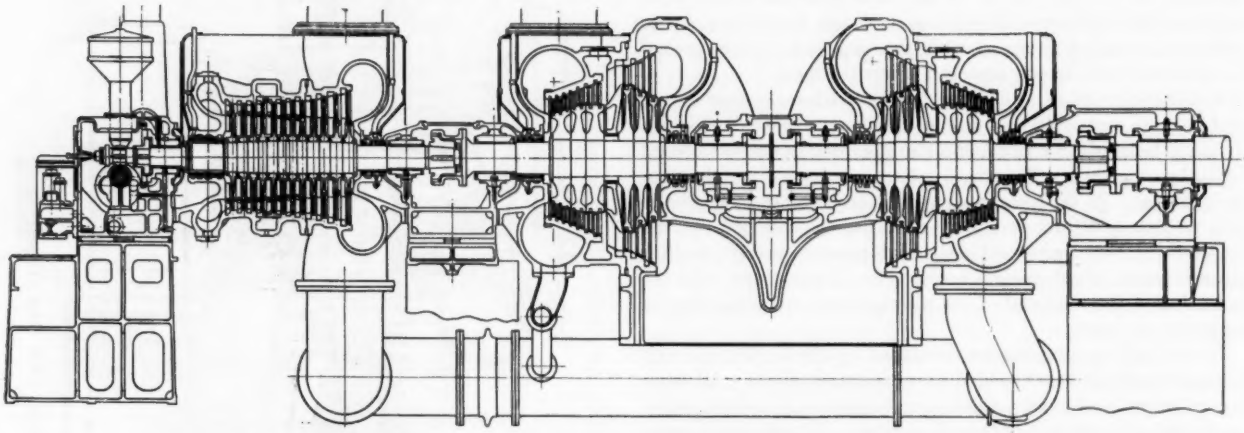


FIG. 16 30,000-KW THREE-CYLINDER TURBINE, 3000 RPM (ENGLISH ELECTRIC COMPANY)

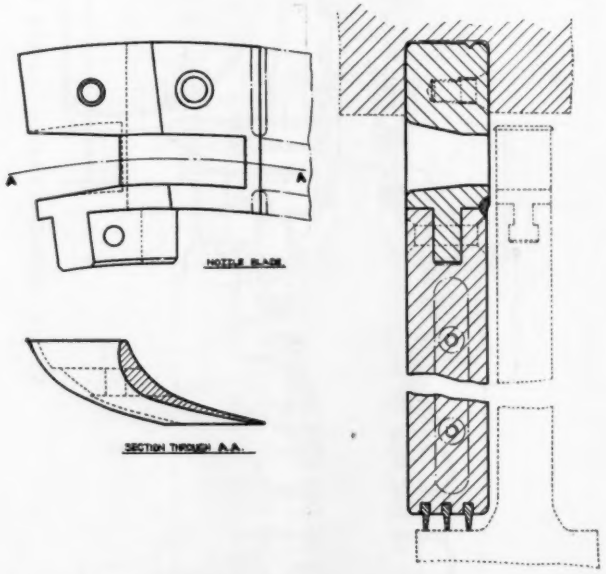


FIG. 17 BUILT-UP DIAPHRAGM CONSTRUCTION (ENGLISH ELECTRIC COMPANY)

As the two turbines are independently supported, the alignment of one line is not likely to be disturbed by variations of temperature in the other. The power distribution between the two units can be chosen to suit the conditions, as it is subject to no special limitations, so long as the high-pressure exhaust is kept within reasonable dimensions. In a reheat unit, for example, the distribution would be dependent on the reheater pressure. The layout can be modified to incorporate a high-pressure and an intermediate-pressure turbine driving one alternator at floor level with either a single- or a double-flow low-pressure turbine above. The arrangement lends itself naturally to the use of side exhausts on the low-pressure turbine leading direct into twin vertical condensers symmetrically disposed on either side, as shown in Fig. 19.

Built-Up Rotors. The output obtainable from a unit at any given rotational speed is limited primarily by the permissible stresses in the disks and blades at the exhaust end of the turbine rotor. Increase of capacity must therefore involve the use of materials capable of sustaining higher stresses, or else a modification of design to allow of higher peripheral speeds without an increase in stress. In the past, advances have been made chiefly by means of the former method, improvements in the steels available leading to the use of higher stresses. It seems possible, however, that a stage is being approached beyond which further increases in tensile strength are hardly to be expected—that is, in materials suitable in other respects to be used commercially for turbine disks. Various designers have therefore directed their attention to means of reducing the stress at a given surface speed, and have adopted or proposed forms of rotor construction that enable profile disks without any central hole to be used.

The ordinary theory of the stress distribution in a rotating disk indicates that a very considerable reduction in the maximum stress can be achieved by avoiding a central hole. Actually, however, this advantage is not ordinarily obtained, primarily because the assumption on which the theory is based—namely, that the material is homogeneous—is not fulfilled. With the usual method of forging disks from cheeses cut from a bloomed octagonal ingot, there is always the possibility of axial weakness; consequently, while the stress in a disk with the center bored out can be estimated with reasonable accuracy, that in the solid disk is much more indeterminate.

A method of casting ingots by which segregation in the center can be avoided has been referred to earlier in the paper, and this

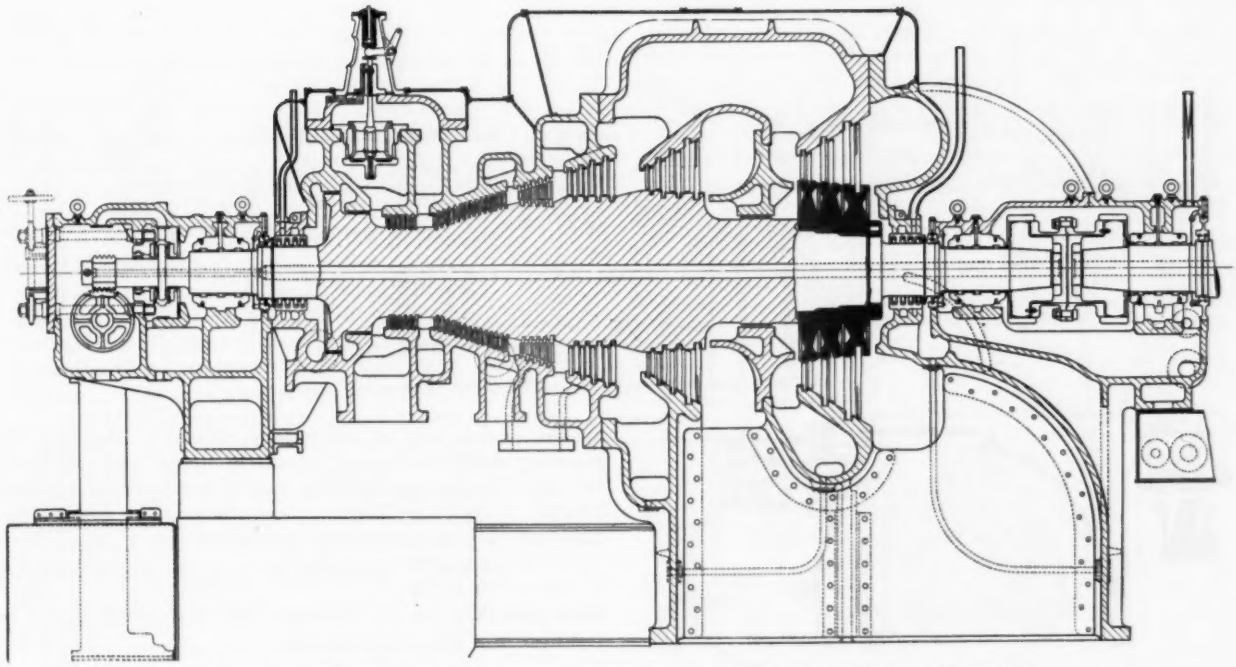


FIG. 18 15,000-KW SINGLE-CYLINDER DUPLEX-EXHAUST TURBINE, 3600 RPM (PARSONS)

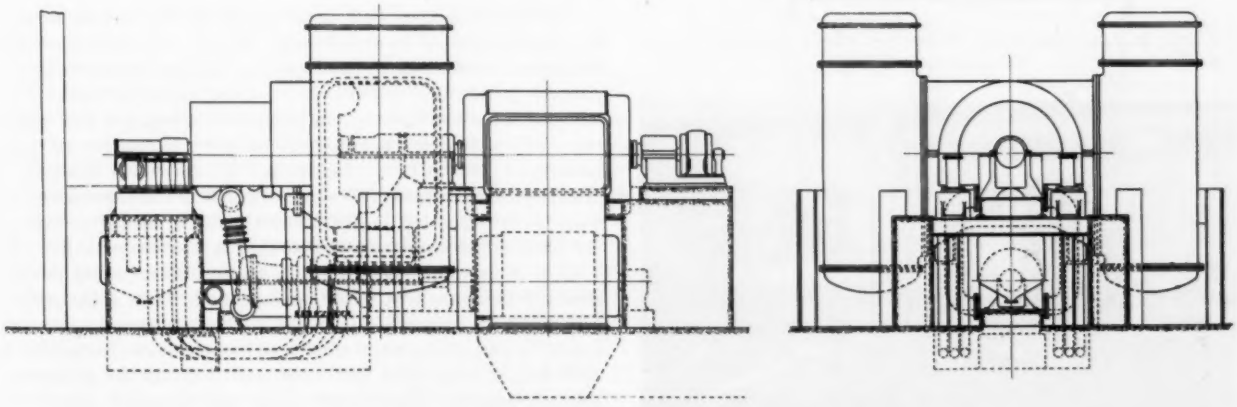


FIG. 19 ARRANGEMENT OF DOUBLE-DECK COMPOUND TURBINE

does make it possible to produce alloy-steel disks reasonably homogeneous throughout. The application of the method to the production of solid disks for built-up rotors does therefore seem to render this a logical development.

Various forms of construction for built-up rotors may be employed. The disks may be bolted together or they may be screwed or shrunk on to spigots. On the Continent, the Brown-Boveri Company has introduced a construction in which the disks are spigoted on each other and welded together around the periphery.

Hollow Blades. The use of solid disks in the manner just described makes possible a higher peripheral speed without an increase in the disk stress. The same result may be achieved in another way by the use of lighter blades. A considerable percentage of the stress in the disk is due to the load of blading which it carries. If, therefore, a lighter form of blade can be employed, the effect on the disk stress will be appreciable, and, what is equally important, there will be a corresponding reduction in the stress of the blade itself.

Attention is therefore being directed to the development of blade constructions in which the material is more economically disposed than is the case with the usual forms of solid-steel blades. The introduction of hollow blades naturally suggests itself as a means of reducing the blade weight, and various proposals have been made for the manufacture of blades of this type. It is customary in present practice to keep the stress at the root of exhaust-end blades within safe limits by tapering, so that the cross-section at the tip is only a fraction of that at the root. The blade illustrated in Fig. 5 affords an example of this. Tapering of a solid blade has, however, limitations, as a reasonable area of tip section has to be maintained. This is necessary both to obtain adequate stiffness and also to preserve an efficient blade profile. Beyond this point, therefore, a reduction in blade stress could be obtained only by enlarging the root section, which would involve increasing the disk stress. If, however, metal was removed from the core of the blades, a greater reduction in stress could be secured without this disadvantage. There would therefore seem to be considerable scope for the application of hollow blades for this purpose.

Light-Alloy Blades. The use of a suitable light alloy instead of

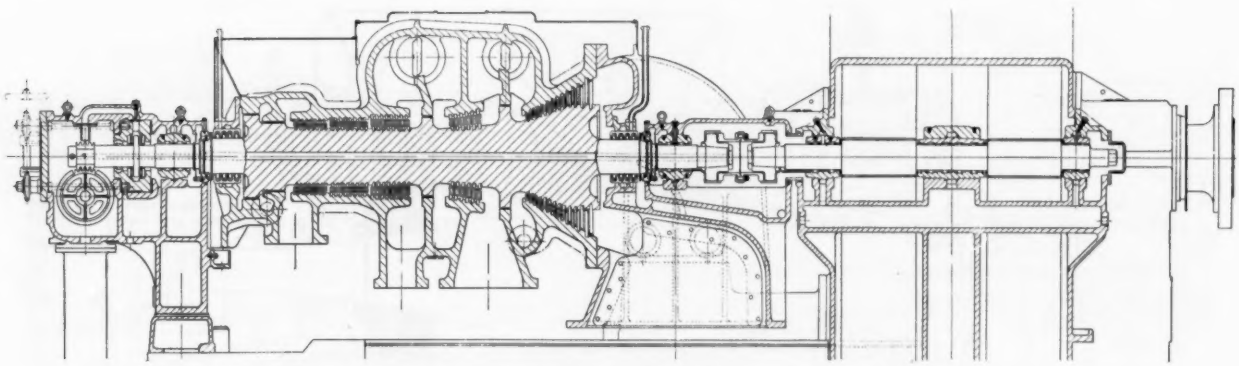


FIG. 20 4000-KW DOUBLE PASS-OFF TURBINE (PARSONS)

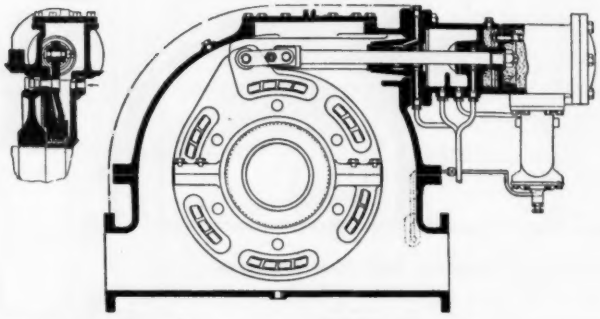


FIG. 21 ROTARY-DISK-TYPE PASS-OUT VALVE (METROPOLITAN VICKERS COMPANY)

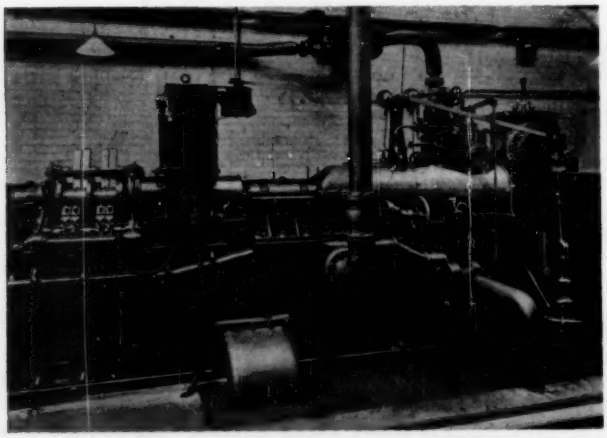


FIG. 22 100-KW TURBO-GENERATOR SET, 1899 (PARSONS)

steel affords another method of securing a reduction in the blade weight. Aluminum alloys are available for which the strength factor, expressed as yield stress divided by density, compares very favorably with that for steels. Similar blades in the two materials would therefore have about the same factor of safety, while the load on the disk for the light alloy would be one-third that for the steel blades, as the density of the former is only one-third that of the latter.

Much progress has been made in recent years in the production of aluminum alloys suitable for moving parts of machines, and their use in various branches of engineering is being constantly extended. The manufacture of blades in these alloys is quite practicable, but their ability to resist corrosion indefinitely in an atmosphere of wet steam is open to question. Another difficulty arises from the fact that the blades would have to be protected against erosion, and no satisfactory method of attaching the necessary shields is at present available. Aluminum solders are not effective, because although a good bond can be obtained initially, corrosion is likely to take place in the solder itself, rendering the joint useless in a comparatively short time. Riveting cannot be employed for a similar reason, as rapid corrosion due to electrolytic action would be set up round the rivets. Until these difficulties can be overcome, light-alloy blades cannot be regarded as a feasible proposition.

PLANTS OF SMALL OUTPUT

Pass-Off Turbines. Industrial plants are for the most part arranged to supply process steam. In the older machines the regulating mechanism for maintaining constant pressure at the pass-off point was accommodated in a separate chest. The whole of the steam had to pass through this chest, and that which was not required for process was returned to the low-pressure blading of the turbine. In present-day machines, however, a much neater arrangement is adopted in which the pass-off control valve is incorporated in the turbine cylinder itself. A typical arrangement of a machine of this type is reproduced in Fig. 20. This is a machine supplied to a paper mill requiring process steam at two pressures, so that two pass-off valves are provided. The boiler pressure is 350 lb per sq in., and the process pressures are 35 lb and 18 lb, respectively. The output of the turbine is 4000 kw, at a speed of 4500 rpm, and it drives d-c generators through gearing. The process-steam valves, which are of the butterfly type, are controlled by an oil-operated servomotor relay responsive to the pressures in the process system, the main throttle valve on the high-pressure steam supply to the turbine alone being controlled by the speed governor.

The Metropolitan Vickers Company employs a rotary disk valve to regulate the pressure in the process main. The disk, which is actuated by a power piston through an oil relay, is provided with ports that control the number of nozzles admitting steam to the low-pressure section of the turbine. If the demand for process steam increases, some of the nozzles in the first low-pressure stage are blanked off, and the heat drop available in the condenser steam is maintained. This arrangement is shown in Fig. 21.

Self-Contained Units. Many of the earliest condensing turbo-generators built by the Parsons Company were arranged as a complete unit, with the turbine and generator mounted directly on top of the condenser and with reciprocating circulating and air pumps both driven by worm gearing from the turbine.

Such a unit, built in 1899, is shown in Fig. 22. So long as the bulk was small enough for the complete unit to be accessible from the floor, this arrangement was retained, but increasing capacity made it necessary to provide a separate operating platform at a

level approximately at the top of the condenser. With still further increase in unit size, two separate floor levels were adopted and so was evolved what is called the condenser basement. The turbine was mounted on the engine-room floor, from which the operation could be performed, while access to the condenser was readily obtained from the basement. The condenser auxiliaries had now become too cumbrous to be driven from the main turbine and were accommodated as independent units in the basement.

This arrangement with the condenser vertically below the turbine became the generally accepted standard for small as well as for large units. In the course of time, however, as developments progressed, the conditions became favorable for a revival of the self-contained unit. A number of factors contributed to this: the dimensions of the turbine and alternator in relation to the output had become very much smaller, condensers had become much more compact, the steam ejector had been introduced and had replaced the bulky reciprocating air pump, centrifugal pumps had become much simpler and more reliable, and mechanical gearing had been developed to a high pitch of refinement. These conditions rendered possible a closing up of the complete equipment into a compact and simplified unit, occupying much less space and requiring much less foundation than the older type. Development of various forms of the self-contained type of plant has proceeded rapidly and it has now become general for units up to about 3000 kw capacity, such as are largely used in industrial plants in England. A considerable number are also being exported for the supply of power to small towns and sparsely populated districts in many other countries, where the saving in cost of buildings and foundations and the ease of transport make them very attractive.

Fig. 23 shows a typical arrangement of a 1500-kw uniplane set manufactured by the Parsons Company, and, as the name implies, the unit is arranged in one horizontal plane. This arrangement has been developed with a view to obtaining ease of erection and maintenance, and simplicity of operation to the greatest possible degree, and has proved extremely successful in a large number of installations. The turbine, which is of the impulse-reaction type, is solidly coupled to a helical creep-cut pinion and so designed that the end pressure of the turbine spindle is toward the turbine exhaust and balances as nearly as possible the end thrust from the gear. A pivoted pad thrust block is fitted to carry the small end thrust that remains and to maintain the axial alignment. The gear wheel is also located by a pivoted pad thrust and is solidly coupled to the alternator rotor, which is of the salient, pole type.

The exhaust portion of the turbine cylinder, the bottom half of the gear case, and the bedplate are made in one casting so that the turbine and gear alignment are not affected by movement of the foundations, the design of the combined casting being sufficiently rigid to withstand distortion. The bedplate also forms the oil tank, and the oil cooler is accommodated within it. The high-pressure portion of the cylinder is a single steel casting, which incorporates the nozzle box and high-pressure blading and in addition the bearing, gland, governor, and oil-pump housings. There is no separate steam chest, but the combined runaway and stop valve and the oil-relay-operated governor valves are mounted in the nozzle box itself. The arrangement of this part is shown in Fig. 24.

The circulating and extraction pumps are driven by a worm gear from the end of the pinion shaft, the arrangement being such that the turbine, gearing, condenser, and all auxiliaries form one complete prime-mover unit, and are quite independent of the apparatus that the turbine has to drive. This may be an alternator or d-c generator, a pump, blower or rope pulley for a mill drive, according to the requirements.

Geared units are adopted for outputs up to 2000 kw, for straight condensing machines, and up to approximately 2500 kw, for

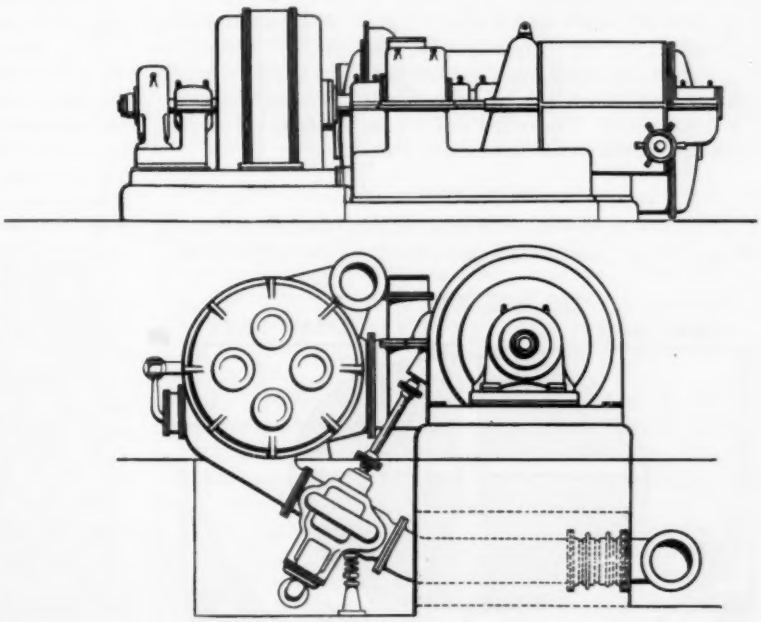


Fig. 23 1500-Kw Uniplane Set

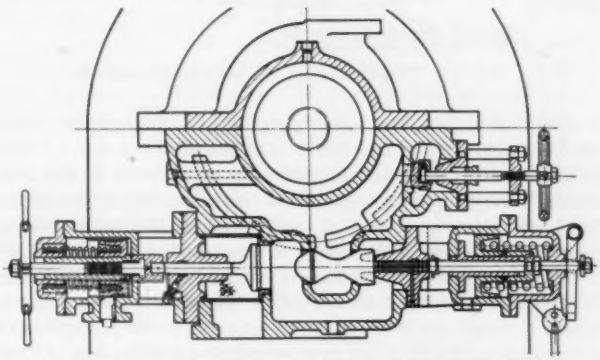


Fig. 24 Arrangement of Uniplane Control Gear

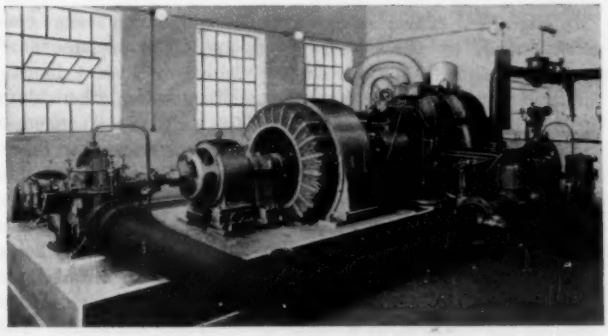


Fig. 25 1000-Kw Self-Contained Unit (Metropolitan Vickers Company)

pass-off machines, although the latter naturally vary somewhat with the steam conditions. Above these outputs the units are direct-coupled for either 3000 or 3600 rpm, and in the case of direct-coupled machines the auxiliaries may be driven either by an extension of the exciter shaft or by gearing from the turbine shaft in a manner somewhat similar to that already referred to.

Somewhat different schemes are adopted by other manufacturers. In Fig. 25 is shown a unit manufactured by the Metropolitan Vickers Company. This figure is taken from a photograph of a 1000-kw machine and is typical of the arrangement standardized by the company. In this case the turbine is supported on the condenser, its center line being at such a height that the gear pinion can be conveniently arranged above the slow-speed wheel. The exciter and circulating pump are coupled in tandem on the end of the generator shaft.

Condensers

The growth in the size of generating units and the persistent demand for more economical generation have led to the advance

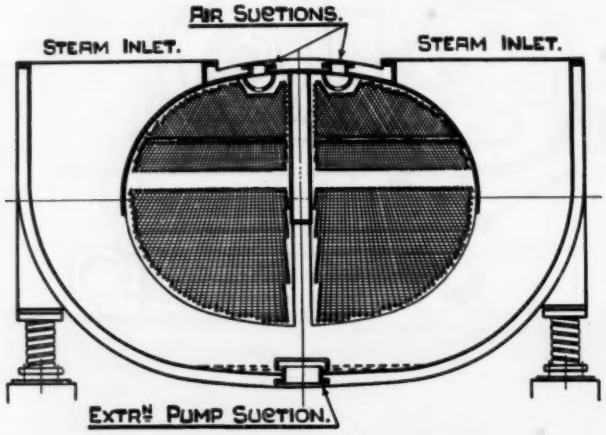


Fig. 26 Double-Inlet Condenser (Parsons)

in steam conditions, and to the progress of turbine design which has been reviewed in the earlier sections of the paper. These same factors have caused attention to be directed to the condenser, a very important section of the plant, which previously had been rather neglected, at least so far as the application of scientific principles to the design is concerned. A large number of new and revolutionary designs have been introduced during the last few years that seek, first, to improve the steam distribution through the tubes so as to make a greater proportion of the cooling surface effective and, second, to avoid the loss of heat that results from cooling the condensate below the vacuum temperature.

In Fig. 26 is shown an arrangement adopted in recent condensers made by the Parsons Company. The example illustrated is designed for a double-flow turbine with two exhausts; the tube nest is separated from the steam space on either side of the shell by a baffle, which extends the full width of the condenser in the upper portion. The steam entering at the top has therefore to pass straight down to the lower part of the shell and in so doing is spread out laterally, so that when it enters the tube nest it is distributed over the full length of the tubes. The flow over the tubes is in an upward direction, the air suctions being right at the top. The air-extraction ports are arranged so as to produce a uniform suction across the condenser, and to maintain the distribution of steam or vapor over the tubes. The condensate falls to the bottom against the steam flow, so that its temperature at extraction is not reduced below the vacuum temperature. On the circulating-water side the condenser is designed for two flows, the first flow being of course at the top and the second at the bottom.

An incidental feature of this form of condenser is the protection of the top row of tubes from the direct impact of water drops thrown off at high velocity from the tips of the last row of revolving blades in the turbine. Tubes punctured by the erosive action of these drops have been a frequent source of trouble, so much so that some power-station engineers have made a regular practice of plugging or fitting dummy tubes in the top row.

The popularity of the single-flow condenser in America has been the subject of some comment in England, where two-flow condensers have been almost invariably employed. This would, however, seem to be due to a difference of conditions rather than to a radical divergence of practice. At all events, in the cases which the author has had occasion to consider, the conclusion has always been in favor of the two-flow arrangement. Each type may have several points in its favor, and a small difference in conditions may influence the choice between them.

Generally speaking, a single-flow design means a smaller surface and longer tubes, compared with a two-flow for the same duty. For condensers of very large capacity these are advantages, as they result in a better proportioned and more easily accommodated shell. For small units, however, long tubes are undesirable because it is not usually convenient to provide the clear space necessary for their withdrawal, and also if the condenser is close to the turbine exhaust the steam will have to spread too suddenly for the tubes to be properly covered.

The single-flow condenser, with its relatively short path and large area through the tubes, requires a much greater quantity of circulating water than a two-flow to give a reasonable water velocity. This is an important consideration where, as is frequently the case, a long pipe line is unavoidable. Although the power absorbed in pumping the water through the condenser itself is not increased, a much greater loss in the pipe line has to be faced or else the expense of larger pipes and sluice valves has to be incurred.

The design of single-flow condensers does present one difficulty not encountered in the multi-flow type. The temperature of the cooling water rises continuously in its passage from the inlet to the outlet end of the tube nest. Hence, the condensing capacity is greater at one end than at the other, and the natural distribution of steam is far from uniform. To provide adequate steadiment for the long tubes of a single-flow condenser, it is necessary to employ several intermediate supporting plates, and the effect of these is to sectionalize the tube nest and to interfere with the free distribution of steam. Further, the non-condensable gases from the hotter end will not be so effectively cooled, unless some special arrangement is adopted to make them traverse a section of the tubes in the cold part before reaching the air-pump suction.

The general problem of tube corrosion still remains a perplexing one, but definite progress has been made in the production of materials more resistive to its ravages. Cupro-nickel with a nickel content of not less than 30 per cent has been used with very successful results. Very promising experience, however, has been obtained also with the aluminum-bronze alloys more recently introduced, and these are likely to be more generally used than cupro-nickel. The difference in the initial cost of the two materials is so great that, even if the cupro-nickel tubes should prove to have an appreciably longer life, it may still be more economical to employ the aluminum bronze. It is only on account of the high initial cost of the nickel alloy that it has not been generally adopted.

The most usual practice for tube fixing in large condensers is to expand the tubes at one end and to retain the packed ferrule at the other. Generally, expansion is adopted at the inlet end, as it

gives a rather smoother bell-mouthed inlet than the ordinary ferrule. In two-flow condensers, therefore, half the tubes are expanded into one tube plate and half into the other. The practice of expanding the tubes at both ends has not been used in England. The principal argument used against it is the possibility of extensive damage to the condenser should the unit, through some accident, go over to atmosphere. Although such an occurrence may be extremely improbable, the risk of it cannot be altogether ignored, and the use of automatic devices to shut down the machine on a falling vacuum is undesirable, because it may cause an interruption in the continuity of supply.

The use of steel plate instead of cast iron for condenser shells was introduced some ten years ago and soon became generally accepted practice. A riveted construction was at first employed for these steel shells, but more recently riveting has in its turn been replaced by welding, and welded shells are now used for all sizes of condensers.

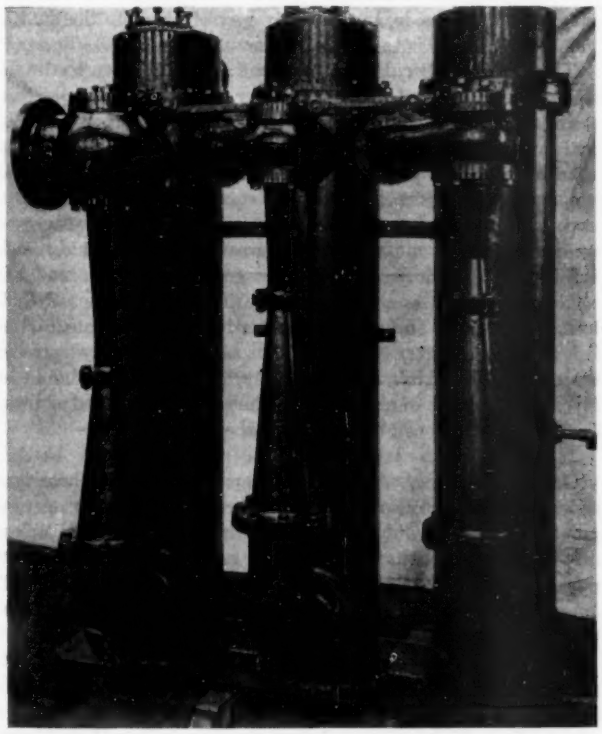


FIG. 27 THREE-STAGE EJECTOR, WITH FABRICATED STEEL SHELL

The great strides made in welding practice and the general tendency to introduce fabricated steel in all branches of power-plant work have led also to the use of steel for water boxes and end doors, particularly for cases where the circulating water is fresh. For salt water it is necessary on account of corrosion to use much thicker plate than otherwise would be required; consequently, the fabricated construction does not possess quite such a definite advantage over the casting.

Another interesting application of fabricated construction is furnished by its use for the condenser elements of air ejectors. An illustration of a three-stage ejector of this type is given in Fig. 27. The three condensers have cylindrical steel-plate shells built up as one unit, but each condenser has a separate water box and tube element. The entrainment chambers and diffuser tubes of the ejectors are supported on facings on the condenser shells. It has become usual to employ three-stage ejectors on the larger plants in preference to two-stage, because with a high vacuum an appreciable saving in the ejector-steam consumption can be effected in this way. For a vacuum of 29 in. the saving amounts to 30 per cent on the consumption of this auxiliary, one of the many small items that assist toward the attainment of a good overall economy of the plant.

Discussion

A. G. CHRISTIE.[3] The Society has had unusually good fortune in securing this paper from so well known a British engineer and particularly so in view of his position in the pioneer firm of British turbine builders. His comments upon current practice have particular weight in view of the long and varied experiences of his firm, C. A. Parsons & Co. Comments on this paper must therefore express variations in practice rather than differences in principles.

The author considers the development of the Grid System in England under the direction of the Electricity Commission. The writer's observations are that the British people as a whole are not as electrically minded as the Canadians. When the use of electricity becomes more general, the consumption per capita will rise rapidly, which will necessitate the construction of large power stations in Britain comparable with the largest in the United States.

A visitor to England is struck with the large number of firms building steam turbines for power-plant use. There are at least seven building the larger units. In the United States there are only three firms building large turbines, and from 1929 to 1931 one of these firms sold about two-thirds of the total output.

An outstanding characteristic of British practice has been the development of tandem units of large capacity at 3000 rpm. The largest tandem unit at 3600 rpm in America is 15,000 kw. The British, however, have several 30,000-kw units at 3000 rpm and can build up to 50,000-kw units at this speed. American engineers have favored single-cylinder units at 1800 rpm, although some efficiency may be sacrificed by their use.

In the matter of initial steam pressures, American practice favors 1200 lb in place of 600 lb for reheating plants. In fact, some engineers hold that the 1200-lb plant can be built for no more money than the 600-lb station. Through the data in a recent N.E.L.A. Report on Higher Steam Pressures and Temperatures it is shown that the average performance of 1200-lb stations over a range of capacity factors is about 8 per cent better than the 600-lb plant at the same capacity factors. Mr. Gardner only allows 5 per cent for this difference. British engineers have in some cases used higher initial steam temperatures than employed in the United States. However, the value of high steam temperature is now recognized here, and new stations will undoubtedly use steam temperatures of 850 to 900 F.

American engineers are keenly interested in all questions dealing with creep in metals. The figures of allowances quoted by Mr. Gardner are in general in close accord with American practice. We are experiencing similar difficulties with bolting materials.

While reheating is an element of many American stations, it is now thought that practice may be altered in new plants. Our engineers favor straight flow to reheating units. Increased steam temperatures will allow of the use of higher pressures without reheating and with moisture at exhaust within safe limits.

One is tempted to ask whether any station can be considered a base-load station. The late Col. P. Junkersfeld held that every new unit was a base-load unit, but only until the next unit supplanted it. Recent data indicate a rapid decrease in capacity factors of most turbines after the sixth year of operation, even in those considered originally as base-load units, and comparatively

[3] Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

low capacity factors after the tenth year. The average capacity factor for the whole useful life of a turbine appears to be much lower than generally assumed. Therefore one may rightly question whether any turbine should be purchased to serve as a base-load unit.

Steam reheating only effects a thermal gain of about 2 per cent over no reheating. However, it improves the moisture content at exhaust, and this is a practical benefit of considerable value. As many as five bleeder heaters have been used on American turbines. Present practice favors the use of three or four combined with steaming economizers. This combination leads to lower cost plant of high economy. Deaeration in special open-type heaters just before the boiler-feed pump has been found preferable in American practice to deaeration in the condenser as favored by Mr. Gardner.

Erosion of the last rows of blades has now become a major problem in American stations and is probably assuming serious proportions abroad. Mr. Gardner's researches on erosion form an outstanding contribution to our knowledge of the subject. A student of the writer made a special study of this subject for the N.E.L.A. last winter and developed some interesting information. The per cent of moisture at the last row of American turbines may be as high as 14 per cent, whereas abroad this is often limited to 10 per cent. The greatest difficulty is experienced with blading with tip speeds of 1000 fps or over. Parsons blading with relatively lower steam velocities has been more subject to erosion than impulse blading, suggesting that supersaturation was less pronounced in the Parsons than in the impulse blades. Devices to drain off moisture have been only partly effective due to the nature of the formation and the path of the water droplets. Other experimental work at Johns Hopkins University indicates that no separating device can be very effective. Comparatively little information is available in America on the development of erosion-resisting steels, and Mr. Gardner's data regarding the performance of hardened tool steels are particularly welcome.

The writer always has been a strong advocate of water-sealed glands. Early experiences with foggy engine rooms and dripping roofs in Northern plants during winter weather was sufficient warrant for the use of water-sealed glands to prevent any loss of steam into the operating room.

British practice favors multi-cylinder units. In America the higher costs of building and equipment have led engineers to select single-cylinder units. One regrets that Mr. Gardner has not included some performance figures on the multi-cylinder type that might be compared with American results.

Mr. Gardner marvels at the use of single-flow condensers in America. These are only purchased when it is evident that their use will lead to the highest over-all economy. American practice is tending toward condensers with tubes rolled at both ends. These are proving cheaper, tighter, and more reliable than packed tubes. When tubes are properly bowed and supported, no difficulty will be experienced should the condenser lose its vacuum.

Mr. Gardner's presentation of British practice leads one to question many of the currently accepted methods in America. A study of these conditions and the adoption of features of British practice will lead to the construction of better and more reliable American units.

M. H. Adams.[3] In both papers on modern turbine design there are one or two points, from the view of one who has to operate and maintain these machines, that the writer would like to raise.

Turbine designers have not given enough consideration to the design from the operator's standpoint. A chief complaint with the modern turbine is that the most important parts of the machine cannot be examined without considerable time, trouble, and expense. In the old days, with smaller units, dismantling for examination was not a very serious matter, but it becomes increasingly difficult as the size of the machine increases, and today the lifting of the top half of the turbine for examination is a laborious and costly business. In the event of any slight noise occurring in the machine or a sudden temporary roughness in running, one has to decide whether it is necessary to open up and put the plant out of commission or whether to run the risk of something far more serious occurring.

There should be no great difficulty in arranging suitable inspection doors or plugs through which the whole of the blading or at least the banding could be examined. Any extra cost involved would be gladly paid by generating engineers, as this cost would probably be saved many times over during the life of the machine. The design of most machines could be cleaned up very considerably from the outside, but the only way of bringing this matter home to designers is to show them actual examples.

There does not seem to be sufficient attention given to the design of the oiling system. In a number of designs it is impossible, owing to pockets, to drain out the whole of the oil and sludge, and the arrangements made for allowing settlement during operation are generally poor. The chief fault lies, however, in that no adequate provision is made for the collection of sludge. Our experience shows that the centrifuge, while removing water and solid matter, does not remove any great quantity of sludge. It is found that the only satisfactory way of collecting the sludge is in the colder part of the oil coolers. The sludge generally collects on these cold surfaces in comparatively large quantities, but it is a difficult matter to clean coolers satisfactorily as at present designed. The writer feels, therefore, that attention should be given to the design of oil coolers, so that they can be used as traps for the sludge and can be speedily and easily cleaned.

Gerald Stoney.[4] These two papers show that there is a general tendency toward increased steam temperatures up to a point where creep and embrittlement may cause trouble unless materials are used that are not only more expensive and often more difficult to work, but also about which comparatively little is known as to their properties when subjected to high temperatures for long periods.

Even as it is, nine-tenths of turbine troubles are caused by expansion, etc., due to temperature, and a point often ignored is that with high temperatures increased clearances have to be allowed for safe running, and thus the gain due to these higher temperatures is to some extent discounted. The usual rules for the gains due to temperature are with the same clearances. Even if the original clearances are fine, the turbine will in time wear itself to larger clearances with high temperature than with low.

Pressure, per se, does not cause such troubles; for example, in hydraulic machinery, where very high pressures are common, there is little trouble.

With moderate temperature, say 750 F, and high pressures, the wetness in the exhaust is large. This can be avoided by reheating; but even without this, by draining and protecting the last rows of blades by hard shields of high-speed tool steel, as described by Mr. Gardner, the trouble due to erosion by water can be completely avoided, and there is no reason, apart from the losses due to wetness, for wetnesses of 20 per cent or even more not being allowed. Many consider that the extra cost and complication of reheating is not worth the extra economy of fuel, and it has always to be remembered that simplicity, reliability, and ease of

[3] Engineer, Generating Department, Lancashire Electric Power Company, Stoneclough, England.

[4] Newcastle-on-Tyne, England.

working count for a great deal and that on an average coal is only 16 per cent of the cost of generation while capital charges amount to 45 per cent. (See *World Power*, January, 1930, p. 43.)

In this regard the advantage of large overloads for turbines has to be considered, such as the Hell Gate turbine of 160,000 kw is able to do. Here down to half or even third load there is good economy. The advantage of such large overloads is seen when it is remembered that in a typical case with a 40 per cent load factor over the year, a load of more than 55 to 60 per cent of the peak load lasts only for about 10 per cent of the total time and accounts only for some 6 to 7 per cent of the whole output. By using turbines with large overload capacities, important capital savings can be made with a very small increase in the coal bill. This was fully dealt with in a paper by Mr. I. V. Robinson at the World Power Conference, Tokyo, 1929, in one by the late Sir Charles Parsons and Mr. R. Dowson at the similar conference in Berlin, 1930, and in an article by the writer in *Engineering* of February 13, 1931.

In this country 50 periods is general, requiring 3000 or 1500 rpm, and as the Parsons constant, $K = NR^2D^2 \times 10^{-9}$, is the same for both, it follows that, representing the volume of the blading by ND^3, the volume is $^1/_4$ for 3000 rpm as compared with that required for 1500 rpm, meaning a large saving in turbine costs and weights. Sizes of 40,000 kw at 3000 rpm can now easily be made, and it is probable that this limit will increase in the future, while in this country sizes of from 50,000 to 75,000 kw are being adopted at 1500 rpm. It is a question whether it would not be better to have more turbines at 3000 rpm rather than somewhat fewer at 1500 rpm, considering the probable lower cost due to the reduced weights. Also the lighter parts make them more easily handled and more quickly overhauled, and the time for running up and getting on load is less. A point to be remembered also is that it is much easier to get reliable forgings and castings of moderate size than very large ones.

H. M. Martin.[5] The paper by Messrs. Alden and Balcke constitutes a remarkable synopsis of present power-station practice in America, and must have involved much labor in the collection of data and much thought in arranging it so clearly and concisely. The only point to which one might be inclined to take exception is the assertion that the over-all efficiency ratio has been found to serve effectively as a basis for comparison of turbines operating under varying conditions, and as a matter of fact this claim is largely discounted a few lines later, where it is emphasized that a direct comparison cannot thus be made between two machines operating under widely different conditions. In times past, the efficiency ratio had the merit of being easily computed, but this, its sole advantage, has largely disappeared with the introduction of progressive feed heating and of reheated steam. As one result, estimates of thermodynamic efficiencies have been apt to vary with the fancy of the individual engineer. M. Darrieus, an able French engineer, has accordingly suggested a modified method of computing thermodynamic efficiencies. This consists essentially in debiting the turbine with the theoretical available heat of all steam entering it, and crediting it with the available heat of all steam withdrawn from it. The balance, divided by the difference between the total heat entering and leaving the turbine (i.e., the indicated work), is the Darrieus efficiency. This procedure leads to perfectly definite results, but it is difficult to see any utility in the figures obtained. The Darrieus efficiency does not represent either the thermal efficiency of a turbine or its efficiency as a "hydraulic engine."

The discussion of blade erosion in Mr. Gardner's paper is illuminative. In 1916 it was officially stated that blade erosion was unknown in the British Navy, owing no doubt to the low steam speeds then ruling. The importance of speed was clearly shown at quite an early date, when the ordinary brass alloy used for reaction blading was adopted for the blades of a small-velocity compounded turbine. The blades were destroyed in a fortnight's run. In these early days, however, the steam commonly had solids in suspension owing to the use of hard feedwater. It was not clear, therefore, whether the damage was due to this suspended matter or to droplets of water. It is now evident that the latter constitute the real destructive agent.

It is of interest to note that in the absence of any cushioning effect of the steam in which the droplets are suspended, the stress produced by impact on a blade is independent of the size of the drops. The free distortion of the latter is resisted by the viscosity of the liquid, and the rate of distortion is independent of the size of the drop. Similar distortions are proportional to the size of the drop, and hence (at a given speed before impact) it takes longer to distort to similar shapes a large drop and a small one. The stresses are all proportional to the rate of distortion, and this rate is the same for all sizes of drop. In practice the cushioning effect of the steam is very important, and by Stokes's law will be inversely proportional to the square of the drop diameter. Actually, some data, for which the writer is indebted to Mr. F. Samuelson, of the British-Thomson-Houston Company, Rugby, show that the effective drop diameter must have been at least $^1/_{16}$ in. in the case of an experiment made by inserting, in a ring of blades, four, equally spaced, which stood $^1/_{16}$ in. above the general level. It was found that the whole of the erosion was concentrated on these four blades.

Mr. Gardner remarks that the "yield point" does not measure the resistance of a material to erosion. The explanation tendered seems, however, inadequate. If pits are present, stresses will be increased, as he states, but the question remains as to how such pits originate. It seems unlikely that a pit of molecular dimensions could be effective.

It would be of great interest and of considerable theoretical importance to obtain really accurate measurements of turbine efficiencies with and without stage-by-stage drainage. In the one case, for which the writer has data, the efficiency seemed quite unaffected, but the precision of the test was insufficient to justify a definite conclusion.

W. F. Ryan.[6] The main objective of the Power Division in regard to the papers by Gardner and by Alden and Balcke, presented at the Bigwin meeting, was to bring out and to *explain* the differences in trends of design in England and in the United States. The papers and the discussion developed the differences adequately, but there was almost nothing to explain these differences.

An editorial in *Engineering* (London) for September 16, 1932, entitled "Steam-Turbine Problems," furnishes a most valuable discussion of these papers. The first three paragraphs of this editorial are as follows:

> American engineers have been more tolerant of high leaving losses in their steam turbines than has generally been considered good practice in this country. An interesting, if partial, explanation of this is to be found in the comprehensive discussion of present-day American practice given in the paper by Messrs. Alden and Balcke, which was reprinted in our issues of July 29 and August 5 and 12. It was there pointed out that a leaving loss of 5 per cent to 6 per cent at full load, under American winter conditions, may correspond to an average loss, taking the year round, of about $1^1/_2$ per cent to 2 per cent. The high summer temperature offsets the winter's cold so that apparently the vacuum averaged over the year does not

[5] *Engineering*, London, England.

[6] Mechanical Engineer, Stone & Webster Corporation, Boston, Mass. Mem. A.S.M.E. Member of Executive Committee of Power Division.

exceed some 28¼ in. Here, climatic conditions are much less extreme, and though we seldom can get circulating water at 35 F, which is customary in Chicago during the winter months, summer temperatures, at once high and sustained, are equally unusual. Even with our mild winters, however, stations like Brimsdown secure an average vacuum during that season of 29.1 in., the average for the twelve months ending on December 31, 1931, being 28.86 in. The still higher annual average of 28.92 in. was, Mr. Donaldson informs us, recorded during the previous year. The average air temperature during the months October to March, both inclusive, is about 45 F in the London area, being lowest in February, when it averages about 5 deg less. The average for the other half of the year is 59 F. The warmest month is August, in which the temperature averages nearly 4½ deg more. The mean winter temperature of New York, on the other hand, is given as but 22 F, the summer mean being 72 F, and the annual average 52 F, which is only 1 deg more than the London average. American engineers have, therefore, to deal with much more extreme conditions.

At the same time there is, we think, reason for believing that we still maintain our lead in the design of surface condensers, which was secured by the early work of Sir Charles Parsons, and has been promoted since by various practical and theoretical investigations. It was here, apparently, that the plan of dividing up the various resistances to heat flow was first introduced, and this has certainly led to clearer conceptions of the phenomena involved. As indicating the hazy ideas previously prevalent, it may be noted that Stanton, as the result of his own observations on actual condensers, came to the conclusion that the steam pressure was a highly important factor. As matters now stand, stations using cooling towers record vacua of about 28½ in. with circulating water supplied at 75 F.

Messrs. Alden and Balcke state that single-pass condensers are most commonly used in the States. This is, perhaps, partly due to the large size of the units generally installed. With small outputs, several passes may be required to get the required combination of a good water speed and adequate cooling surface. Thus the small condenser described in a paper read before the Junior Institution of Engineers by Mr. J. M. Newton, in 1912, had six passes. It was coupled up to a Brush turbo-generator rated at 300 kw. A vacuum of 28.8 in. was obtained with a circulating water speed of only 3.4 ft per sec and an inlet temperature of 48 F, rather over 8 lb of steam being condensed per square foot of cooling surface.

Author's Closure

Professor Christie draws attention to the fact that in Great Britain the consumption of electricity per capita is still relatively low, and it is encouraging to hear the view that a rapid expansion may be expected. Nevertheless, there are in England, as he points out, several large firms that build generating plants, and there seems little prospect that the requirements for the home market will be sufficient to keep the factories of all these firms employed.

Regarding the discussion on steam pressures, there is little doubt that, as conditions exist today in England, a 1200-lb installation would cost more than a 600-lb one, but it is safe to assume that costs will tend to level up. In the past, every advance in steam conditions has at first been hampered by its higher initial cost, but this is a temporary phase, and as the use of the higher conditions has become more general, the cost of the installation has gradually adjusted itself to an economic level. Professor Christie's remark that "Some engineers hold that the 1200-lb plant can be built as cheaply as the 600-lb" suggests that in the States this condition has already been reached.

In comparing the difference in economy of the 1200-lb and 600-lb installations, the figure of 5 per cent, given in the paper, represented the theoretical difference in heat consumption of the turbines. Since the paper was written, the author has seen the operating data published by the N.E.L.A., which indicate a difference in overall performance of 8 per cent. It would be valuable to have a complete analysis of the comparative figures showing how this difference of performance is made up—whether it is all in the turbines or whether it is partly accounted for by other factors, such, for example, as improved boiler efficiency.

Dealing with the problem of blade erosion, Professor Christie notes that reaction blading has been more subject to erosion than impulse blading. This is in accordance with what might be expected from an examination of the velocity triangles for the two types. In the case of the reaction blade, the velocity of impact with the water drop is greater for a given blade speed. The same factors which make an impulse blade less liable to erosion, however, also make it more difficult to protect an impulse blade by the method of shielding described in the paper.

Mr. Martin raises the question as to how pits in the surface of a blade originate in the initial stages of erosion. An examination of the surface of any commercial blades under a reasonable magnification will show that there is no shortage of irregularities of sufficient dimensions to start the action, even though, as appears to the eye, the blade has a polished surface. It is not practicable to produce commercial blading in which the irregularities are reduced to anything approaching molecular dimensions. Even if there were no such irregularities present in the first instance, they would soon be produced in service by the sand-blast action of minute solid particles in the steam or by surface oxidation.

With regard to the size of drops, the author's calculations indicate that in the exhaust of a high-speed reaction turbine the largest drop that can persist in the steam stream is only a few thousandths of an inch in diameter. (Strictly this applies only to the drops that leave the blade edge with little or no velocity, but it is these drops that produce erosion.) Owing to the higher steam velocity in a pure impulse turbine, the drops should be considerably smaller than those in the case of reaction.

Mr. Adams' remarks, in which he draws attention to features of design that should be improved, are particularly welcome, as real progress in design can be achieved only by eliminating the defects that the experience of the operating engineer brings to light. With reference to the provision of means of sludge collection and removal from the lubricating system, the arrangement adopted some years ago by the Parsons Company is worth mentioning, for it has proved very successful. The main oil reservoir, which is of sufficient capacity to allow of settlement, is made with a sloping bottom and terminates in a sludge trap consisting of a vertical length of straight pipe. A valve is provided at each end of this pipe, the upper one being normally full open and the lower one closed. The angle of the tank bottom is too steep for solids to rest on it, and they therefore slide down and accumulate in the pipe. The sludge, together with any water present, can be drained off easily at appropriate intervals by closing the upper valve and opening the other. A centrifugal or other type of continuous oil filter is of course provided in addition. This draws oil from the body of the tank, and not from the sludge-collecting chamber.

In most lubrication systems the oil coolers are located at the lowest point, and this may account to some extent for the accumulation of sludge in them. Some form of sludge trap, such as described, could be incorporated with the coolers.

Dr. Stoney questions the wisdom of adopting higher steam temperatures. It is of course true that a large percentage of turbine troubles is due to temperature effects. Advances in temperature have in the past always been attended by new or aggravated expansion troubles, and sometimes by quite unforeseen difficulties. For example, at one period much trouble resulted from growth in cast iron, a phenomenon which had not previously been understood. Nevertheless, in each case the difficulties have been overcome and the advantages of the development have eventually been realized, and this affords the justification for further advances. For the time being, however, temperature limits seem to be rather definitely settled by the known physical properties of the materials available.

FSP-55-4

Steam and Power in the Bleached-Sulphite-Pulp Mill

By W. R. GIBSON,[1] PORT ANGELES, WASH.

Methods of effecting economies in the use of steam in the paper industry are given, together with possibilities of water saving. After a brief description of the processes in a sulphite mill, a table is given of the probable requirements in steam and power for a 150-ton mill. The possibilities of combining the operations of a pulp mill and a sawmill are stated, the sawmill having a surplus of waste wood for fuel and of low-grade wood for pulping and usually exhausting large quantities of steam to atmosphere. The effect on refractories of hog-fuel ash is related.



MAKING bleached sulphite pulp involves quite a few operations. A suitable wood, hemlock, spruce, or alder, alone or in combination, is cut into chips 7/8 to 1 in. long, 1 in. wide, and 1/8 in. thick. These chips are cut off in such a way as to open the end grain to aid penetration by the acid in the cooking operation. The chips, screened and free of dust, are cooked in sulphurous acid in a digester 8 to 11 hr at about 70 lb pressure to free the fibers of the ligneous binder. The cooking being completed, the fibers are blown from the bottom of the digester to a blow pit with a vomit stack, the vapors being released through the stack and the spent cooking liquor being drained away through a perforated floor. The pulp is then washed, and is pumped to a mixing tank, where several "cooks" are combined to secure uniformity. From the mixing tank the pulp is passed over (1) knotter screens to remove knots and uncooked particles of wood, (2) rifflers, to remove grit, sand, and heavy impurities, and (3) flat screens, or vibrating plates that have slots from 0.006 to 0.011 in. wide to exclude everything but fibers. A great amount of water is used in all these operations.

The pulp is now passed over deckers or stock thickeners, where the surplus water is removed, and the thickened stock then goes through two or three stages of bleaching, the stock being washed and again thickened after each stage. With the bleaching finished, the pulp is pumped to stock-storage chests. If the pulp is to be shipped, it is run over a machine where the water is removed by (1) gravity, (2) pressure between rollers, (3) suction, and (4) passing over steam-heated rolls. When dried, the pulp is cut into sheets and is baled for shipment.

This brief statement will give some idea of the need for the large amounts of water and steam and power required in a sulphite mill. Assuming a mill of 150 tons' daily capacity, the quantities should be about as given in Table 1.

TABLE 1 STEAM AND POWER USED IN 150-TON SULPHITE MILL

	Steam usage		Power usage
	Old mill, lb per ton	Modern mill, lb per ton	modern mill, kwhr per ton
Acid plant	300	300	2
Digesters	7,000	5,500	...
Unbleached stock	...	...	100
Bleached stock	3,000	1,500	110
Machine drying	7,000	3,800	80
Heating, etc	500	500	15
Total	17,800	11,600	307

The difference in steam consumption between the old mill and the modern mill is accounted for by the improved methods of cooking, the recovery of hot gases from cooking, the increased density of the stock before bleaching, and the improved mechanical methods of removing water from the pulp before passing it over the heated rolls that complete the drying. A recent development is the vacuum drier, with the steam heated rolls contained in a box in which a vacuum of 27 to 28 in. of mercury is maintained, the sheet of pulp entering and leaving the box through ingeniously designed seal rolls, with a resultant saving in steam consumption. Owing to the lower drying temperature, this results in an improved and stronger pulp.

The water required for such a mill would be 15,000,000 gal per day, or 100,000 gal per ton.

The power requirements, as shown, are good average figures for today. There has not been much change in power requirements, the improved efficiency in the use of power having been offset to a great extent by other changes that, while effecting economy or improving the product, called for a further use of power.

The large demand for steam for processing and for power offers an attractive field for the use of high-pressure steam with back-pressure or bleeder turbines or engines, and the changes in this respect have been rapid in the last 10 years.

An interesting possibility of this field is the combining of a sawmill and a pulp mill. The sawmill has a surplus of fuel and of low-grade wood and usually exhausts large quantities of steam to atmosphere. By combining operations, the pulp mill can use in its chipping plant much of the waste from the outside of the log, which makes the best grade of pulp, and also the lower grade lumber for which the sawmill cannot find a profitable market, and can make use of some of the exhaust steam to heat water for feed or process. All waste wood not used for pulp can be used as fuel. Much progress has been made in the economical use of this hogged fuel. Table 2 gives the performance in a plant of this kind, the figures being as accurate as is reasonably possible with a fuel of this character. The figures are for waste hemlock fuel and are believed to show better performance than has been reached previously with such fuel.

The chief factors in obtaining this performance probably are the highly preheated air and the provision for admitting this preheated air as secondary air for combustion. This is admitted as close as possible to the front wall of the dutch oven, through checkerwork in the inside and the battery walls just on and above the normal fuel line. In practice, separate control of this secondary air has not been found necessary.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 2 PERFORMANCE OF COMBINED PULP MILL AND SAWMILL

Boiler readings at avg rating (max rating, 275%, avg rating, 160%):	
Draft, main breeching, in. of water	−1.4
Drop across preheater, in. of water	−0.8
Drop across last pass, in. of water	−0.35
Drop across second pass, in. of water	−0.1
Furnace draft, in. of water	−0.15
Pit pressure, in. of water	+0.5
Available induced draft, in. of water	−6.0
Flue-gas temperature, last pass, F	560
Flue-gas temperature, after preheater, F	440
CO, per cent	15.3
O_2, per cent	4.6
CO_2, per cent	0.0
Total air, per cent	128
Air temperature to preheater, F	130
Air temperature from preheater, F	410
Flue-gas temperature at stack (after washing), F	350
Evaporation:	
Oil (per barrel), lb	5,000[a]
Hog fuel (per unit of 4700 lb), hemlock, lb	12,000[a]
Avg moisture, per cent	51.5
Feed-water temperatures:	
Cold water, F	50–55
After steam killer (using sawmill exhaust), F	210
After Cochrane heater (30 lb steam), F	250
After closed heater (150 lb steam), F	345

[a] Unit of evaporation = 1000 Btu.

Perhaps the greatest difficulties in operation of such a plant at high ratings arise from the hog-fuel ash attacking the refractories, forming a slag that builds up on the boiler and the superheater tubes so as finally to bridge across the tubes. The effect on the refractories is due to a combination of factors, all tending to lower the fusion point of the firebrick. Some of these are: (1) high moisture content, (2) high alkaline content, (3) presence of tannic acid, and (4) presence of acetic acid. The high CO_2 possible with this fuel appears also to be a factor, and a most destructive action is sometimes set up by carbon monoxide due to local lack of oxygen. Cases of slag fusion forming masses 3 to 4 ft thick have been cured by the admission of secondary air at the proper point; in other cases, firebrick would disintegrate without visible evidence of heat, apparently due to pockets of CO.

There is also the occasional formation of sodium sulphate due to combination of alkaline ash from hog fuel and sulphur from fuel oil burned together or separately in the same furnace. When this happens, even high-grade firebrick will run like molasses, and the engineer may find more damage to his setting in a few hours than normally would take place in months.

An analysis of slag from different parts of a boiler burning hog fuel is given in Table 3, the fuel being salt-water-borne hemlock with oil as an auxiliary fuel.

TABLE 3 ANALYSIS OF SLAG FROM BOILER BURNING HOG FUEL

	Sample from First pass	Second pass	Last pass	Air preheater
Silica, SiO_2, %	1.6	2.2	1.6	1.3
Iron oxide and alumina, R_2O_3, %	7.7	6.4	15.9	8.9
Calcium sulphate, $CaSO_4$, %	4.9	3.2	7.1	5.8
Calcium oxide, CaO, %	11.5	8.7	6.2	8.3
Magnesium oxide, MgO, %	8.1	6.3	5.7	5.3
Sodium sulphate, Na_2SO_4, %	6.0	14.4	6.5	16.3
Sodium chloride, NaCl, %	55.1	52.7	57.8	53.6
Sodium carbonate, Na_2CO_3, %	1.6	1.6	1.3	1.3
Sodium hydrate, NaOH, %	0.3	0.3	...	...
Organic matter, %	0.3	1.3	0.3	1.1
Undetermined, %	2.9	2.9	...	...
Totals, %	100.0	100.0	102.4	101.9
Fusion temperatures:				
First sign of fusion, F	1050	880	1090	880
Complete fusion, F	1500	1480	1450	1440

The formation of slag on the boiler tubes appears to be due to a kind of cementing action by the sodium-chloride and sodium-sulphate particles on the flying ash. In some parts of the boiler the coating will be soft and fluffy and will disperse easily by blowing and even by a change of temperature through opening an inspection door. Under different conditions, however, the mass will build up quite heavily, and if the boiler rating is raised even for a short period, the mass will fuse into a hard but porous slag that is very adherent. This process is cumulative until the space between tubes is bridged.

At normal rating, using soot blowers and taking all ordinary precautions, we cannot operate more than five weeks without shutting down for cleaning; at high ratings this period is reduced to three weeks.

It should be remarked that in the case referred to all logs are sea-water-borne, and the salt content of the fuel, which is mainly the outside and the bark of the tree, is naturally high.

Hog fuel from mills bringing their logs by rail or fresh water is free from many of the characteristics referred to and offers a different problem in many ways.

Discussion

H. W. BEECHER.[2] It is difficult to add to the paper, but a few points might be brought out. The greatest difference in steam consumption of pulp mills, as reported in the paper, is in the drying of the pulp, and this is attributed in a large measure to the vacuum drier. It should be pointed out that great improvements have been made in press rolls and that these improvements may account for considerable of the difference. To illustrate the possible saving from better pressing prior to admission to the drier section of the paper machine, the stock comes off the press rolls between 27 and 32 per cent stock. The pulp leaves the driers at about 94 per cent stock. In an original pound of 28 per cent stock, there is 0.28 lb of stock; in a final pound of pulp, there is 0.94 lb of stock; so that it takes 3.36 lb of original pulp and water to produce 1 lb of pulp at the finish end of the machine, and this contains 2.425 lb of water. Therefore there has been removed 2.365 lb of water in the driers. Comparing this with a machine furnished with 31 per cent stock, the original pulp and water to the machine would be 3.025 lb per pound of finished pulp, of which 2.080 lb would be water, and the amount of water removed in the driers would be 2.02 lb. In other words, a difference of 3 per cent in the percentage of stock in the original material to the driers would bring about a saving of 17 per cent in the steam required to dry the pulp.

The boiler-performance figures in Table 2 are considerably above the average.

In the author's mill nothing but hemlock fuel is burned. It has been the impression that efficient results could not be obtained with hemlock. As a matter of fact, the heat content per pound of dry wood is as high in hemlock as in fir. Wood refuse is usually measured by volume rather than by weighing, and the weight per cubic foot of hemlock on a dry basis is considerably less than fir. If fuels were weighed rather than measured, hemlock would give results closely approximating those obtained with fir. The chief difference would be due to the hemlock as a rule running higher in moisture than fir.

The author has pointed out the difficulty in operation arising from the ash and foreign matter in the hog fuel. His mill is the first one in which an attempt has been made to secure high evaporations with hog fuel made from logs that have been in salt water. The bark absorbs considerable salt water, and the sodium chloride forms a considerable percentage of the ash that lodges in the tubes. The percentage of ash in burning wood is very small, and the sodium chloride thus becomes a high percentage of the actual weight of ash. It is difficult to account for the presence of the sulphate radical, except that around a sulphite mill the waste liquors are discharged into the bay close to the mill pond in which the logs are stored. A further possible source

[2] Manager, Charles C. Moore & Co., Seattle, Wash. Mem. A.S.M.E.

of sulphate radical is that SO_2 may be present in the air in such quantities and in such a state that it combines in the furnace. As approximately 10 lb of air is passed through the furnace per pound of dry fuel, it would not take a very high percentage of SO_2 in the air to provide the necessary sulphur for the sulphur content of the ash.

The writer does not agree that high percentage of CO_2 has any deteriorating effect on the brickwork other than the general increase in furnace temperature resulting from efficient combustion with low excess air. He does not believe that CO_2 can chemically deteriorate the aluminum and silicate oxides forming the refractories.

The slag that forms in the boiler tubes fuses at about 1100 F, and the character of the slag, as deposited throughout the boiler, has varying chemical characteristics, depending upon the character of the gases at the point of deposit. In the front part of the boiler the slag lodges on the tubes in a molten condition. In the back of the boiler, where the gases are comparatively cool, the ash is soft and fluffy and can be brushed off readily.

It might be interesting to notice that the boiler performance figures given in the paper show a combined furnace, boiler, superheater, and air-preheater efficiency of about 60 per cent. This may appear to be poor efficiency to those who are used to dealing with efficiency obtainable with oil and coal fuel, as these are very considerably higher than the average for hog fuel. The difficulty in obtaining high efficiency with hog fuel is due to the high moisture content of the fuel and the high hydrogen content of wood refuse.

FSP-55-5

Economic Balance Between Steam and Hydro Capacity

By K. McH. IRWIN[1] and JOEL D. JUSTIN,[2] PHILADELPHIA, PA.



K. M. Irwin

There is a popular fallacy that hydroelectric development and additional steam-plant generating facilities are competitive sources of power. Balancing one against the other is often discussed by engineers who are prejudiced in favor of hydro or of steam. It is the object of this paper to point out that each proposed hydro development should not be measured against an equivalent steam installation as a sole source of power, but that it is necessary to forecast the characteristics of the particular system in which additional capacity is required and to determine for the particular conditions whether the addition of steam or hydro will give the lowest total cost of power supply for the system.



Joel D. Justin

THE downward trend of steam-plant operating costs has often been cited as a reason for favoring steam as a source of power. During the past 20 years there has been a remarkable increase in steam-plant efficiencies which seem impossible or unlikely to be duplicated, although it may still be unsafe to predict what the future may offer. It is interesting to refer to a chart that was presented in the July 4, 1925, number of *The Electrical World* and taken from a paper presented before a meeting of the A.I.E.E. The footnote to this graph states: "This curve . . . represents the weighted average in connectiou with the performances plotted for 22 stations. . . . The dotted extension of the curve is our estimate as to the future possibilities in the way of improved performance for a station designed to operate on the straight steam cycle with a single stage of reheating."

Points on this graph are as follows: 1913, 22,500 Btu per kwhr; 1916 to 1921, 19,500 Btu. Then appears the sharp downward break which reached 14,500 Btu by 1925. The "dotted extension" from 1925 straightens out until in 1929 it reaches 13,000 Btu and has been extended flat at 13,000 Btu as far as 1932. In 1931, however, at least one plant with a single stage of reheat averaged 12,000 Btu for the year and was operating between 200 and 300 Btu less than 12,000 for six months. While predictions of future steam efficiency seem extremely unreliable, it is safe to say that the best which can be foreseen is approximately 10,000 Btu when utilizing steam at 2500 lb and 1000 F and 8500 Btu with the mercury cycle.

In considering steam-generated power costs considerable speculation is necessary regarding fuel costs. At the present time coal is probably being sold at prices that do not provide an adequate return on the coal-mining investment and may even yield little more than the mining cost. While it would seem reasonable to assume that this condition will not persist, the competition of other fuels will tend to hold coal prices down.

Component Parts of Steam-Operating Costs

When estimating the cost of blocks of power under system-load curves from various existing or contemplated steam plants, it is necessary to break these costs into their component parts. The commonly used division is between the fixed charges and the operating costs, considering the fixed charges as a cost per kw of capacity and the operating expenses as costs per kwhr. A very much more accurate forecast can be prepared if the operating expenses are also divided into their component parts, which might be termed "fixed component," "peak prepared," and "increment cost of energy." There are certain annual costs which accrue in a steam plant regardless of the peak load carried or the amount of energy generated. This is termed the "fixed component" and includes portions of the superintendence, labor, supplies and expenses, maintenance, and a small amount of fuel. The second component is caused by having the station prepared to supply a given load. Included in this item, besides superintendence, labor, and maintenance, must be the fuel necessary to keep the required generating capacity turning over at no-load output. This expense varies with the kw load that the plant prepares to carry. If the first two components have been properly allocated, the variable cost is directly proportional to the amount of energy generated, and therefore can be reduced to a cost per kwhr.

For preliminary economic studies, such as those discussed herein, it is feasible to use only two factors (annual capacity cost per kw and energy cost per kwhr), the first as the total of the first three items mentioned, and the second the variable. It should be emphasized that in using two factors a portion of the

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[2] Hydroelectric Engineer, United Gas Improvement Company. Mr. Justin was graduated from the College of Civil Engineering, Cornell University, in 1906. In recent years he has devoted much time to studies and investigations relating to the economics of power supply for various power companies. For the past 18 years he has been connected with the investigation, design, and construction of hydroelectric projects. He is co-author of "Hydro Electric Handbook" (Creager and Justin) and author of "Earth Dam Projects," and various articles and papers.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

operating expense is treated as constant per kw and added to the fixed charges in place of treating only the fixed charges as constant and all of the operating expenses as varying with output, and also that peak prepared for is assumed equal to installed capacity.

In Table 1 are given the cost elements of a few typical plants.

The capital cost of steam plants varies over 100 per cent. Some of the factors causing the variation are the price of materials, the foundation conditions, local building restrictions, size of installation, number of units, and the amount of advance expenditure made for future extensions. The size of the plants also has a marked effect in reducing the fixed operating cost per kw.

For a new plant to replace an old one on the load, the cost of operating plus the cost of fixed charges must be less than the operating cost of the old plant. From Table 1, therefore, the cost developed from columns 6 and 7 must be less than the cost derived from column 5 plus column 7. With relatively high load factors or long hours' use, the low increment cost of the new plant makes this replacement of new for old plants economical, but for low load factors it is often less expensive to continue to operate high operating cost plants.

TABLE 1

1	2	3	4	5	6*	7
Plant	Date built	Fuel cost, cents per million Btu	Capacity, kw	Fixed operating per kw	Annual capacity cost per kw	Variable operating, mills per kw-hr
A	1930	13.5	120,000	2.10	14.10	1.73
B	1926	15.35	130,000	4.50	16.50	2.42
C	1920–1926	18.05	135,000	4.73	16.73	2.63
D	1924–1927	17.85	215,000	4.52	16.52	3.32
E	1923	9.54	27,000	11.42	23.42	3.46
F	1917–1921	17.75	160,000	11.80	23.80	3.27
G	1911–1915	9.54	65,000	11.46	23.46	4.10
H	1917	16.05	20,000	8.15	20.15	4.60
I	1905	17.85	17,000	13.50	25.50	6.70

* The sum of column 5 and the fixed charges assumed 12 per cent on $100 per kw in every case.

Thus, the total annual cost of Plant A of Table 1 is less than the annual operating cost for Plants D, E, F, G, H, and I at 30 per cent load factor, but at 10 per cent load factor only less than Plant I. Likewise, Plant B is less than H and I at 30 per cent, but only less than I at 10 per cent load factor.

The fixed charges referred to are made up of the return on the investment, an amount set aside for replacement fund, insurance, and taxes. The first item varies from 6 to 8 per cent of the capital invested. The depreciation is variously estimated by different engineers from 3 per cent to 4 per cent on steam plants; insurance payments generally average one-half of 1 per cent. The taxes on real estate and personal property rarely exceed 1 per cent. For studies of this kind, these are the only taxes that need be considered. Some states levy gross-receipts taxes in lieu of personal-property taxes; and for general corporate purposes, these, together with income taxes, are often apportioned as percentage of investment and are included in the fixed charges. It is not correct to include such taxes in an economic study, because they are dependent on output or profit and bear no relation to investment.

In Table 1, carrying charges have been assumed as 7 per cent return on investment, $3^1/_2$ per cent for depreciation, 1 per cent for taxes, and $^1/_2$ per cent for insurance, a total of 12 per cent.

Changing Economic Function of the Hydro Plant

In the past many hydro plants were constructed as the sole source of power supply for a community or section. Under such conditions, it is necessary that the minimum flow of the river should be sufficient at time of maximum power demand to give all the power that is required to meet the load.[3] Except where seasonal regulation through natural or artificial storage was available, there resulted a tremendous wastage of water over the dams during the greater part of the year. Utilization factors of as low as 35 per cent were not uncommon in such plants. Utilization factor as used in this connection means the ratio of the annual amount of energy obtained from the plant to the annual energy that might have been obtained from the same plant with the same stream flow if the connected load had been capable of absorbing all of it.

In such plants the investment in lands, dams, and riparian rights was necessarily relatively high compared to the investment in power house and equipment because of the relatively small installation. As a result the total cost of the development was high per unit of installed capacity.

Except in the case of certain favored sections where development costs are low and natural or artificial storage is readily available, or in sections where fuel is extremely high in cost, it is no longer economic to install hydro plants as a sole source of power supply.

Most companies having hydro plants as the only source of power supply based their installation on flows somewhat higher than the minimum, because the cost of the additional installation was very low. During the greater part of the year, such companies had available large amounts of secondary power which they could not utilize on a normal public-utility load curve. A part of this secondary power was salable on a when, as, and if available basis to industries which could afford to shut down a part of their operations during periods of low water. With the development of long-distance transmission, these companies extended their transmission lines to interconnect with other companies served exclusively by steam plants, thus being able to sell their surplus hydro energy during periods of plentiful stream flow and receive in return steam-generated electric energy during periods of low stream flow.

Through a continuing process of integration, the consolidated companies now contain groups of steam and hydro plants. It has thus come about that steam and hydro, which used to be considered as highly competitive alternatives, have become complementary sources of power supply.

Firm Hydro Capacity

In any large system the power supply of which consists of both steam and hydro plants, it is always feasible and generally quite economical to supply the requirements of the increasing load through the construction of additional steam capacity. If, however, the system includes available hydro sites, it is necessary to conduct quite an extensive economic study in order to determine whether the additional load can be most economically supplied by a new steam plant or by a new hydro plant.

As steam is the yardstick by which the economic advisability of hydro must be measured, about the first things which must be determined with regard to any hydro project (after its physical features have been investigated) is whether the proposed hydro capacity can do the same work that an alternative steam plant might perform on the existing load curve. If the hydro plant can do this, its capacity is said to be "firm." The firm capacity of a hydro plant may be defined as that portion of its total capacity which can perform the same in that portion of the load curve assigned to it which alternative steam capacity could perform.

The minimum 24-hour power available at the site is sometimes incorrectly spoken of as the firm capacity. This can be true only in the case where there is absolutely no pondage. With

[3] An exception to this rule is furnished by such industries as the paper mills and aluminum industry, which can afford to utilize a certain amount of seasonal power.

large pondage and favorable load conditions, the firm capacity of a hydro plant may be many times the minimum 24-hour power available. In the case of some existing plants which are parts of large systems and are possessed of ample pondage, the firm power is ten or more times the minimum 24-hour power available.

The concept of firm power will be plain by reference to Fig. 1, which is the load curve of a large power system for the maximum-demand and maximum-energy week. In the system under consideration, the maximum demand always occurs in December. The hydro project under consideration would have a large pond of ample capacity to provide weekly regulation with an insignificant drawdown. It is apparent from Fig. 1 that the upper part of the load curve contains relatively little energy. If we measure down from the peak of the curve a distance equal to a proposed capacity for the hydro plant, the energy in that portion of the curve above a horizontal line drawn through this point may be determined by planimetering or otherwise. If the energy as determined is equal to or less than the amount of energy obtainable from the stream flow in a week of minimum December flow, then the proposed capacity will all be firm capacity, because it can perform the same function on the load curve as alternative steam capacity.

The foregoing is the most simple curve which assumes that there are no other hydro plants in the system. If other hydro plants are already in the system, the cumulative total of their firm capacity must first be laid off, measuring down from the peak of the load. The firm capacity of a proposed additional plant will then be indicated by the width of the horizontal band immediately below, which contains an amount of energy just equivalent to energy available at the proposed new hydro plant during a week of minimum December flow.

If the characteristics of the load are not already well known, it may be necessary also to study load curves for many other weeks of the year, because, although the load at other times of the year is less, the load factor may be higher and the stream flow less.

The determination of firm capacity, utilizing the actual load curves and the cut-and-try methods indicated, is very laborious, and consequently the work is greatly abbreviated by the use of mathematical curves.

Peak Percentage Curves

To prepare a mathematical curve to be used in determining firm capacities, it is first necessary to plot a load-duration curve for the period under consideration. This is done by arranging the loads hour by hour in order of magnitude and plotting, using loads as ordinates and time as abscissas. Areas underneath the curve thus represent energy. In the most convenient form of load-duration curve, per cent of total load is plotted as ordinates and per cent of time as abscissas. Such a curve may then be applied to many different years (assuming no change in shape of load curve). A load-duration curve is shown in Fig. 2.

The peak-percentage curve is obtained from the load-duration curve as follows. Beginning at the top of the curve, cumulative areas are determined by increments by planimetering or otherwise. From this data, a peak-percentage curve, such as that

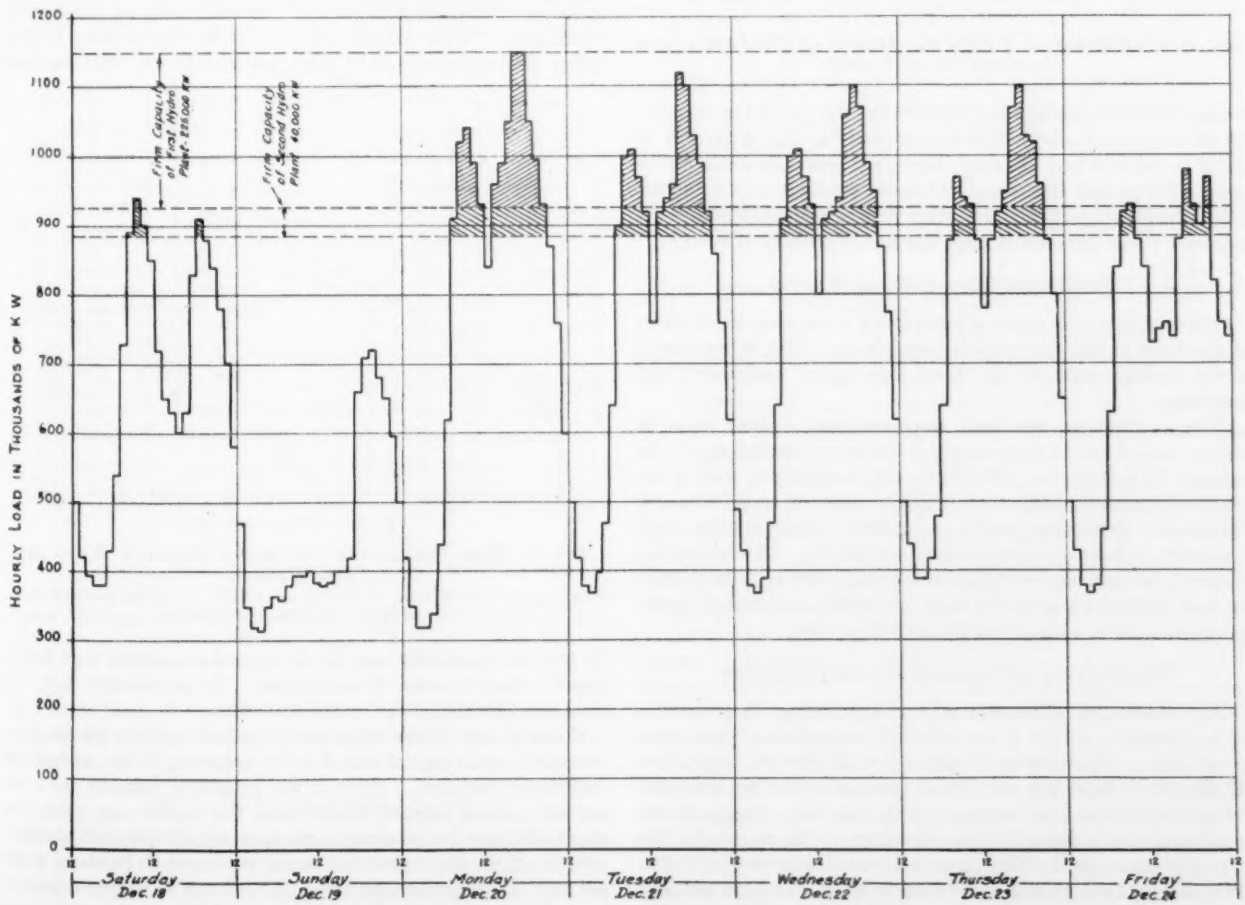


Fig. 1 Load Curve for Peak-Load Week in December, 1926, Large Power System
(Kwhr for week, 122,000,000; hydro kwhr, 5,860,000.)

shown in Fig. 3, is obtained. (Figs. 1, 2, and 3 are all for different power systems.)

In order to illustrate the use of the curve in as simple a manner as possible, it will be assumed that at the present time there is no hydro capacity in the system, but that the company is contemplating the development of a site to be completed for the 1934 load.

At the proposed site, which has ample pondage, there will be available in a week of minimum December stream flow, 900,000 kwhr, which from the table in Fig. 3 is 5.4 per cent of the total energy for the week. Using the curve, this corresponds to 24.5 per cent of the load. In other words, in the top 24.5 per cent of the load, there is only 5.4 per cent of the total energy. As the

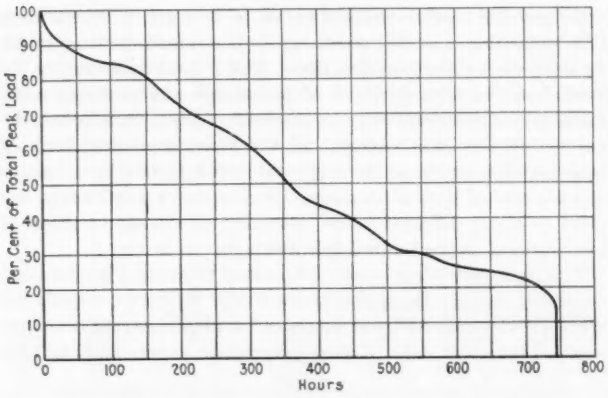


FIG. 2 LOAD-DURATION CURVE FOR MONTH OF DECEMBER OF A REGIONAL POWER SYSTEM

total estimated load in 1934 is 187,000 kw, the amount of capacity which may be installed and be firm capacity in 1934 is 187,000 × 0.245 = 46,000 kw. It may, however, prove desirable to so design the project that some additional capacity may be added at a later date, particularly if river flow is such that additional capacity would generate a large amount of additional energy.

INCREMENT COST OF HYDRO PLANTS

The capital cost of hydro plants per kw varies over much wider limits than is the case with steam plants. This is because a much smaller part of the total cost is in equipment and buildings.

For any given site and head, the investment in land, riparian rights, and dams is very nearly a constant, regardless of the amount of installation. However, the incremental cost of installation, meaning those costs which roughly are proportional to installation, consisting mainly of building, intake conduit, and equipment, varies within much narrower limits. An examination of cost data for a number of plants varying in capacity from 3000 to over 200,000 kw indicates that the incremental cost of hydro plants in general varies from $55 to $70 per kw.

SIGNIFICANCE OF INCREMENTAL COST OF HYDRO

The importance of incremental cost as affecting the economics of hydroelectric plants is not generally recognized. Thus, at a given site, a plant having a capacity of 20,000 kw might cost $7,500,000, or $375 per kw. In all probability the requirements for additional power and energy could be met more cheaply by the construction of a steam plant. However, at the same site, the capacity might be doubled at an additional cost of $1,000,000. The cost per kw installed would then be $212, and quite possibly the project might prove economic at such time as all the capacity became firm capacity on the load curve.

EFFECT OF INCREMENT CAPITAL COST ON ECONOMIC FUNCTIONS OF HYDRO PLANTS

The effect of the unusually low increment capital cost of hydro installation in determining the advisable use to be made of a hydro project may be fully understood by a study of Fig. 5. Curve *A* in this figure gives the total cost of energy per kwhr for a modern highly efficient steam plant.

Curve *C* gives the total capital cost per kw of installation for a hydro plant if installed for various annual capacity factors. The plant is a fairly typical hydro plant on a fairly typical American river, having an average annual runoff of about 1.5 cfs per square mile and a range of flow varying from a weekly minimum of 0.15 cfs per square mile up to over 28 cfs per square mile. Thus the larger the installation, the greater the amount of energy turned out by the plant during any given year. It should, however, be remembered that the annual capacity factor of a hydro plant decreases with increased installation. The plant has ample pondage for weekly regulation, which is a prime requisite for any hydro project if all its capacity above that required for minimum stream flow is ever to become firm on the load curve.

In order to arrive at the capital cost per kw of installation for various capacity factors, an actual operating modern plant on this river was selected. The actual total costs were segregated in items which would be the same regardless of installation, such as lands, water rights, dams, railroad changes, and highway, and the items which vary with the amount of installation, such as power house, equipment, turbines, and transmission. This latter group constitutes the increment cost of installation and was found to be approximately $70 per kw. Starting with an installation which gives 100 per cent capacity factor (or that required to just utilize the minimum weekly flow), the installation was increased

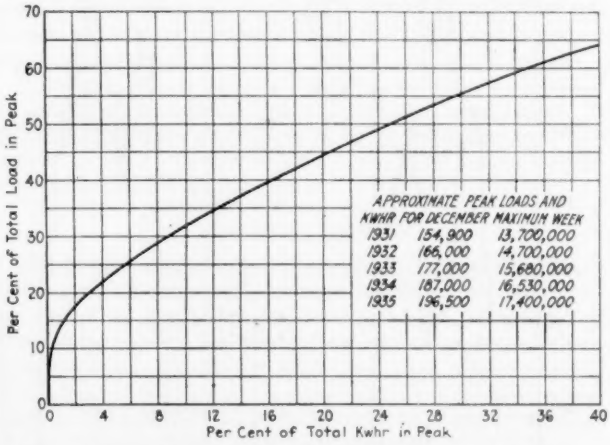


FIG. 3 PEAK PERCENTAGE CURVE FOR MAXIMUM WEEK OF DECEMBER

(Showing per cent of total kwhr in peak of load of regional power system Load factor of curve, 55 per cent.)

by increments and the capacity factor and cumulative cost determined for each increase in installation. To get the unit cost, the total was divided by the total installation.

Curve *C* then shows for a fairly typical case the tremendous variation in unit capital cost of hydro according to the amount of installation installed. Thus, if the project is planned on a 90 per cent annual capacity-factor basis, the capital cost would be about $580 per kw, whereas if planned on a 20 per cent annual-capacity basis, the capital cost would (by curve *C*) be about $100 per kw. It should here be noted that in order to be firm capacity on such a low capacity-factor basis, it would probably be necessary for the connected load to be three or four times the installed

hydro capacity, and thus the load would absorb all hydro energy generated. The curves are applicable only for the case where all hydro capacity installed would be firm capacity on the load curve. In fact, it seldom pays to consider hydro installations on a low capacity-factor basis unless all the capacity is firm.

In order to obtain curve *B*, it was necessary to obtain the amount of energy available for the installation required to give the various capacity factors. For this purpose, flow and power duration curves of the fairly typical stream selected were utilized.

Total annual charges on hydro were taken at 10 per cent of capital cost, and to get the total cost of energy, the annual cost per kw was divided by the energy produced per kw of capacity. Curve *B* shows the typically very high cost of hydro energy as compared to steam at high capacity factors and its relatively low cost compared to steam on low capacity factors. At a 70 per cent capacity factor, it will be noted that in the given case the costs of steam and hydro power are equal.

While there are many situations in which, if curves similar to those shown were plotted, the results obtained would be quite different from those shown, nevertheless it is believed that the curves are fairly typical for many situations.

Most emphatically the curves should not be interpreted as showing that either steam or hydro is the cheaper source of power supply. The only thing that ever makes it feasible to install a hydro project on a relatively low annual capacity-factor basis is the existence of a connected load several times the hydro capacity served largely by steam plants.

Under more or less exceptional conditions, hydro plants having very high annual capacity factors are economically feasible. Niagara Falls is a frequently cited example of this. There are

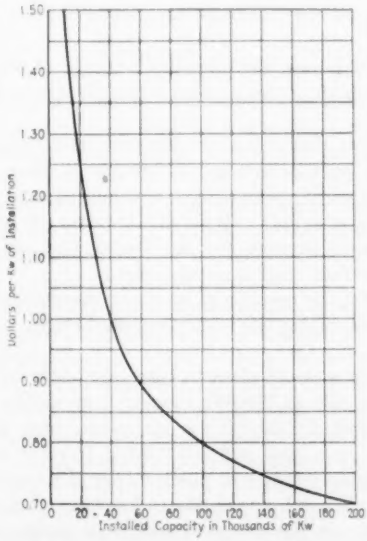


FIG. 4 OPERATION PLUS MAINTENANCE COSTS AT TYPICAL MODERN HYDRO PLANTS
(Based on a study of records of over 50 plants.)

also many projects on rivers which are well regulated by natural or artificial storage where costs on high capacity factors are relatively low.

ANNUAL COSTS OF HYDRO PLANTS

There is a wide variation in the practices of various companies with regard to the fixed charges set up for economic studies. The actual rate utilized is not of as great importance as it is that the fixed charges for both steam and hydro plants should be on a truly comparable basis.

Cost of money should be the same in either case, but the total fixed charges on hydro should almost always be less for hydro than for steam. Depreciation should be much less for hydro than for steam, because a very high percentage of the total investment is in items that are not depreciable or that are subject to a very slight depreciation, such as lands, riparian rights, and structures of earth and concrete.

In any particular case the annual depreciation should be

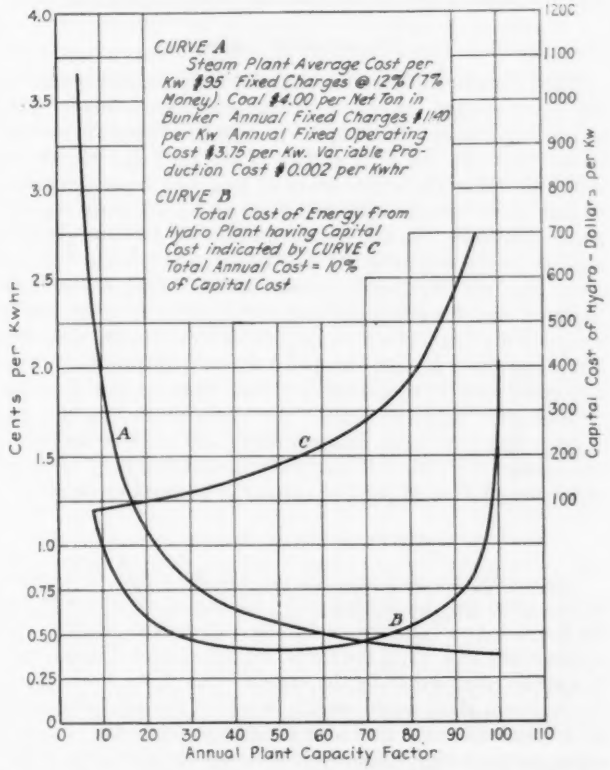


FIG. 5 TOTAL COST OF ENERGY AT VARIOUS CAPACITY FACTORS AT STEAM AND HYDRO PLANTS

computed item by item by the sinking-fund method. The overall annual rate so computed will in general vary from 1.5 per cent for projects where most of the investment is in the power plant, down to 0.5 per cent for a project where a large proportion of the nvestment is in water rights, land, earth dams, and tunnels. In making preliminary studies it is usually satisfactory to select a rate of depreciation between these two limits, based on well-informed judgment. For illustrative purposes only, 1 per cent per year on the total cost of the project has been selected as representing depreciation on an average hydro plant and being fairly comparable with the $3^1/_2$ per cent used on steam plants herein.

The cost of taxes and insurance is generally less for hydro plants. The relative investment in items on which insurance is issued is much less, and taxes are usually at a lower rate because hydro plants are usually located in rural sections remote from the centers of population.

Some small hydro plants have very high annual costs for operation and maintenance. Some of them with capacities of a few thousand kw or less have costs for operation and maintenance up to $6 per kw per year. However, with the present availability of remote control and automatic apparatus, there is probably no need for such costs exceeding about $2 per kw per year. Operation and maintenance costs at hydro plants should vary roughly with capacity, size, and number of units. Based on a study of

the cost records of a large number of operating plants, Fig. 4 shows approximate operation and maintenance costs for typical modern hydro plants.

Based on the foregoing data, the annual costs for hydro plants have been taken as follows for the illustrative purposes of this paper: Money 7 per cent; depreciation, 1 per cent; taxes and insurance, 1 per cent—total fixed charges, 9 per cent.

To obtain total annual cost, there should be added the annual cost of operation and maintenance as obtained from Fig. 4.

Transmission Costs

When considering the advisability of adding a hydro plant to any power system, the cost of transmission is sometimes of paramount importance. Some hydro projects are located in sections remote from the center of gravity of the load, and require for their utilization the construction of long and expensive transmission lines. On the other hand, in the case of some regional power companies, the additional transmission facilities required must be just as extensive if a new steam plant is built. In any given case the transmission requirements should be carefully analyzed, and the additional cost of transmission which will be necessary if a hydro plant is built, but which will not be necessary if a steam plant is built, should be carefully determined. This differential cost of transmission is sometimes spoken of as the "transmission liability" against the hydro plant. In the case of large hydro projects requiring 60 to 150 miles of additional transmission for their sole use, the "transmission liability" may vary from $2.50 to $7 per kw per year of installed capacity.

Value of Hydro Energy

It is usually assumed that the value of the output of a hydroelectric plant may be determined by the annual capacity cost and the increment cost of energy at an alternative steam plant. This is necessarily true only if the annual capacity factor of the alternative steam plant would be the same as that of the hydro plant. A new steam plant would usually be one of the most efficient in the system and would have a very high capacity factor. In such a case, excess energy which would annually be turned out, multiplied by the difference in increment steam cost between the proposed new plant and existing steam plants, should be subtracted from the total value of the hydro energy.

It should, however, be recognized that this is only a temporary liability of the hydro plant. As new steam plants are added, the capacity factor of the alternative steam plant will decrease, until it will soon be no greater than that of the hydro plant. For this reason it is often sufficient to consider this liability as counterbalanced by certain hydro advantages which may not always be readily evaluated in dollars and cents, such as peak-load service and reserve capacity.

Set-Up for a Typical Hydro Project

A power system requires additional capacity of about 50,000 kw in order to be able to serve the anticipated load of 1934. A suitable hydro site is available, and the company desires to determine whether it will be more economical to install a hydro plant or a steam plant for the next step in the development of their power-supply resources. Thorough field investigations, office studies, and estimates of cost for the hydro project have been made. The pondage at the site has been determined as ample for weekly regulation with only slight drawdown. By methods already outlined, it has been determined that 50,000 kw of capacity may be installed at the site and be firm capacity on the load curve of 1934. The energy available during a peak-load week coincident with minimum stream flow has been determined as 2,000,000 kwhr. Accordingly, during such a week, the plant would be operating on a load factor of 23.8 per cent. The output of the plant during the average year has been determined as 240,000,000 kwhr, giving an annual capacity factor of 55 per cent. The characteristics of the project are evidently such that although only 50,000 kw capacity would be firm in 1934, a much larger installation will prove advisable at some time in the future, and accordingly provision has been made in the design for readily extending the power house and increasing the capacity. The total capital cost of the project with an initial installation of 50,000 kw has been estimated as $8,750,000.

In accordance with the principles outlined, a careful study has been made of the transmission liability of the project, which has been determined as $3.72 per kw per year of installed capacity.

The alternative steam plant would be located at the load center, and its annual cost has been determined as $15 per kw per year, plus 2 mills per kwhr for energy.

The transmission losses are 6 per cent in capacity at time of full load, and average loss of energy would be 4 per cent. The delivered power would be $0.94 \times 50{,}000 = 47{,}000$ kw, and the delivered energy would be $0.96 \times 240{,}000{,}000 = 230{,}400{,}000$ kwhr. Based on this data, a set-up for the project may be made as follows:

1	Annual value of hydro power and energy delivered in an average year (based on costs at alternative steam plant):	
	47,000 kw delivered at $15	$705,000
	230,400,000 kwhr delivered at 2 mills	460,800
	Total annual value of hydro output delivered	$1,165,800
2	Annual total cost of hydro power delivered:	
	Fixed charges on hydro development, 9 per cent on $8,750,000	$787,500
	Operation and maintenance of plant (Fig. 4) 0.95 × 50,000	47,500
	Total annual cost of transmission (as above)	186,000
	Total annual cost of hydro power delivered	$1,021,000

Hence in this case the annual advantage of the hydro plant over the alternative steam plant is $1,165,800 — $1,021,000 = $144,800.

It is believed that this method is preferable to any attempt to show return on the investment, because it shows directly the effect on the earnings statement of adopting one of alternative sources of power supply. It should, however, be recognized that the investment required for the hydro alternative is usually much greater than that of a steam plant.

Accordingly, the reaction of an executive charged with the responsibility of raising the money required might be expressed as follows:

"If we built a steam plant to take care of the increasing load, it will cost us only $4,700,000, whereas if we build a hydro plant, we must spend $8,750,000 for the plant, plus $1,550,000 for transmission, or a total of $10,300,000. In other words, we will have to spend over twice as much money if we adopt the hydro alternative. What I should like to know is, what return are we going to make on this proposed additional investment?"

This question may be answered, with the aid of the data already given, as follows:

ANNUAL COST OF DELIVERED HYDRO POWER EXCLUDING INTEREST ON EXCESS INVESTMENT IN HYDRO

Depreciation, taxes and insurance on hydro development, 2 per cent on $8,750,000	$175,000
Operation and maintenance of hydro plant	47,500
Operation, maintenance, depreciation, taxes and insurance on transmission facilities, 5 per cent on $1,550,000	77,500
Interest charges on cost of alternative steam plant, 7 per cent on $4,700,000	329,000
Total annual cost of delivered hydro, exclusive of interest on excess investment of $5,600,000 ($10,300,000 — $4,700,000)	$629,000
Annual sum available for return on excess investment in hydro, $1,165,800 — $629,000	536,800

Return on excess investment in hydro = 536,800 ÷ 5,600,000 = 9.6 per cent.

Economic Balance Between Steam and Hydro Capacity

The hydro ratio is sometimes used to express the relationship between the hydro capacity in a given system and the annual peak load of the system. The "economic hydro ratio" may be defined as that ratio of hydro installation to peak load which for the given company gives the minimum annual cost of power supply. The economic balance between steam and hydro capacity for a given system at any given time is largely dependent on the following factors: shape and size of load curve, availability of suitable hydro sites, additional transmission facilities required for hydro ("transmission liability"), runoff and its seasonal distribution, availability and cost of storage, cost of fuel, and availability of condensing water. Clearly the "economic hydro ratio" may vary from zero in sections where hydro sites are extremely expensive and fuel cheap up to 100 per cent for sections where fuel is extremely expensive and suitable hydro sites with well-regulated stream flow abundant.

In some locations where fuel is cheap, but where favorable hydro sites are available, the economic hydro ratio has been shown to be from 25 to 40 per cent. In certain other regions where fuel is somewhat more costly, but where very favorable hydro sites are available with relatively cheap storage, the economic hydro ratio is from 60 to 70 per cent. In certain sections of Canada, the ratio is even higher.

A further decrease in the cost of steam-generated power will to some extent affect the economic hydro ratio, but will have more effect in changing the function of the new hydro plants constructed; that is, decreases in cost of steam will dictate the construction of hydro plants designed to operate on lower annual-capacity factors.

Discussion

W. F. Uhl.[4] This paper is a brief but comprehensive exposition of the economic relation between hydro and steam power in a large interconnected public-utility system.

There are still instances, of course, where hydro and steam power are truly competitive, as in the case of industrials producing their own power and some of the smaller public utilities. In the case of industrials, the problem is often complicated on account of the demand for heat for various purposes.

Except for a few companies located in the Rocky Mountain states and at Niagara Falls, all of which operate with high load factors, dependence on hydro power alone has not been found to be economical. The earliest industrial water-power developments of importance, such as those at Lawrence, Lowell, Holyoke, etc., recognized the economic importance of development much beyond the minimum stream flow, and arranged to sell surplus mill powers which were supplemented by steam plants installed by the industries purchasing such surplus water. Such surplus water has become less valuable as steam-power generation has become more efficient.

The contrary is the case with large interconnected public-utility systems operating with comparatively low load factors, if storage and pondage can be developed at reasonable cost, and this has been pointed out by the authors, who have shown simple methods for the determination of the necessary economic facts.

It is comparatively simple to make an economic study of the relations between hydro and steam power for any assumed conditions of load, but it must be recognized that since conditions which may exist in the future are always dealt with in such studies, the results should not be taken too seriously. Much time and effort are often wasted in making such studies with too great precision.

Perhaps our public-utility systems will not change so rapidly in the future as they have in recent years, and calculated comparisons may then hold true for longer periods. In some cases it has proved helpful to make two sets of assumptions covering future conditions as to load, interconnected plants, etc., thus obtaining comparisons within certain rational limits.

The authors have defined certain terms that are often misused, such as "increment cost of energy," "firm capacity," etc. It is of the utmost importance that the meaning of such terms be properly understood, as much confusion has resulted from the careless use of such terms in the past.

The authors have pointed out that there is a wide variation in the practices of various companies with regard to the fixed charges set up for economic studies. This is true, and it is believed that in many cases this is the result of a misunderstanding of the facts. The cost of money and the rate of return have been badly confused. Since the fixed charges on hydro plants are the only annual charges of importance, and since the interest charges (cost of money) form a large percentage of such charges, it becomes very essential that this cost be quite definitely established when the annual cost of hydro and steam power is compared. The cost of power (the product that a public utility sells) has nothing to do with the rate of return on the value of all the property, tangible and intangible, that the utility uses in its business, and which rate of return includes interest and profit.

The average cost of money, interest and dividends, has been found to amount to about 6 per cent on their assets for a number of public utilities recently coming under our observation.

Our examination of the actual cost of taxes and insurance on hydro plants and steam plants owned by the same company, covering a period of 11 years, showed that the average taxes for this period amounted to 0.81 per cent for the hydro and 1.23 per cent for the steam plants; the insurance amounted to 0.0307 per cent for the hydro and 0.264 per cent for the steam plants. In both cases the percentages are on the total actual investment in plants, including all overheads.

The relative depreciation between hydro and steam plants is often a matter of controversy. Recently a hydro plant built in 1900 was carefully examined, and it was decided to install one new modern vertical unit in place of one of the old horizontal double-runner units. Three of the old units are still in operation and in fair mechanical and electrical condition. The same penstock that supplied the old unit was utilized for the new unit. This penstock was thoroughly examined and found to be in substantially perfect condition. No important changes were made in the power house and none whatever on the intake, dam, tailrace, etc. These structures are all in excellent condition, and very little has ever been spent on them for maintenance and repairs. What rate of depreciation should have been used to calculate the yearly cost of power from this plant?

Where are the steam plants built in 1900, and what rate of depreciation (including obsolescence) is applicable to such plants?

The situation as described for this particular hydro plant is probably unusual for one built as long ago as 1900 or thereabout, but it certainly should not be unusual for plants of any importance built more recently, if designed and the construction supervised by competent engineers.

A hydro plant does not always have to carry a transmission liability. A case recently came to our attention where a large company had a hydro development near the outskirts of its system, where it supplies most of the local load. In determining the value of power lost by diversion of water for municipal use, this plant was credited with savings of both transmission investment and transmission losses. Voltage regulation is also frequently

[4] Hydraulic Engineer, Chas. T. Main, Inc., Boston, Mass. Mem. A.S.M.E.

improved by even small hydro plants near the ends of transmission or distribution feeders.

Our experience seems to indicate that the apparent saving in operating cost of hydro plants by the installation of automatic equipment is generally more than wiped out by the additional fixed and maintenance charges on the equipment. If the plant is of any importance, a watchman is often required in any case and this also should be taken into account. Partially automatic plants will often indicate a saving where a fully automatic plant would not make such a showing.

GEO. A. ORROK.[5] This paper is a welcome addition to the Society proceedings, since it covers the newer ideas regarding economic balance between steam and hydro capacity and the economic hydro ratio as well as certain features of increment cost that have been perhaps too little emphasized in the past. These studies follow the earlier peak-load–base-load studies of steam-plant combinations and make very clear the values of the addition of pondage and storage to the hydro plant in its connection with a steam-hydroelectric supply system. Many of us who have been accustomed to paying $2^1/_2$ per cent for taxes may not feel easy at the suggestion that taxes rarely exceed 1 per cent when they concern property values; but this is a minor matter, and the argument in the paper has been carefully thought out and fully explained.

It is a question whether the Hopkinson method of calculating the kilowatt-hour cost of steam plants gives any better results than the more ordinary methods that have been used by Professor Christie and myself, and it is my personal opinion that a reserve should be taken into account in connection with the steam costs, particularly in important installations. I note that the differences in the steam cost between the figures in this paper and my paper at Pittsburgh last November are of the order of tenths or hundredths of a mill when the reserve is taken out of consideration.

The authors are to be congratulated on their presentation of a difficult subject.

A. G. CHRISTIE.[6] This paper is timely and will be of especial interest to engineers. The extended interconnection of power systems in which there are frequently available certain capacities of water power that are now or may be developed has increased the importance of the economic balance between steam and hydro. As the authors emphasize, these two supplies are no longer competitive, but are now complementary.

In the second paragraph the authors discuss the trend of steam-plant performance and indicate 10,000 Btu per kwhr with 2500 lb and 1000 F. This figure checks closely with some computations on plant efficiency made by the writer and appears to be about the best that can be hoped for with present processes of steam generation.

In connection with fixed charges on the steam plant the authors use 3 to 4 per cent as a depreciation allowance, which, on a straight-line basis, would presume an average life of 25 to 33 years for the plant and equipment. This life allowance appears to be too great when full consideration is taken of obsolescence and supersession. It is a fair presumption that gains in economy are still possible in steam-station equipment, and these will lead to supersession of the older units, particularly if fuel prices tend to increase to any great extent above current levels. A higher allowance for depreciation would seem justifiable.

The discussion of firm hydro capacity might be amplified by considering the case where the hydro plant operates as a base-load plant during stages of high water and the steam station as a base-load plant during low water when ample pondage is provided at the hydro plant to store over the 24 hours. These changes in operating conditions involve considerable modifications in load dispatching. Many systems operate successfully on this method of loading.

The authors say nothing about the economics of pumped storage, although provision for such has been made in one or more recent plants of large size.

Fig. 3 shows a curve that, together with the load-duration curve, should receive more study by engineers. The writer has emphasized for years the relatively small output represented by the upper portion of the system s installed capacity. For instance, the upper 20 per cent of the total system capacity generates only 3 per cent of the total output in kwhr. This warrants the installation of plants of minimum total production cost of energy for such peak-load service.

Ample pondage is assuming an important place in the consideration of hydro developments. As the authors point out, the increment cost of added capacity to a given plant is relatively small. On the other hand, if pondage is sufficient, this added capacity can carry portions of the peak load at a lower cost than equivalent steam plant.

The figures in the comparisons under the heading, "Set-Up for a Typical Hydro Project," show a rational engineering analysis of the problem. All hydro-steam system problems are not so susceptible of easy solution. In one case, the problem of evaluating secondary hydro power available only during high water that occurred in summer time presented many difficulties. The steam station at the end of a transmission line 175 miles long had to be maintained at full capacity to carry the winter peak loads should outage occur. This placed a small value on the secondary hydro power. Another problem in the same system was to determine the proper place of steam-generated power if used as base load in the winter months and to assess a proper valuation for such power. In this particular system the hydro plant and the steam plant were owned by different interests. Problems involving costs were therefore approached from different viewpoints.

In Ontario and Quebec, reliance has been placed on hydro power alone due to the fact that foreign coal must be imported into these areas. If this restriction could be removed by ample supplies of Canadian or British coals, some interesting problems of economic balances between steam and hydro power would develop. The result would probably lead to the development of steam stations in the larger centers, although Diesel plants need consideration in certain cases.

W. P. CREAGER.[7] The authors give a rather lengthy computation to indicate the percentage return on the excess investment in hydro. Instead of this means of determining it, I would suggest a more abbreviated method.

The total return on the excess hydro investment is equal to the annual saving by the hydro plus the interest charges on the excess investment. The percentage return may then be computed as follows:

$$\text{Percentage return} = \frac{144{,}800}{5{,}600{,}000} + 7.00 = 9.6$$

R. L. THOMAS.[8] A perennial question on which engineers seem to disagree within an astonishingly wide range and which the authors handle with much discrimination is the rate or rates of fixed charges to be applied to investment in steam and hydro

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generating plants. In 1929 Mr. W. W. Tefft[9] used 18 per cent for steam and 10 per cent for hydro. In an A.I.E.E. paper[10] presented at Cleveland in June, 1932, Mr. R. E. Dillon said of steam plants that "the fixed charges per kilowatt of new capacity at the present time would be $17." Messrs. Irwin and Justin use 12 per cent or $12 per kilowatt for steam (exclusive of the fixed operating cost) and 9 per cent for hydro. They call attention to the often neglected fact that "the actual rate utilized is not of as great importance as it is that the fixed charges for both steam and hydro plants should be on a truly comparable basis."

Nevertheless, the absolute values, regardless of the differential, do have considerable influence on the result of the authors' method of comparison. Referring to their numerical example showing an advantage of $144,800 in favor of hydro, this advantage would be increased if both rates of fixed charges were decreased alike, and it would be decreased if the rates of fixed charges were increased. If both rates were increased sufficiently, the credit to hydro would become a debit. It follows that the omission of an item of fixed charges, such as a tax, does affect the result, even if it is the same in per cent for both hydro and steam.

The authors' treatment, or rather their disregard, of income and gross receipts taxes in computing fixed charges raises some interesting and puzzling questions. Under the heading, "Component Parts of Steam-Operating Costs," there is the statement that "It is not correct to include such taxes in an economic study, because they are dependent on output or profit and bear no relation to investment."

The writer does not believe that these taxes should be thus disregarded. If these taxes are not correctly included in an economic study, where should they be included? Someone must pay them. It would seem that neglect of them is justifiable only on the assumption that they must be paid out of the allowed return, or in effect by the stockholders, and it is not supposed that the authors would agree to this.

Confining the discussion for the time being to income tax alone, the authors' development of their subject does not seem to bear out the statement that this tax "bears no relation to investment." The authors have quite properly included as the first component of the fixed charges return on the investment rather than bare cost of money. This return is or at least includes the "profit." The income tax depends on this return minus bond interest. The larger the investment, the larger the return; the larger the return, the larger the income tax. Therefore under the authors' method of varying the return (i.e., in dollars) with the investment the income tax does seem to bear a relation to the investment.

The question can best be illustrated by means of the authors' numerical example under the heading, "Set-Up for a Typical Hydro Project." (The writer assumes that "fixed charges" throughout the paper includes return, in spite of the use of the terms "Money, 7 per cent" and "Interest charges, 7 per cent.")

There are two ways of looking at the problem. One is that the company's rates and gross revenue are varied just the proper theoretical amount to yield the specified rate of return, whether the hydro or the steam plant is built. The other is that the rates and gross revenue are the same in both cases (hydro and steam). The inclusion of "return" in fixed charges seems to commit us to the first set-up. That is, the "annual advantage" of $144,800 really means that in order to yield the return which is included in the figures the rate payers would have to pay $144,800 more with the steam plant than with the hydro plant. However, the writer does not believe the answer is correct because the income tax has been omitted. If the financing (i.e., ratio of bonds to stock, net price of bonds, etc.) is the same in the two cases, the income tax in the hydro case will be greater than in the steam case in the ratio of $10,300,000 to $4,700,000 and the advantage of the hydro plant will be correspondingly less than $144,800.

If we change to the assumption that the rates and gross revenue are the same in both cases (hydro and steam), then to compute correctly the relative advantage the writer believes that we must use actual interest instead of nominal return, compute the income tax in each case, and express the final difference in terms of dollars available for dividends per share.

As to gross receipts tax, it seems obvious that if a nominal return is assumed, the revenue and the tax thereon will vary with investment. If the rates and revenue are assumed the same in both cases, the gross receipts tax will also be the same.

The writer's conclusion is that the nominal-fixed-charge method is useful for approximations, but that the rate of fixed charges should include all taxes. Any final decision, however, must be based on a detailed estimate of every individual cost, such as interest and the several taxes.

The authors are to be congratulated and thanked for their clear and complete analysis of the economic balance between hydro and steam.

CARROLL F. MERRIAM.[11] This paper is particularly apropos, because it is too little appreciated that when hydro is properly coordinated with steam, it will generally lead to a total cost of power that is less than that for either source alone. It is frequently assumed that steam and hydro are in direct competition, and consequently the more favorable of the two can be determined by comparison of set-ups made on the assumption that either will be the sole source of power. This paper gives consideration to the mutual advantages, which interlock so closely that steam and hydro are supplementary rather than competitive.

From the standpoint of an operating company it is appreciated that advantages of more or less intangible nature have not been attempted to be evaluated by the authors. The consideration of these is largely a matter of engineering judgment rather than of formal analysis. Those that have had experience in operating a steam-hydro system for maximum over-all economy realize that the two become so intimately allied that practically all problems involve a careful balancing of manifold elements before a true solution can be found. Furthermore, it is impossible to secure the highest degree of economy if contractual restrictions prevent the system from being planned and operated as a unit.

When water power is to be developed, an all-important element in the success or failure of the enterprise is not only the market for energy, but the relation between hydro and steam capacity, as well as the service that hydro can render, not only in kilowatt-hours, but in firm power. Neglect of these facts may lead to failure of a project that otherwise might be a financial success.

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[9] "Power—Steam or Hydro, or Both," by William W. Tefft, presented at the Annual Meeting, New York, N. Y., December 2 to 6, 1929; Transactions (1930), vol. 52, no. 1, paper no. HYD-52-6, p. 65.

[10] "Combined Reliability and Economy in Operation of Large Electric Systems."

[11] Pennsylvania Water and Power Company, Baltimore, Md. Mem. A.S.M.E.

FSP-55-6

Stresses in Boiler Tubes Subjected to High Rates of Heat Absorption

By Wm. L. De Baufre,[1] Lincoln, Neb.

In order to get a more nearly accurate picture of the stresses in thick boiler tubes subjected to high rates of heat absorption, an investigation of the subject, under the supervision of the author, was made using as nearly accurate experimental data as could be found for mild steel and employing theoretically correct relations for the stresses in thick tubes. After calculating the stresses by relations derived for a constant ratio of stress to strain, the results were corrected for plastic flow in the tube wall where the stresses exceeded the elastic limit of mild steel, and also for creep which occurs when mild steel is stressed for long times at elevated temperatures. The stresses were investigated for non-uniform heat absorption around the tube circumference as well as for a uniform rate of heat absorption. Details of the investigation and the results obtained are set forth at length in the paper.



ATTENTION has been called in recent years to the high stresses set up in the tubes of steam boilers due to high rates of heat transfer from burning fuel and products of combustion to water boiling within the tubes. It has been suggested that it would be desirable to use thinner tubes than called for in the A.S.M.E. Boiler Code in order to reduce the total stresses due to heat transfer and internal fluid pressure.

Previous investigations of these stresses have in general been based on approximate data and relations and on a constant ratio of stress to strain, although the calculated stresses have exceeded the elastic limit of the mild steel usually used in such tubes. In order to get a more nearly accurate picture of the stresses in thick boiler tubes subjected to high rates of heat absorption, an investigation was made in the Technical Research Department of International Combustion Engineering Corporation, using as nearly accurate experimental data as could be found for mild steel and employing theoretically correct relations for the stresses in thick tubes. After calculating the stresses by relations derived for a constant ratio of stress to strain, the results were corrected for plastic flow in the tube wall where the stresses exceeded the elastic limit of mild steel, and also for creep which occurs when mild steel is stressed for long times at elevated temperatures. The stresses were investigated for non-uniform heat absorption around the tube circumference as well as for a uniform rate of heat absorption.

The investigation has shown that where the stresses produced by heat transfer and internal fluid pressure exceed the elastic limit of the material near the inner and outer surfaces of the tube wall, the calculated stresses based on perfect elasticity are greatly reduced by plastic flow, but residual stresses are thereby set up when heat transfer ceases and the internal fluid pressure is reduced to zero. Every time a boiler goes into and out of operation, a stress cycle from tension to compression occurs of considerable stress range. This stress range is greatest for the circumferential stress at the inner surface of the tube wall when the tube is heated uniformly. When heated non-uniformly, the axial-stress range may exceed the circumferential-stress range under certain abnormal conditions. It is shown by comparison with experimental data, however, that a boiler tube will not fail within any reasonable period of time by fatigue from these repeated stress reversals.

Before a boiler tube can fail by bursting, all portions of the tube wall must be stressed to a point where more or less rapid creep will occur to rupture. The investigation shows that before this can take place, all heat-transfer stresses will be entirely neutralized by creep. Failure of the tube then occurs by the stresses due to internal pressure only. A measure of the safety of the tube is the ratio of the stress which will cause creep under the mean operating temperature of the tube wall to the mean stress caused by internal fluid pressure. The investigation shows that an increase in tube safety is secured by using thicker rather than thinner tubes than called for by the A.S.M.E. Boiler Code.

An increase in safety may also be obtained by using a material which will not creep until a higher stress is reached, such as a medium-carbon steel in place of a mild carbon steel. Even with mild carbon steel subjected to a very high rate of heat absorption, however, the investigation shows that failure can occur only when the coefficient of heat transfer from the inner surface of the tube wall to the water-steam mixture within the tube is reduced to an abnormally low value. Boiler-tube failures therefore occur by reason of poor circulation of the water-steam mixture within the tubes, by accumulation of scale on the inner surface of the tube wall, by corrosion, or by defective material, rather than by any stresses due to unequal expansion within the tube wall caused by a high rate of heat absorption.

Boiler-Tube Temperatures for Various Rates of Heat Absorption

When heat is transferred through a boiler tube from burning fuel and products of combustion to water boiling within the tube, the tube wall becomes hotter than the inclosed water-steam mixture. The temperature of the tube wall depends upon the rate of heat absorption, the dimensions of the tube, the thermal conductivity of the material, the temperature drop from the

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 BOILER-TUBE TEMPERATURES (DEG F) FOR VARIOUS RATES OF HEAT ABSORPTION (3-IN. O. D. TUBES)

Gage pressure and corresponding saturated-steam temperature lb per sq in. and deg F	Wall thickness		Heat-absorption rate, Btu per sq ft of outside area per hr									
			30,000		45,000		60,000		75,000		90,000	
	Gage no.	Inch	Inside surface	Outside surface	Inside surface	Outside surface	Inside surface	Outside surface	Inside surface	Outside surface	Inside surface	Outside surface
125	12	0.109	363.6	372.5	369.0	382.3	374.4	392.2	379.8	402.0	385.2	411.9
(352.86)	11	0.120	363.7	373.5	369.2	383.8	374.6	394.2	380.0	404.6	385.5	414.9
	10	0.134	363.9	374.9	369.2	385.8	374.8	397.0	380.3	408.2	385.8	419.2
250	10	0.134	417.0	428.3	422.5	439.4	428.0	450.6	433.5	461.8	439.0	472.9
(406.02)	9	0.148	417.1	429.5	422.7	441.2	428.2	453.0	433.8	464.9	439.3	476.6
	8	0.165	417.3	431.1	422.9	443.7	428.5	456.3	434.0	468.9	439.7	481.6
400	9	0.148	459.2	471.7	464.8	483.6	470.3	495.4	475.9	507.3	481.4	519.2
(448.12)	8	0.165	459.4	473.4	465.0	486.0	470.6	498.8	476.1	511.3	481.8	524.2
	7	0.180	459.5	474.8	465.2	488.3	470.8	501.7	476.4	515.0	482.2	528.7
450	8	0.165	470.7	484.8	476.4	497.5	482.0	510.2	487.6	523.0	493.2	535.9
(459.50)	7	0.180	470.9	486.3	476.5	499.7	482.2	513.2	487.9	526.8	493.6	540.4
	6	0.203	471.1	488.6	476.9	503.2	482.6	517.9	488.4	532.7	494.2	548.0
600	7	0.180	500.1	515.7	505.8	529.2	511.5	542.8	517.2	556.4	522.9	570.1
(488.77)	6	0.203	500.3	518.0	506.1	532.8	511.9	547.6	517.7	562.4	523.5	577.3
	5	0.220	500.5	519.8	506.4	535.4	512.2	551.1	518.1	566.8	523.9	582.6
900	4	0.238	545.8	567.1	551.7	583.9	557.7	600.7	563.6	617.6	569.5	634.5
(533.87)	3	0.259	546.0	569.4	552.0	587.3	558.1	605.3	564.1	623.3	570.1	641.5
	2	0.284	546.2	572.2	552.4	591.5	558.6	610.8	564.7	630.3	570.9	649.9
1350	0	0.340	596.5	629.0	603.0	651.8	609.5	675.0	615.9	693.7	622.4	721.4
(583.60)	00	0.380	597.0	633.9	603.7	659.2	610.4	684.9	617.1	705.4	623.8	736.7
	000	0.425	597.6	639.6	604.5	667.8	611.5	696.5	618.5	725.4	625.5	754.7

TABLE 2 BOILER-TUBE TEMPERATURES (DEG F) FOR VARIOUS RATES OF HEAT ABSORPTION (3.50-IN. O. D. TUBES)

Gage pressure and corresponding saturated-steam temperature, lb per sq in. and deg F	Wall thickness		Heat-absorption rate, Btu per sq ft of outside area per hr									
			30,000		45,000		60,000		75,000		90,000	
	Gage no.	Inch	Inside surface	Outside surface	Inside surface	Outside surface	Inside surface	Outside surface	Inside surface	Outside surface	Inside surface	Outside surface
125	11	0.120	363.6	373.3	369.0	383.5	374.3	393.8	379.7	404.1	385.1	414.4
(352.86)	10	0.134	363.7	374.6	369.1	385.5	374.5	396.4	379.9	407.4	385.4	418.2
	9	0.148	363.8	375.9	369.2	387.4	374.7	398.9	380.2	410.5	385.6	422.1
250	9	0.148	416.9	429.2	422.4	440.8	427.9	452.5	433.3	464.2	438.8	475.8
(406.02)	8	0.165	417.1	430.8	422.6	443.2	428.1	455.7	433.6	468.2	439.1	480.7
	7	0.180	417.2	432.2	422.8	445.4	428.3	458.6	433.9	471.8	439.5	485.0
400	8	0.165	459.2	473.0	464.7	485.6	470.2	498.1	475.7	510.7	481.2	523.4
(448.12)	7	0.180	459.3	474.5	464.9	487.7	470.4	501.0	476.0	514.3	481.6	527.8
	6	0.203	459.4	476.7	465.1	491.1	470.7	505.5	476.4	519.9	482.1	534.6
450	7	0.180	470.7	485.9	476.2	499.2	481.8	512.5	487.4	525.9	493.0	539.3
(459.50)	6	0.203	470.8	488.2	476.5	502.6	482.1	517.0	487.8	531.6	493.4	546.1
	5	0.220	470.9	489.8	476.7	505.1	482.4	520.4	488.1	535.8	493.8	551.2
600	6	0.203	500.1	517.6	505.7	532.1	511.4	546.6	517.1	561.3	522.7	575.9
(488.77)	5	0.220	500.2	519.3	505.9	534.6	511.7	550.1	517.4	565.5	523.1	581.1
	4	0.238	500.3	521.1	506.1	537.4	511.9	553.7	517.7	570.1	523.5	586.6
900	2	0.284	545.8	571.4	551.8	590.3	557.8	609.2	563.7	628.3	569.7	647.7
(533.87)	1	0.300	545.9	573.1	552.0	592.8	558.0	612.6	564.1	632.6	570.1	652.8
	0	0.340	546.3	577.5	552.5	599.4	558.7	621.5	564.9	643.7	571.1	666.2
1350	00	0.380	596.4	632.3	602.8	657.0	609.1	681.7	615.5	706.9	621.9	732.3
(583.60)	000	0.425	596.8	637.8	603.4	665.1	610.0	692.8	616.6	720.8	623.2	749.1

TABLE 3 BOILER-TUBE TEMPERATURES (DEG F) FOR VARIOUS RATES OF HEAT ABSORPTION (4.00-IN. O. D. TUBES)

Gage pressure and corresponding saturated steam temperature, lb per sq in. and deg F	Wall thickness		Heat-absorption rate, Btu per sq ft of outside area per hr									
			30,000		45,000		60,000		75,000		90,000	
	Gage no.	Inch	Inside surface	Outside surface	Inside surface	Outside surface	Inside surface	Outside surface	Inside surface	Outside surface	Inside surface	Outside surface
125	10	0.134	363.6	374.4	368.9	385.2	374.3	396.1	379.7	406.9	385.0	417.8
(352.86)	9	0.148	363.7	375.7	369.1	387.1	374.5	398.6	379.9	410.1	385.3	421.5
	8	0.165	363.8	377.2	369.2	389.4	374.7	401.7	380.1	414.0	385.6	426.2
250	8	0.165	416.9	430.5	422.4	442.9	427.8	455.2	433.3	467.7	438.7	480.1
(406.02)	7	0.180	417.0	431.9	422.5	445.0	428.0	458.0	433.5	471.2	439.0	484.3
	6	0.203	417.2	434.1	422.7	448.2	428.3	462.4	433.9	476.6	439.4	490.9
400	6	0.203	459.3	476.4	464.8	490.6	470.4	504.9	476.0	519.2	481.5	533.6
(448.12)	5	0.220	459.4	478.0	465.0	493.1	470.6	508.2	476.2	523.3	481.8	538.5
	4	0.238	459.5	479.8	465.2	495.7	470.8	511.7	476.5	527.7	482.2	543.8
450	6	0.203	470.6	487.8	476.2	502.1	481.8	516.4	487.3	530.8	492.9	545.2
(459.50)	5	0.220	470.7	489.5	476.4	504.5	482.0	519.7	487.6	534.8	493.2	550.1
	4	0.238	470.9	491.2	476.5	507.2	482.2	523.2	487.9	539.3	493.6	555.4
600	4	0.238	500.1	520.7	505.8	536.7	511.5	552.8	517.2	569.0	522.8	585.3
(488.77)	3	0.259	500.3	522.8	506.0	539.9	511.8	557.1	517.5	574.3	523.2	591.7
	2	0.284	500.4	525.3	506.3	543.7	512.1	562.1	517.9	580.7	523.8	599.3
900	1	0.300	545.6	572.4	551.5	591.8	557.4	611.5	563.3	631.1	569.2	650.9
(533.87)	0	0.340	545.9	576.7	552.0	598.2	558.0	620.1	564.0	641.9	570.0	663.8
	00	0.380	546.2	581.0	552.4	604.7	558.6	628.8	564.8	652.8	570.9	676.9
1350	000	0.425	596.3	636.4	602.7	663.2	609.0	690.2	615.4	717.6	621.7	745.2
(583.60)												

inner surface to the water boiling within the tube, and the pressure under which steam is being generated. The temperatures of the outside and inside surfaces of mild-steel boiler tubes are given in Tables 1, 2, and 3 for several rates of heat transfer with 3-, 3.5-, and 4-in. tubes such as might be used for the steam pressures tabulated, the thinnest tube wall corresponding to that specified by the A.S.M.E. Boiler Code.

In calculating the data for the above-mentioned tables, the temperature of saturated steam was taken from Keenan's "Steam Tables" published by the A.S.M.E. The temperature drop from the inner surface of the tube wall through the water film to the water boiling within the tube was calculated by the formula

$$\Delta T = \frac{Q}{U}\frac{r_2}{r_1}$$

where Q = heat-absorption rate to outside of tube, Btu per hr per sq ft
r_1 = inside radius of tube, inches
r_2 = outside radius of tube, inches
U = heat-transfer factor through water film, Btu per hr per sq ft per deg F, and
ΔT = temperature drop through water film, deg F.

The heat-transfer factor U was taken equal to 3000 Btu per hr per sq ft per deg F, a value which has been found to exist under favorable circumstances between condensing steam and evaporating water. Measurements by Kreisinger and Barkley upon a Heine boiler, as described in U. S. Bureau of Mines Technical Paper 114, indicate a higher heat-transfer factor, so that the value selected is considered conservative for good operating conditions.

The temperature of the inside surface of the tube wall is equal to the saturated-steam temperature plus the temperature drop through the water film. The temperature of the outside surface of the tube wall is equal to the temperature of the inside surface plus the temperature drop through the tube wall as calculated by the formula derived in Appendix No. 1, namely:

$$T_2 - T_1 = \frac{Qr_2}{12\lambda}\log_e\frac{r_2}{r_1}$$

where Q = rate of heat absorption to outside surface of tube, Btu per hr per sq ft
λ = thermal conductivity of metal in tube wall, Btu per hr per sq ft per deg F for 1 ft thickness
T_1 = temperature of inside surface, deg F
T_2 = temperature of outside surface, deg F.

The thermal conductivity λ was selected in accordance with the data in Appendix No. 2 for the average temperature in the tube wall. This necessitated a trial-and-error method of computation. The outside-surface temperature was first assumed and the correctness of the assumption then determined by substituting the corresponding value of the thermal conductivity in the above formula.

The data in Tables 1, 2, and 3 are of interest in showing the rise of the tube-wall temperature with higher steam pressures and with increased rates of heat absorption. In water-cooled furnaces, the rate of heat absorption in the first rows of the boiler tubes varies from 30,000 to 60,000 Btu per hr per sq ft. In furnaces without water cooling, the rate of heat absorption in the first rows of boiler tubes may be higher and may reach 90,000 Btu per hr per sq ft due to the higher temperatures in refractory-lined furnaces. For a discussion of heat absorption by water-cooled surfaces exposed to radiation in a furnace, see "Heat Absorption in Water-Cooled Furnaces," by the present author, in Trans. A.S.M.E., Vol. 53 (1931), pp. 253–264.

Calculated Stresses in a Thick Boiler Tube

As shown in Appendix No. 1, the total radial, circumferential, and axial stresses at any point in a tube wall may be obtained by adding the stresses calculated separately for fluid pressure and for heat transfer. The formulas as derived in Appendix No. 1 are:

For fluid pressure:

$$p_p = \frac{r_2^2 - r^2}{r_2^2 - r_1^2}\frac{r_1^2 p_1}{r^2}$$

$$t_p = \frac{r_2^2 + r^2}{r_2^2 - r_1^2}\frac{r_1^2 p_1}{r^2}$$

$$s_p = \frac{r_1^2 p_1}{r_2^2 - r_1^2}$$

For heat transfer:

$$p_t = \frac{E\alpha m(T_2 - T_1)}{2(m-1)(r_2^2 - r_1^2)\log_e\frac{r_2}{r_1}}\left[-(r_2^2\log_e r_2 - r_1^2\log_e r_1) + \frac{r_1^2 r_2^2}{r^2}\log_e\frac{r_2}{r_1} + (r_2^2 - r_1^2)\log_e r\right]$$

$$t_t = \frac{E\alpha m(T_2 - T_1)}{2(m-1)(r_2^2 - r_1^2)\log_e\frac{r_2}{r_1}}\left[(r_2^2\log_e r_2 - r_1^2\log_e r_1) - (r_2^2 - r_1^2) + \frac{r_1^2 r_2^2}{r^2}\log_e\frac{r_2}{r_1} - (r_2^2 - r_1^2)\log_e r\right]$$

$$s_t = \frac{E\alpha m(T_2 - T_1)}{2(m-1)(r_2^2 - r_1^2)\log_e\frac{r_2}{r_1}}\left[2(r_2^2\log_e r_2 - r_1^2\log_e r_1) - (r_2^2 - r_1^2) - 2(r_2^2 - r_1^2)\log_e r\right]$$

For fluid pressure and heat transfer:

$$p = p_p + p_t$$
$$t = t_p + t_t$$
$$s = s_p + s_t$$

where r_1 = inside radius of tube, inches
r_2 = outside radius of tube, inches
p_1 = inside fluid pressure, lb per sq in. (outside fluid pressure being zero)
T_1 = temperature of inside surface, deg F
T_2 = temperature of outside surface, deg F
$1/m$ = Poisson's ratio
E = coefficient of elasticity at mean tube-wall temperature, lb per sq in.
α = coefficient of thermal expansion at mean tube-wall temperature, inch per inch per deg F
p_p = radial compressive stress at radius r due to fluid pressure, lb per sq in.
t_p = circumferential tensile stress at radius r due to fluid pressure, lb per sq in.
s_p = axial tensile stress at radius r due to fluid pressure, lb per sq in.
p_t = radial compressive stress at radius r due to heat transfer, lb per sq in.
t_t = circumferential tensile stress at radius r due to heat transfer, lb per sq in.
s_t = axial tensile stress at radius r due to heat transfer, lb per sq in.

p = total radial compressive stress at radius r, lb per sq in.

t = total circumferential tensile stress at radius r, lb per sq in.

s = total axial tensile stress at radius r, lb per sq in.

By means of the foregoing formulas, the stresses were calculated in an exceptionally thick boiler tube required for a very high steam pressure, 1840 lb gage. The diameter of this tube is

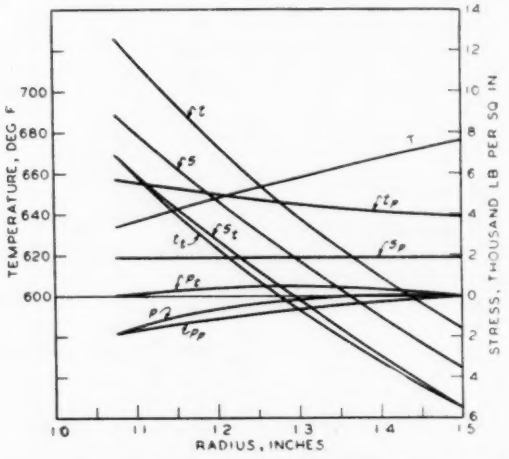


FIG. 1 CALCULATED STRESSES IN TUBE WALL FOR HEAT ABSORPTION OF 30,000 BTU PER HR PER SQ FT

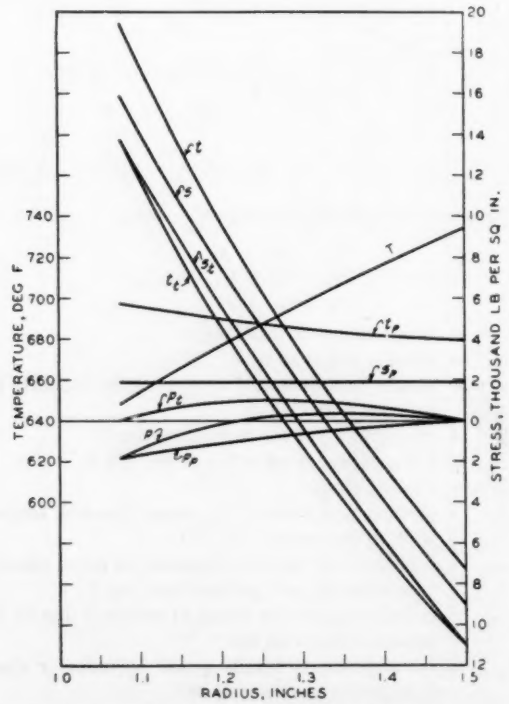


FIG. 2 CALCULATED STRESSES IN TUBE WALL FOR HEAT ABSORPTION OF 60,000 BTU PER HR PER SQ FT

3 in. and its thickness is 0.425 in., so that $r_2 = 1.5$ in. and $r_1 = 1.075$ in. The resulting stresses are shown in Figs. 1, 2, and 3 for rates of heat absorption of 30,000, 60,000, and 90,000 Btu per hr per sq ft of outside surface. The temperatures in the tube wall are also shown in Figs. 1, 2, and 3 for the several rates

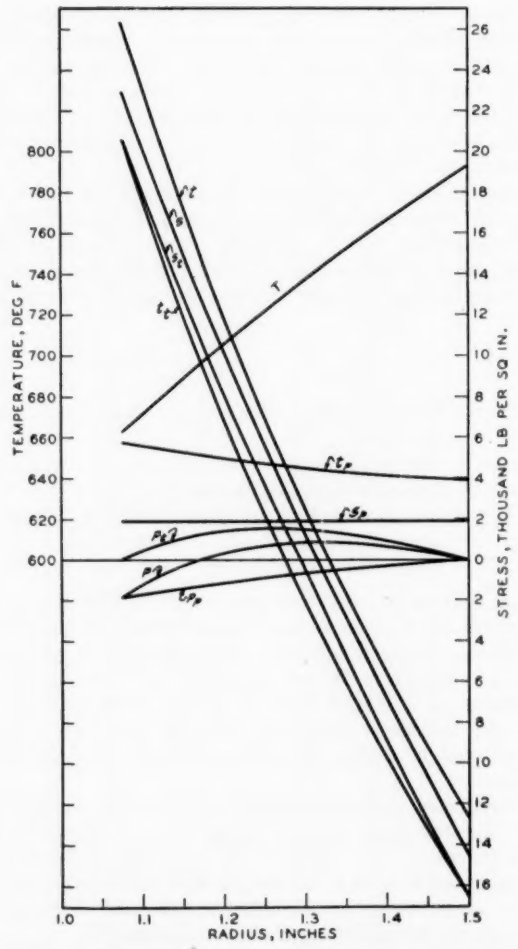


FIG. 3 CALCULATED STRESSES IN TUBE WALL FOR HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

of heat absorption. The maximum and minimum temperatures are as follows:

Rate of heat absorption, Btu per hr per sq ft	Saturated-steam temp., deg F	Temperature of inside surface, deg F	Temperature of outside surface, deg F
30,000	620	634.0	676.5
60,000	620	647.9	734.1
90,000	620	661.9	792.5

In calculating the stresses due to heat transfer, it was necessary to select proper values of the modulus of elasticity E, the coefficient of thermal expansion α, the coefficient of thermal conductivity λ, and Poisson's ratio $1/m$. As explained in Appendix No. 2, the last-mentioned quantity was taken as 0.30, while the values of the other three quantities were taken to correspond to the mean temperature in the tube wall.

In Figs. 1, 2, and 3 the stresses due to fluid pressure are the same for all three rates of heat absorption. The radial compressive stress p_p varies from the internal fluid pressure of 1840 lb per sq in. at the inside surface of the tube wall to zero at the outside surface. The circumferential tensile stress t_p varies from a maximum of 5726 lb per sq in. at the inside surface to a minimum of 3886 lb per sq in. at the outside surface. The summation of the circumferential tensile stress over the tube wall is equal to the total internal fluid pressure along a radius, that is,

$$\int_{r_1}^{r_2} t_p dr = p_1 r_1$$

The axial tensile stress is constant over the tube-wall cross-section and is equal to 1943 lb per sq in. The product of this axial stress and the cross-sectional area of the tube wall is equal to the total axial internal fluid pressure.

In Figs. 1, 2, and 3 the stresses due to heat transfer differ greatly for the three rates of heat absorption selected. In general, the circumferential and axial stresses t_t and s_t vary from the same maximum tensile stress at the inside surface to the same maximum compressive stress at the outside surface, but do not coincide at points within the tube wall. The maximum tensile stress at the inside surface is greater than the maximum compressive stress at the outside surface as shown in the following table:

Rate of heat absorption, Btu per hr per sq ft	Maximum circumferential and axial tensile stress at inside surface, lb per sq in.	Maximum circumferential and axial compressive stress at outside surface, lb per sq in.
30,000	6,841	5,486
60,000	13,721	11,004
90,000	20,618	16,536

The summations of the circumferential and axial stresses over the tube wall are each equal to zero. The radial stress p_t due to heat transfer only is zero at both the inside and outside surfaces; within the tube wall, the metal is in tension radially, but the maximum tension is small.

Adding the stresses due to fluid pressure and to heat transfer for the several rates of heat absorption, we find the total circumferential tensile stress at the inside surface of the tube wall to be the greatest stress in the material. The total circumferential stress varies from this maximum value in tension at the inside surface through zero at a point within the wall to a much lower maximum compressive stress at the outside surface. Its summation over the tube wall is equal to the total internal fluid pressure along a radius, that is,

$$\int_{r_1}^{r_2} t dr = p_1 r_1$$

The maximum total axial stress also occurs at the inside surface of the tube wall and varies from this maximum value in tension at the inside surface to a maximum value in compression at the outside surface, its summation over the cross-section of the tube wall being equal to the axial internal fluid pressure, that is,

$$\int_{r_1}^{r_2} s r dr = \frac{p_1 r_1^2}{2}$$

The total radial stress is low and will not receive further consideration.

The summation over the tube wall of the total stress due to both heat transfer and internal fluid pressure is evidently equal to the summation of the stress due to internal fluid pressure only.

The maximum values calculated for the total circumferential and axial tensile stresses at the inside surface of the tube wall are as follows:

Rate of heat absorption, Btu per hr per sq ft	Maximum circumferential tensile stress, lb per sq in.	Maximum axial tensile stress, lb per sq in.
30,000	12,567	8,784
60,000	19,447	15,664
90,000	26,344	22,561

These stresses were calculated on the assumption of perfect elasticity. By reference to Fig. 37 in Appendix No. 2 in which stress-strain diagrams are given for the mean tube wall temperatures at the several rates of heat transfer, it will be seen that the above stresses extend into regions of plastic flow beyond the elastic limits of mild steel at these temperatures. This fact is also shown by Fig. 36 in Appendix No. 2 giving the proportional limit stress of mild steel at elevated temperatures. Hence these calculated stresses will not be reached by reason of plastic flow in the tube wall adjacent to the inner surface.

Probable Circumferential Stresses in a Thick Boiler Tube

The variation of the circumferential stress in a thick boiler tube is due to relative strains in the tube wall. Thus, the heat-transfer stress is due entirely to relative expansion of the inner and outer portions of the tube wall. The variation in the fluid pressure stress is due to the inner portions of the tube wall first slightly elongating before the outer portions can offer any resistance to the internal pressure. Hence if we divide the stresses in Figs. 1, 2, and 3 by the modulus of elasticity of mild steel, we may consider the results as representing the deformations in the several directions, although the maximum stresses indicated in these figures are not reached owing to plastic flow.

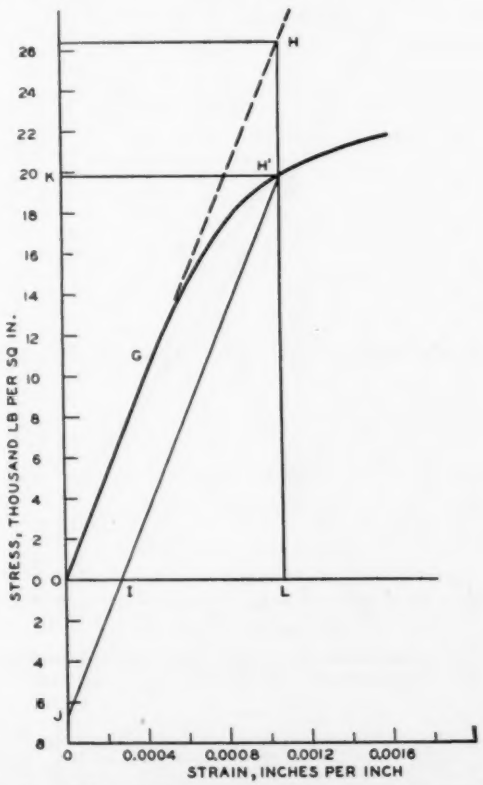


Fig. 4 Stress-Strain Diagram for Mild Steel at 717 F Mean Temperature of Tube Wall for Heat Absorption of 90,000 Btu per Hr per Sq Ft

Now refer to Fig. 4 where the point H represents the circumferential tensile stress of 26,300 lb per sq in. calculated for the constant stress-strain ratio represented by the straight line OH having a slope equal to the modulus of elasticity E. The actual stress-strain curve deviates from the straight line OH; so that for the same strain, the actual stress is 19,800 lb per sq in. as represented by the point H'. In this way, the calculated stresses in Figs. 1, 2, and 3 were reduced to more probable values whenever they fell on the straight line OH above the actual stress-strain diagram OH'.

It may be questioned whether this construction is entirely correct due to the fact that the actual stress-strain diagram was determined from experimental data obtained by loading a test specimen in one direction only, while in the boiler tube under discussion stresses exist in three directions at right angles. One of the three stresses, however, is very small, namely, the radial stress. Under this condition it is probable that the proportional limit for the circumferential tensile stress is but little affected by the axial tensile stress, although the strains are all reduced in a nearly constant ratio as determined by Poisson's ratio and a nearly proportionate relation between the circumferential and axial stresses. Such a distortion of the stress-strain diagram does not affect the accuracy of the construction.

The upper curve of Fig. 7 has been reproduced from the calculated circumferential-stress curve of Fig. 3 for a heat-absorption rate of 90,000 Btu per hr per sq ft. The point A' was plotted at 19,800 lb per sq in. vertically below the point A representing the calculated stress of 26,300 lb per sq in. By a similar procedure to that explained above for determining the point A' corresponding to the point A, other points on the curve $A'B$ were obtained. Below the point B there is no appreciable plastic flow of the material. The new curve $A'BC$, however, does not fulfil the condition

$$\int_{r_i}^{r_2} t dr = p_i r_i$$

But by moving the zero line down in order to increase the positive tensile stresses and decrease the negative compressive stresses, a new position is found where this condition is again fulfilled 689 lb per sq in. below the old position.

Replotting the circumferential-stress curve to the new zero line in Fig. 8 raises the maximum stress from 19,800 lb per sq in. to 20,489 lb per sq in. as represented by the point A'. Referring to the stress-strain diagram of Fig. 4, it is seen that plastic flow will lower this stress to 19,900 lb per sq in. Similarly, the other stresses along the line $A'B$ will be reduced to those represented by $A''B$. This necessitates a further lowering of the zero line in order to meet the condition

$$\int_{r_i}^{r_2} t dr = p_i r_i$$

But as this lowering is only 82 lb per sq in., no further corrections are considered necessary for plastic flow.

The final circumferential-stress curve is plotted in Fig. 9 as $A''BC'$ in comparison with the original calculated curve AC. A

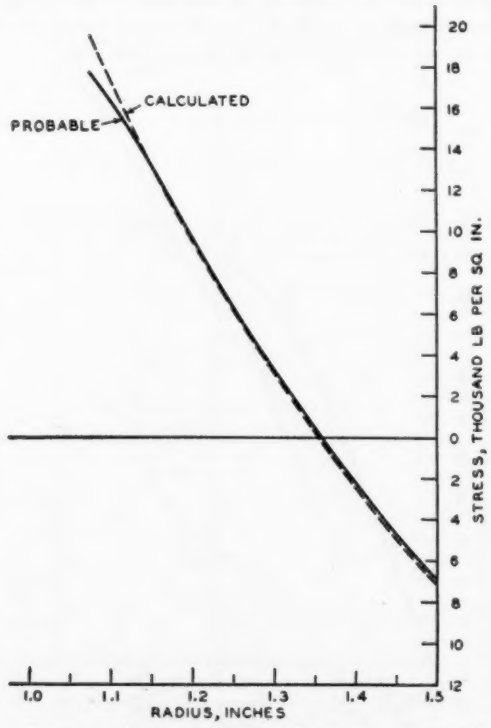


FIG. 5 PROBABLE CIRCUMFERENTIAL STRESSES IN TUBE WALL FOR HEAT ABSORPTION OF 60,000 BTU PER HR PER SQ FT

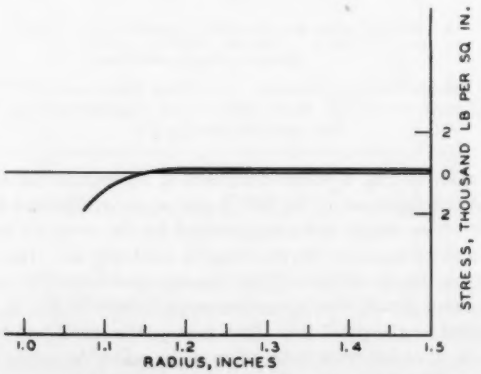


FIG. 6 RESIDUAL CIRCUMFERENTIAL STRESSES IN TUBE WALL FOR HEAT ABSORPTION OF 60,000 BTU PER HR PER SQ FT

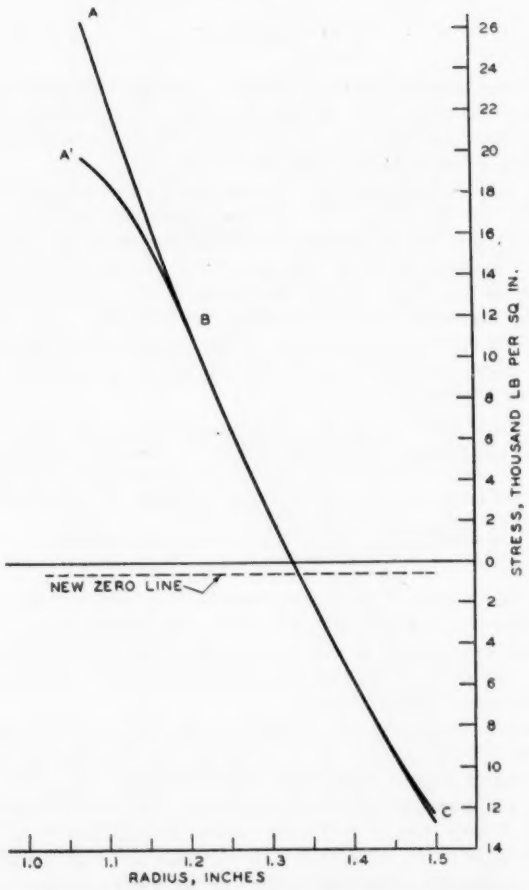


FIG. 7 FIRST CORRECTION OF CIRCUMFERENTIAL STRESSES IN TUBE WALL FOR HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

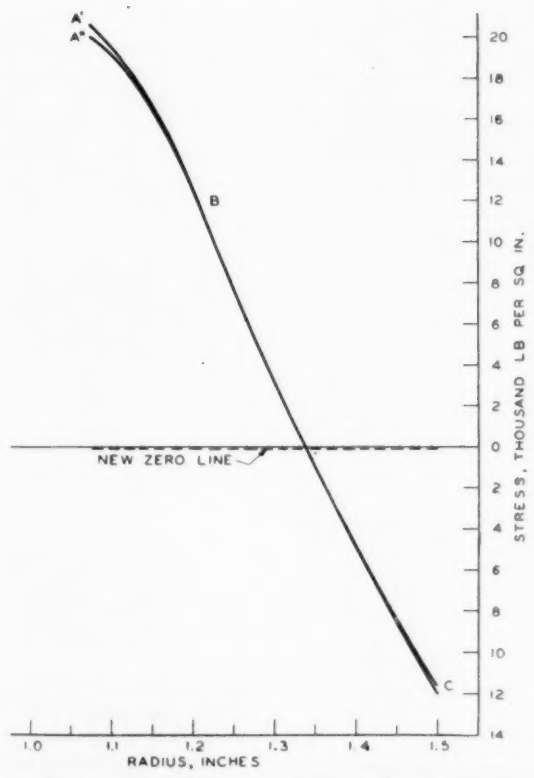


FIG. 8 SECOND CORRECTION OF CIRCUMFERENTIAL STRESSES IN TUBE WALL FOR HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

comparison of these two curves shows that the probable maximum circumferential tensile stress at the inside surface of the tube wall is about 6300 lb per sq in. less than the calculated stress for perfect elasticity, while farther within the tube wall the probable stresses are about 800 lb per sq in. higher than calculated. In other words, when portions of the tube wall tend to be stressed above the proportional limit of the material, the stresses in these portions are reduced by plastic flow below the values calculated on the assumption of perfect elasticity. A considerable reduction in tensile stress adjacent to the inside surface of the tube wall is balanced by a small additional tensile stress distributed over the remaining portion of the tube wall where the proportional limit of the material is not exceeded.

In a similar way the modifications in the circumferential stresses due to plastic flow adjacent to the inside surface of the tube wall were obtained for the heat-absorption rate of 60,000 Btu per hr per sq ft. The results are shown in Fig. 5. No modifications were made in the stresses for the heat-absorption rate of 30,000 Btu per hr per sq ft because the maximum stresses exceeded but slightly the proportional limit of the material at the operating temperatures.

RESIDUAL CIRCUMFERENTIAL STRESSES DUE TO PLASTIC FLOW

If a material is subjected to plastic flow by being stressed above the elastic limit, it takes a permanent set; that is, the material will not come back to its original length when the forces producing the stress are removed. In a tube wall subjected to high rates of heat absorption, plastic flow and consequently a permanent set occur only in portions of the material near the inner and outer surfaces. Therefore residual stresses will be produced in the material when heat transfer ceases and the internal fluid pressure is reduced to zero.

This may be seen by referring to Fig. 4. When a tensile stress corresponding to the point H' above the elastic limit of the material is removed, the strain will not retrace the line $H'GO$ but will trace the line $H'I$ parallel to GO. The strain OI is the permanent set remaining in the material when the stress is reduced to zero. In order to return the material to its original length, a compressive stress equal to OJ must be applied. The deformation OL was originally produced by the stress OK; but to return the material to its original length requires a change in stress equal to KJ.

In a tube wall the tensile stress at the inside surface follows the line $H'IJ$ when heat transfer ceases and the internal fluid pressure is reduced to zero. Farther within the tube wall, however, where the material was not strained beyond the elastic limit, the stress returns along the line GO. When the stress at the inside surface has reached zero at the point I, other portions of the tube wall are still in tension. Consequently the portion of the tube wall adjacent to the inside surface will be subjected to a compressive stress by other portions of the tube wall approaching zero stress. Equilibrium will occur when the summation of the compressive stresses set up adjacent to the inside surface just balances the summation of the tensile stresses remaining in other portions of the tube wall.

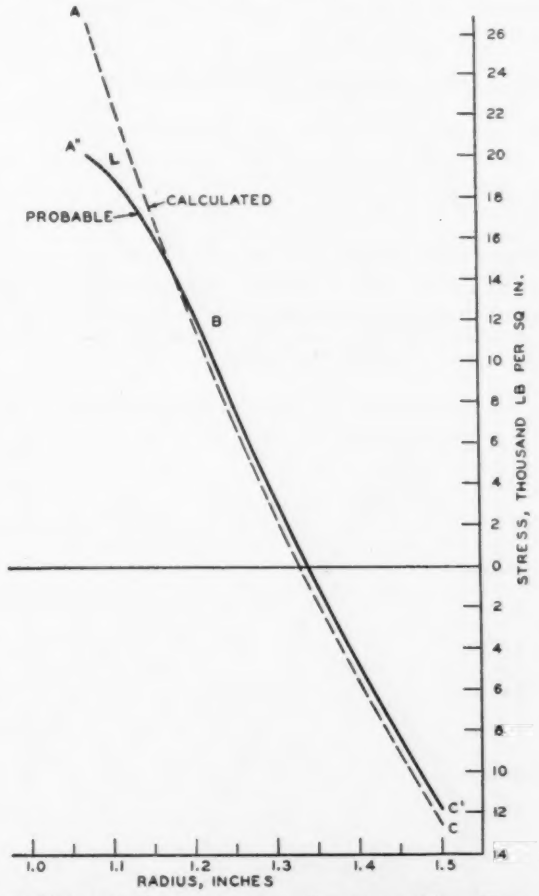


FIG. 9 PROBABLE CIRCUMFERENTIAL STRESSES IN TUBE WALL FOR HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

The residual stresses in the boiler tube under discussion for the heat-absorption rate of 90,000 Btu per hr per sq ft were calculated from the curves in Fig. 9 by first assuming all portions of the tube wall to return to their original lengths and then adjusting the zero line to balance the compressive and tensile stresses. For illustration, consider point L in Fig. 9 at a stress

of 19,800 lb per sq in. which is represented by point H' on the stress-strain diagram of Fig. 4. When the material at this point in the tube wall is returned to its original length, the stress is represented by OJ, or 6600 lb per sq in. in compression. This stress was plotted at point M in Fig. 10. Several additional points obtained in the same manner gave the broken-line curve. In order to make the summation of the residual stresses equal to zero, this curve was moved up 800 lb per sq in. to the position shown by the solid line. The solid line indicates a residual compressive stress of 6300 lb per sq in. at the inside surface of the tube wall. At a distance of 0.09 in. from the inside surface, the residual stress is zero. Over a large portion of the tube wall there is a residual tensile stress of 800 lb per sq in.

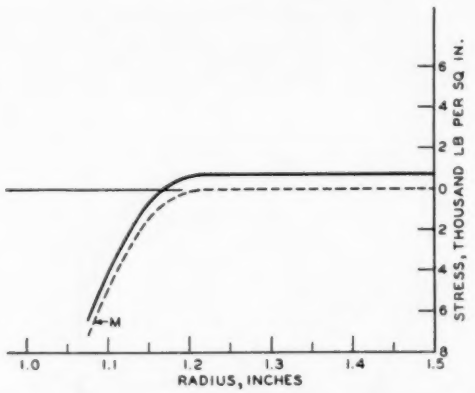


FIG. 10 RESIDUAL CIRCUMFERENTIAL STRESSES IN TUBE WALL FOR HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

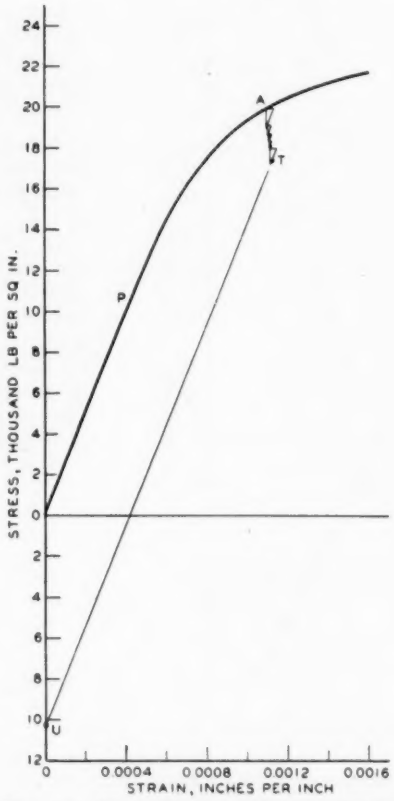


FIG. 12 STRESS-STRAIN DIAGRAM ILLUSTRATING CREEP

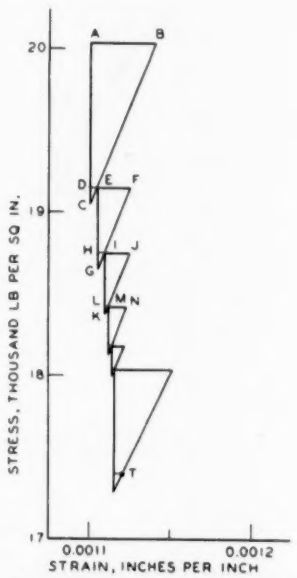


FIG. 13 ENLARGEMENT OF PORTION OF STRESS-STRAIN DIAGRAM SHOWN IN FIG. 12

Dividing 800 lb per sq in. by the modulus of elasticity of the material, 24.7×10^6 lb per sq in., gives 32.4×10^{-6} in. per inch as the permanent deformation of the tube as a whole. The outside diameter of the tube will therefore be increased from 3 in. to $3 + (3 \times 32.4 \times 10^{-6}) = 3.000097$ in. This increase in diameter will increase the stress due to internal fluid pressure, but the change is so small as to be negligible.

In Fig. 6 are shown the residual stresses for a heat-absorption rate of 60,000 Btu per hr per sq ft obtained in a manner similar to that explained for the higher heat-absorption rate. For a heat-absorption rate of 30,000 Btu per hr per sq ft, the residual stresses are negligible.

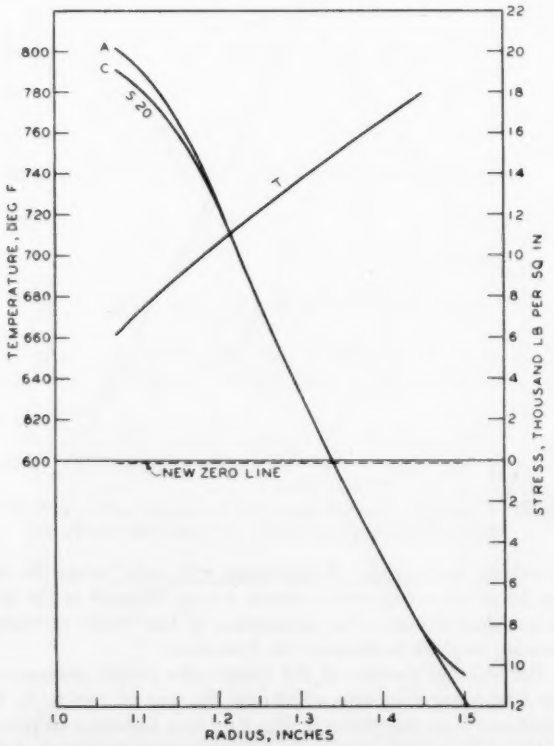


FIG. 11 REDISTRIBUTION OF CIRCUMFERENTIAL STRESSES IN TUBE WALL AFTER 20 HOURS' OPERATION AT HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

EFFECT OF CREEP UPON CIRCUMFERENTIAL STRESSES

The plastic flow above the elastic limit represented by the upper part of the stress-strain diagram of Fig. 4 corresponds to the comparatively rapid rate of loading used in an ordinary tension test of material. When the load on a test specimen is maintained for a long period, additional plastic flow will occur if the stress is above the creep-limit stress corresponding to the temperature of the material. The rate of creep in inches per inch per hour will vary for mild steel with the stress and the temperature somewhat as shown in Fig. 39 of Appendix No. 2. Sometimes this secondary plastic flow is called secondary creep and the first plastic flow is called primary creep. In speaking of creep in this paper, secondary plastic flow will be meant.

By reference to Fig. 39 in Appendix No. 2, it will be seen that the circumferential tensile stress near the inside surface of the tube wall is above the creep limit of the material for the heat-absorption rate of 90,000 Btu per hr per sq ft. Actual flow of the material is prevented, however, by adjacent portions of the tube wall which are not stressed beyond the creep limit. What happens to the stress under these conditions? Creep tests are conducted by subjecting a test specimen to a constant stress

and measuring the elongation at intervals of time. No tests have apparently been made in which the elongation was maintained constant and the variable stress measured. The stress, however, would undoubtedly decrease with time.

The following method was employed to calculate the reduction in stress with time in those portions of the tube wall where the stress exceeded the creep limit stress. The redistribution of

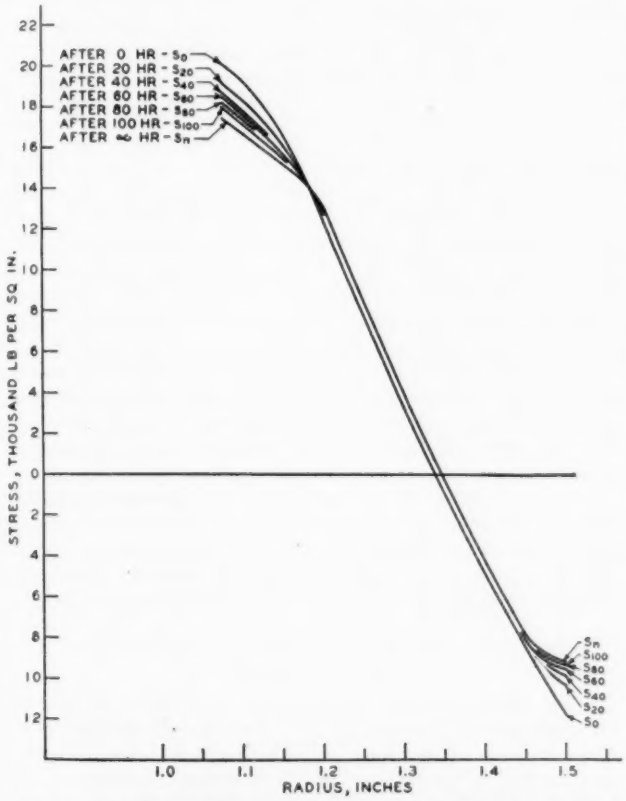


FIG. 14 REDISTRIBUTION OF CIRCUMFERENTIAL STRESSES IN TUBE WALL AFTER VARIOUS PERIODS OF OPERATION AT HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

stress after 20 hr of operation was first calculated. Referring to Fig. 11, the upper curve for zero hours of operation was reproduced from Fig. 9. At point *A*, the stress is 20,031 lb per sq in. after the initial plastic flow has occurred, but before creep has taken place. The temperature at this point in the tube wall is 662 F as shown by the temperature curve. Referring to Fig. 39 in Appendix No. 2, it will be seen that if the stress were kept at 20,031 lb per sq in., creep would occur at the rate of 0.000002 in. per inch per hr. For 20 hr the total creep would be 0.000040 in. per inch as represented by the line *AB* in Fig. 13. If this creep actually occurred under constant stress and the stress were then reduced until the material reached its original length, the reduction in stress would follow the line *BC* parallel to the portion *PO* of the stress-strain diagram below the proportional limit of the material. The point *C* directly under the original point *A* would represent the stress when the material reached its original length. This point was taken to represent the stress which would be reached after 20 hr if the elongation were kept constant. The reduction in stress represented by *AC* may be calculated from the deformation *AB* and the modulus of elasticity of the material; thus, $AC = 0.000040 \times 24.7 \times 10^6 = 988$ lb per sq in. Hence point *C* in Fig. 11 is 20,031 — 988 = 19,043 lb per sq in.

The stresses after 20 hr at several other points in the tube wall where the original stresses exceeded the creep limit stresses, were calculated in a similar manner and plotted on Fig. 11. For the curve so obtained, however, the summation of the stresses across the tube wall do not balance the internal fluid pressure along a radius. To obtain this balance, the zero line was moved down 97 lb per sq in.

This depression of the zero line is represented by the distance *CD* in Fig. 13. The tube as a whole expands an amount *DE* inch per inch due to reduction in stress near the inside surface by creep and consequent increase of stress over the remaining portions of the tube wall. This expansion may be calculated from the stress increase of 97 lb per sq in. and the modulus of elasticity of the material, thus, $97/(24.7 \times 10^6) = 0.0000039$ in. per inch. The increase in outside diameter during 20 hr would be $3 \times 0.0000039 = 0.0000117$ in.

During the second 20 hr, creep will take place in a similar manner to that explained for the first 20 hr, but at a slower rate, due to the reduction in stress which has occurred during the first 20 hr. The calculations for the second 20 hr are represented by the stress-strain diagram *EFGHI* in Fig. 13. Calculations for additional intervals of 20 hr were made until a total interval of 100 hr was reached. The resulting stress-distribution curves are shown in Fig. 14, the various time intervals being denoted by s_0, s_{20}, etc. The change in stress from 80 to 100 hr is very small, indicating that the creep-limit stress has nearly been reached in the portions of the tube wall where the stresses exceed this limit. The next step taken was to determine the stress-distribution curve s_n at the creep limit, which would be reached only after an infinite interval of time. This was done by reducing all stresses for the s_{100} curve where they were above

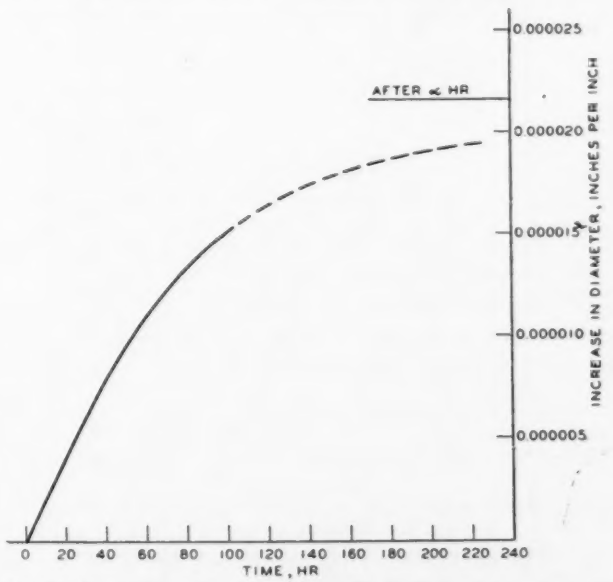


FIG. 15 INCREASE IN TUBE DIAMETER WITH TIME OF OPERATION AT HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

the creep limit to the creep limit, and then adjusting the zero line. The increase in outside diameter of the tube as the higher stresses are reduced by creep is shown in Fig. 15.

RESIDUAL CIRCUMFERENTIAL STRESSES AFTER CREEP

Owing to creep in those portions of the tube wall where the stresses exceeded the creep-limit stress of the material, the residual stresses after heat transfer ceases and the internal fluid pressure is removed will be altered from those shown in Fig. 10

for conditions before creep has taken place. The residual stresses corresponding to the creep-limit curve in Fig. 14 are shown by the solid line in Fig. 16. The broken-line curve is reproduced from Fig. 10 and shows the residual stresses before creep has occurred. The solid-line curve was obtained in a

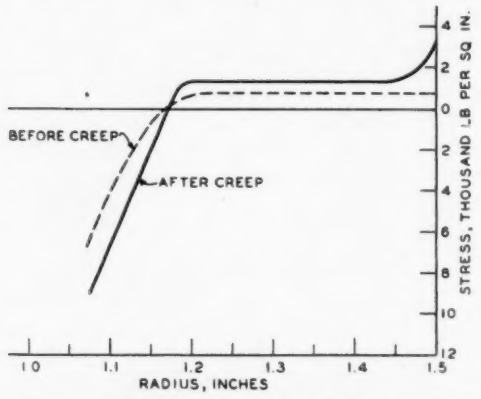


FIG. 16 RESIDUAL CIRCUMFERENTIAL STRESSES IN TUBE WALL AFTER OPERATION FOR INFINITE TIME AT HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

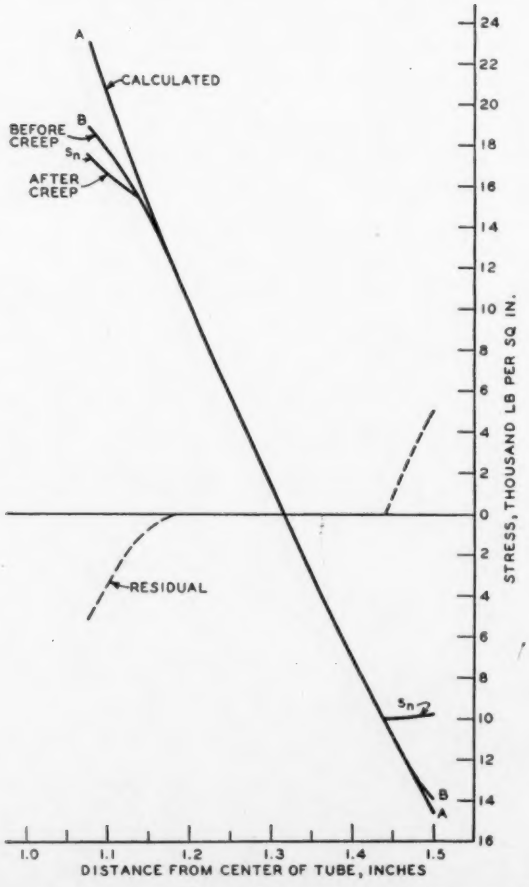


FIG. 17 AXIAL STRESSES IN TUBE WALL FOR UNIFORM HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

manner similar to that explained for the broken-line one. Thus, point *T* in Fig. 12 represents the condition in the material at the inner surface of the tube wall after the creep-limit stress has been reached. As heat transfer ceases and the internal fluid pressure is produced to zero, the stress and strain will follow the line *TU* drawn parallel to *PO*. The point *U* will represent the compressive stress required to return the material to its original length. A curve was first drawn for the residual stresses, assuming all portions of the tube wall to be returned to their original lengths. The zero line was then adjusted to obtain a balance between the tensile and compressive stresses. The final location of the residual-stress curve was thus obtained as shown by the solid line in Fig. 16.

PROBABLE AND RESIDUAL AXIAL STRESSES IN A THICK BOILER TUBE

At the heat-absorption rate of 30,000 Btu per hr per sq ft the calculated axial stresses in Fig. 1 are all within the proportional limit of mild steel. At 60,000 Btu per hr per sq ft the calculated axial stresses in Fig. 2 exceed the proportional limit near the inside surface of the tube wall, but corrections for plastic flow were applied only to the axial stresses calculated for 90,000 Btu per hr per sq ft in Fig. 3. The axial-stress gradients in the tube wall are shown by the curves in Fig. 17. The curve *AA* represents the axial stresses as calculated for perfect elasticity. The curve *BB* represents the axial stresses as modified by the initial plastic flow of the material. The curve $s_n s_n$ represents the axial stresses after a long period of time when secondary plastic flow has occurred to reduce all stresses above the creep limit to the creep-limit stresses. The residual axial stresses after creep has occurred are represented by the broken lines. These modifications in the calculated stresses due to plastic flow were made in the same manner as explained for the circumferential stresses. It happens, however, that no adjustment of the zero line was necessary because the reduction in tensile stresses just balanced the reduction in compressive stresses.

STRESSES IN A BOILER TUBE SUBJECTED TO NON-UNIFORM HEAT ABSORPTION

When the rate of heat absorption is not uniform around the circumference of a boiler tube, as is the case with the first rows of boiler tubes subjected to radiation on one side only from the burning fuel in the furnace, the circumferential stresses at the place of maximum heat absorption will evidently not differ greatly from those determined as if the maximum rate of heat absorption were uniform around the tube. The axial stress, however, will be increased under certain conditions of non-uniform heat absorption and may exceed the circumferential stress under abnormal conditions.

Thus, when heat is absorbed at a high rate on the furnace side of a boiler tube and at a low rate on the opposite side, the average metal temperature on the furnace side will be greater than on the opposite side. This causes a greater expansion on the furnace side of the tube than on the other side, so that the tube will bend into an arc of a circle convex to the furnace. If restrained from bending, axial stresses will be set up due to the differences in expansion on the two sides of the tube caused by the unequal temperature rises. These axial stresses will be in addition to those calculated by assuming a uniform rate of heat absorption around the tube. If the tube is allowed to bend, these additional axial stresses will be more or less relieved.

The following analysis of the axial stresses in a boiler tube subjected to non-uniform heat absorption will be based on assumed absorption rates of 30,000, 60,000, and 90,000 Btu per hr per sq ft around 180 deg of the circumference while the remaining 180 deg receives no heat. The results will then be modified to show the effects of a less abrupt change from the maximum to the minimum rate of heat absorption. Under these assumptions, the temperatures in the tube wall will be as follows:

Rate of heat absorption, Btu per hr per sq ft	Furnace side of tube: Temperature of inside surface, deg F	Furnace side of tube: Temperature of outside surface, deg F	Rear side of tube: Temperature of whole wall, deg F
30,000	634.0	676.5	620
60,000	647.9	734.1	620
90,000	661.9	792.5	620

The axial stresses in the tube wall will first be calculated for the tube before any bending has taken place. These axial stresses will tend to bend the tube, and their bending moment will be determined. It will then be assumed that the tube bends until these bending moments are reduced to zero, and the redistribution of stresses will be found. Corrections will be made for plastic flow of the material in certain portions of the tube wall, and the residual axial stresses will be determined.

Calculated Axial Stresses Before Bending in a Thick Boiler Tube Heated on One Side Only

The larger longitudinal expansion due to the greater temperature rise on the furnace side of a boiler tube is resisted by the smaller expansion on the opposite side. This sets up compressive stresses in the tube wall on the furnace side and tensile stresses in the tube wall on the opposite side. The summation of the tensile stresses will exceed the summation of the compressive stresses by the total axial internal fluid pressure.

The following formulas were derived in Appendix No. 1, for calculating the axial stresses in the tube wall on the furnace side and on the opposite side of a boiler tube absorbing heat through 180 deg only:

$$s_1 = A - s_3$$

$$s_2 = B - s_3$$

$$s_3 = \frac{r_1^2 p_1}{r_2^2 - r_1^2} + \frac{A}{6}\frac{r_2 + 2r_1}{r_2 + r_1} + \frac{B}{6}\frac{2r_2 + r_1}{r_2 + r_1}$$

$$A = (T_1 - T_3)\alpha_1 E_1 - t_1/m$$

$$B = (T_2 - T_3)\alpha_2 E_2 + t_2/m$$

$$t_1 = (T_2 - T_1)\alpha E/2$$

$$t_2 = -(T_2 - T_1)\alpha E/2$$

where r_1 = inside radius of tube wall, inches

r_2 = outside radius of tube wall, inches

p_1 = internal fluid pressure, lb per sq in. (outside fluid pressure being zero)

T_1 = temperature at inside surface of tube wall on furnace side, deg F

T_2 = temperature at outside surface of tube wall on furnace side, deg F

T_3 = temperature of tube wall on opposite to furnace side, deg F

E_1 = modulus of elasticity corresponding to T_1, lb per sq in.

E_2 = modulus of elasticity corresponding to T_2, lb per sq in.

E = modulus of elasticity corresponding to $(T_1 + T_2)/2$, lb per sq in.

α_1 = coefficient of thermal expansion corresponding to T_1, inch per inch per deg F

α_2 = coefficient of thermal expansion corresponding to T_2, inch per inch per deg F

α = coefficient of thermal expansion corresponding to $(T_1 + T_2)/2$, inch per inch per deg F

$1/m$ = Poisson's ratio

s_1 = axial compressive stress at r_1 on furnace side, lb per sq in.

s_2 = axial compressive stress at r_2 on furnace side, lb per sq in.

s_3 = axial tensile stress in tube wall on opposite to furnace side, lb per sq in.

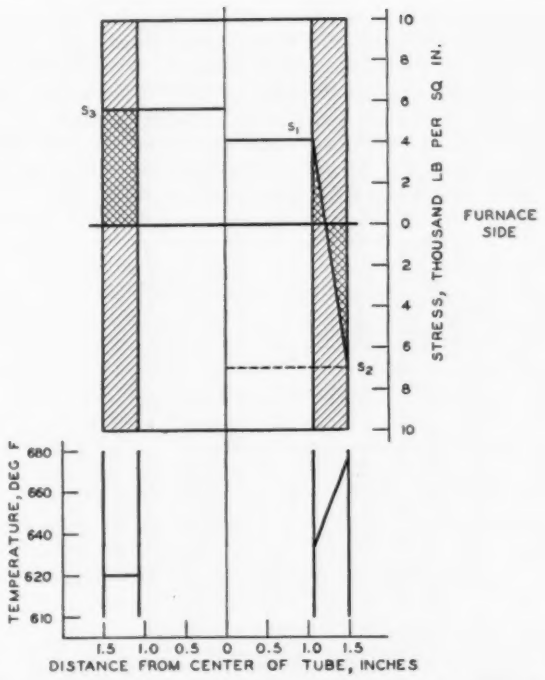


Fig. 18 Axial Stresses in Tube Wall Before Bending for Non-Uniform Heat Absorption of 30,000 Btu per Hr per Sq Ft

To illustrate the use of the above formulas, a numerical example will be worked out for a heat-absorption rate of 30,000 Btu per hr per sq ft. Under these conditions, we have

r_1 = 1.075 in.

r_2 = 1.5 in.

p_1 = 1,840 lb per sq in.

T_1 = 634.0 F

T_2 = 676.5 F

T_3 = 620.0 F

E_1 = 25.8×10^6 lb per sq in. from Fig. 35

E_2 = 25.3×10^6 lb per sq in. from Fig. 35

E = 25.5×10^6 lb per sq in. from Fig. 35

α_1 = 7.86×10^{-6} inch per inch per deg F from Fig. 40

α_2 = 7.97×10^{-6} inch per inch per deg F from Fig. 40

α = 7.92×10^{-6} inch per inch per deg F from Fig. 40

$1/m$ = 0.30

$$t_1 = \frac{676.5 - 634.0}{2} \times 7.92 \times 10^{-6} \times 25.5 \times 10^6 = 4292$$ lb per sq in.

t_2 = —4292 lb per sq in.

A = $(634.0 - 620) \times 7.86 \times 10^{-6} \times 25.8 \times 10^6 - 0.30 \times 4292 = 1551$ lb per sq in.

B = $(676.5 - 620) \times 7.97 \times 10^{-6} \times 25.3 \times 10^6 + 0.30 \times 4292 = 12{,}575$ lb per sq in.

s_3 = $1943 + 0.237A + 0.264B = 5630$ lb per sq in.

s_1 = $1550 - 5630 = -4079$ lb per sq in.

s_2 = $12{,}575 - 5630 = 6945$ lb per sq in.

From the manner in which the above equations were derived, s_3 is a tensile stress and s_1 and s_2 are compressive stresses when preceded by a plus sign. The fact that s_1 works out with a negative sign shows that it is a tensile stress.

The distribution of the axial stresses in the tube before bending is shown in Fig. 18 for the heat-absorption rate of 30,000 Btu per hr per sq ft. The figure represents a longitudinal section through the tube with the furnace on the right-hand side. Tensile stresses have been plotted above the zero line

and compressive stresses below. On the furnace side, the axial stress in the tube wall varies from a compressive stress of s_2 = 6945 lb per sq in. at the outside surface to a tensile stress of s_1 = 4079 lb per sq in. at the inside surface. These axial stresses extend half-way around the tube as indicated by the horizontal lines at s_1 and s_2. On the opposite side of the tube, the tensile stress is s_3 = 5630 lb per sq in. across the tube wall and around one-half of the tube as indicated by the horizontal line s_3.

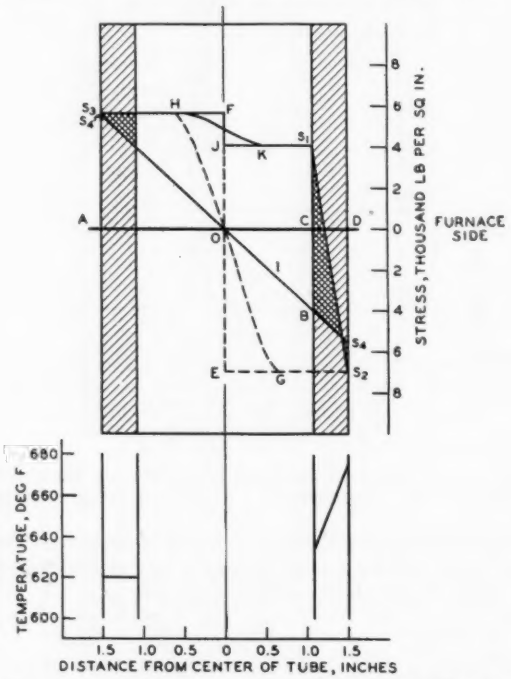


FIG. 19 AXIAL STRESSES IN TUBE WALL AFTER BENDING FOR NON-UNIFORM HEAT ABSORPTION OF 30,000 BTU PER HR PER SQ FT

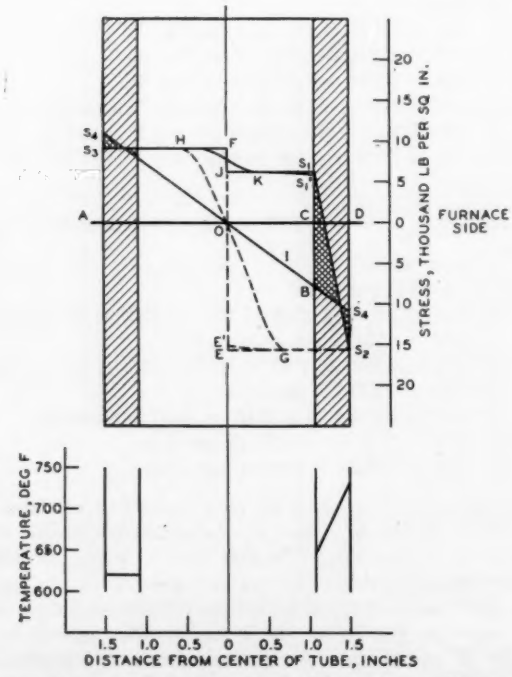


FIG. 20 AXIAL STRESSES IN TUBE WALL AFTER BENDING FOR NON-UNIFORM HEAT ABSORPTION OF 60,000 BTU PER HR PER SQ FT

The axial stresses before bending are shown for heat-absorption rates of 60,000 and 90,000 lb per sq in. by similar lines in Figs. 20 and 21. These figures have additional lines on them to show the modifications in these axial stresses due to bending of the tube, which will now be discussed.

AXIAL STRESSES AFTER BENDING IN A THICK BOILER TUBE HEATED ON ONE SIDE ONLY

That the stresses induced in a boiler tube by being heated on one side only tend to bend the tube convex toward the furnace is indicated in Fig. 18 by the moments of the stresses there shown being clockwise on both the furnace and the opposite sides. As the tube bends, the moments of these stresses will be reduced; and bending will cease when the moments of the stresses remaining are reduced to zero. As the tube bends, the axial stresses are relieved in proportion to their distances from the neutral axis of the tube cross-section. That is, bending is equivalent to superimposing upon the original stresses additional stresses varying from zero at the neutral axis to a maximum at the outermost fiber as represented by the line s_4s_4 in Fig. 19. If not restrained, the tube will bend to a curvature where the moments

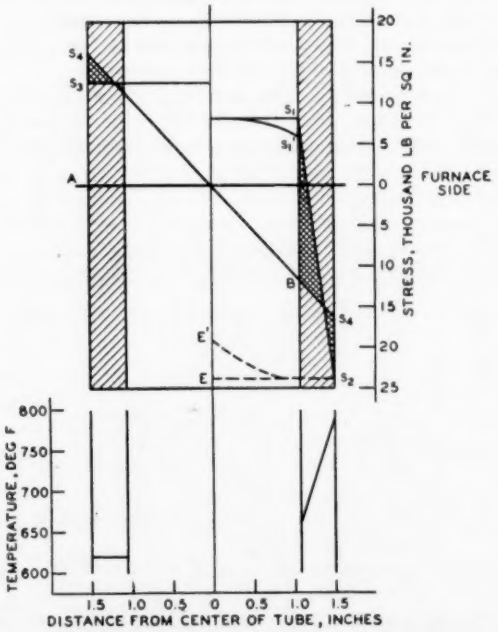


FIG. 21 AXIAL STRESSES IN TUBE WALL AFTER BENDING FOR NON-UNIFORM HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

of these superimposed stresses just balance the moments of the original stresses calculated for the tube before bending as previously explained.

The formulas derived in Appendix No. 1 for calculating the maximum stress at the outermost fiber superimposed by bending of an unrestrained tube upon the stresses before bending are as follows:

$$s_4 = \frac{4(M_1 + M_2)r_2}{\pi(r_2^4 - r_1^4)}$$

$$M_1 = 2\left[\frac{s_1(r_2^3 - r_1^3)}{3} + \frac{(s_2 - s_1)(r_2^4 - r_1^4)}{4(r_2 - r_1)} - \frac{(s_2 - s_1)r_1(r_2^3 - r_1^3)}{3(r_2 - r_1)}\right]$$

$$M_2 = \frac{2s_3(r_2^3 - r_1^3)}{3}$$

where r_1 = inside radius of tube wall, inches
r_2 = outside radius of tube wall, inches
s_1 = axial compressive stress before bending at r_1 on furnace side, lb per sq in.
s_2 = axial compressive stress before bending at r_2 on furnace side, lb per sq in.
s_3 = axial tensile stress before bending in tube wall on opposite to furnace side, lb per sq in.

For a heat-absorption rate on the furnace side of 30,000 Btu per hr per sq ft, we have previously calculated s_1 = —4079 lb per sq in., s_2 = 6945 lb per sq in., and s_3 = 5630 lb per sq in. Substituting these values and r_1 = 1.075 in. and r_2 = 1.5 in.,

M_1 = 2890 lb-in.
M_2 = 8005 lb-in.
s_4 = 5584 lb per sq in.

This value of s_4 has been plotted in Fig. 19 as a reduction in the compressive stress at the outermost fiber on the furnace side of the tube and as a reduction in the tensile stress at the outermost fiber on the opposite side of the tube. The straight line s_4s_4 represents the reductions in stress at all points in the tube wall and therefore constitutes a new base line from which to measure the stresses remaining after bending from those calculated for conditions before bending.

Thus, at the middle point of the furnace side of the tube, the compressive stress at the outside surface has been reduced by bending from Ds_2 to s_4s_2. This compressive stress at the outside surface after bending gradually increases from s_4s_2 as we pass around the tube until it reaches the original value OE at the center line where heat absorption was assumed to decrease suddenly to zero. At the middle of the opposite side of the tube, the tensile stress at the outside surface has been reduced by bending from As_2 to s_4s_2. This tensile stress at the outside surface after bending gradually increases from s_4s_2 as we pass around the tube until it reaches the original value OF at the center line where heat absorption was assumed to start suddenly. The stresses at the inside surface of the tube wall can be traced in a similar manner. The increase due to bending in tensile stress at the inside surface on the furnace side of the tube from Cs_1 to Bs_1 may be noticed. The stress gradients through the tube wall for the extreme front and rear portions of the tube are shown by the double cross-sectioning in Fig. 19. The stress gradient will evidently be different in each portion of the wall.

Actually, the heat-absorption rate will not remain the same on the furnace side of the tube up to the center line and then suddenly cease. Heat absorption may remain practically constant over a portion of the area exposed to the furnace, but it will gradually change from this value to zero or to a relatively low rate over the rear portion of the tube. Consequently, the stress before bending will not change suddenly as indicated in Fig. 19 from E to F at the center line of the tube, but will change more gradually along some line such as GH. At the inside surface, there will be a gradual change KH rather than the abrupt change JF. The maximum compressive stress at the outside surface after unrestrained bending will therefore be IG instead of OE.

All stresses in Fig. 19 for a heat-absorption rate of 30,000 Btu per hr per sq ft are within the proportional limit of mild steel. For the higher heat-absorption rate of 60,000 Btu per hr per sq ft, however, some of the stresses shown in Fig. 20 are above the proportional limit and should be reduced in the manner previously explained for circumferential stresses. This has been done, the lines $E'G$ and $s_1'K$ showing the stresses as reduced from EG and s_1K, respectively. These stress reductions upset the balance between the axial stresses in the tube wall and the internal fluid pressure, but the unbalance is so small that no change in the zero line is necessary and the stresses shown in Fig. 20 may be taken as representative of the axial stresses when one side only of the tube absorbs heat at the rate of 60,000 Btu per hr per sq ft.

When the rate of heat absorption is increased to 90,000 Btu per hr per sq ft, the axial stresses at some points in the tube wall shown in Fig. 21 are above the creep limit of the material as well as above the proportional limit. If the stresses at these points are reduced to the creep limit, the stresses at other points in the tube wall must be increased to balance the internal fluid pressure as previously explained for circumferential stresses. In order to make this adjustment, it was necessary to divide the furnace side of the tube wall radially into 32 sections and plot the stress gradient for each of the 16 pairs of similar sections on either side of the center. It was unnecessary to do this for the opposite side of the tube where the stresses are all within the creep limit and but slightly exceed the proportional limit at some points. The final result is shown in Fig. 22 with the new zero line. It may be mentioned that before reducing the stresses to the creep limit, the probable shift in the zero line was first estimated in order that a second reduction to the creep limit would not be necessary after the new zero line was determined.

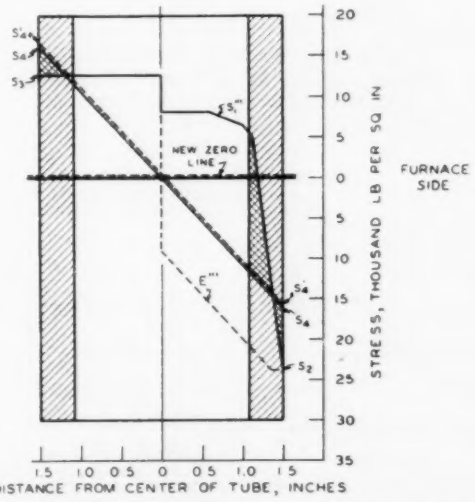


FIG. 22 AXIAL STRESSES IN TUBE WALL AFTER CREEP FOR NON-UNIFORM HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

RESIDUAL AXIAL STRESSES IN A THICK BOILER TUBE HEATED ON ONE SIDE ONLY

When heat transfer ceases and the internal pressure is reduced to zero, the boiler tube will tend to return to its original length. The portions which have not been stressed beyond the elastic limit will return very nearly to their original lengths and the residual stresses therein will be nearly zero. But in the portions of the tube wall where a permanent set has occurred due to plastic flow, the stresses will be reversed and high residual stresses may be produced. For a heat-absorption rate of 30,000 Btu per hr per sq ft there will be no residual axial stresses because the elastic limit of the material was not exceeded. For an absorption rate of 60,000 Btu per hr per sq ft there will be some residual axial stresses but these will be relatively small. For 90,000 Btu per hr per sq ft, however, the residual axial stresses are comparatively large. This case only will be given here.

The residual axial stresses for a heat absorption rate of 90,000 Btu per hr per sq ft on one side only of the tube were calculated in the same manner as previously explained for circumferential stresses. It was necessary to determine the residual stress for

each of the 32 sections into which the furnace side of the tube was divided from the stress gradient for each pair of sections, and then adjust the zero line for each section so as to balance the axial stresses over the whole cross-section of the tube wall with the total axial internal fluid pressure. The residual axial stresses at the inner and outer surfaces of the tube wall are plotted in Fig. 23 and Fig. 24, respectively.

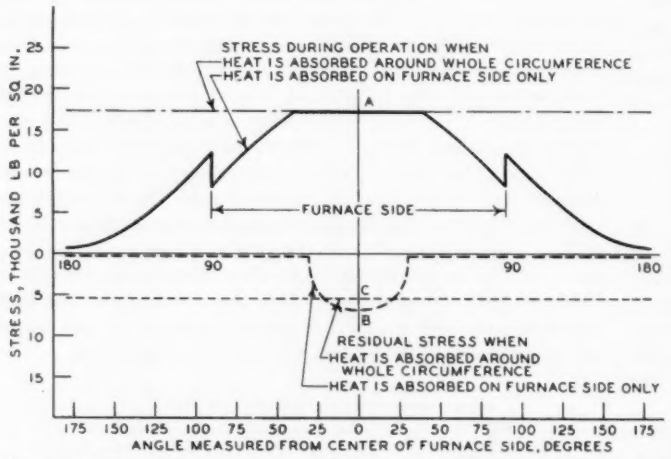


FIG. 23 DEVELOPMENT OF AXIAL STRESS AT INNER SURFACE OF TUBE WALL FOR NON-UNIFORM HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

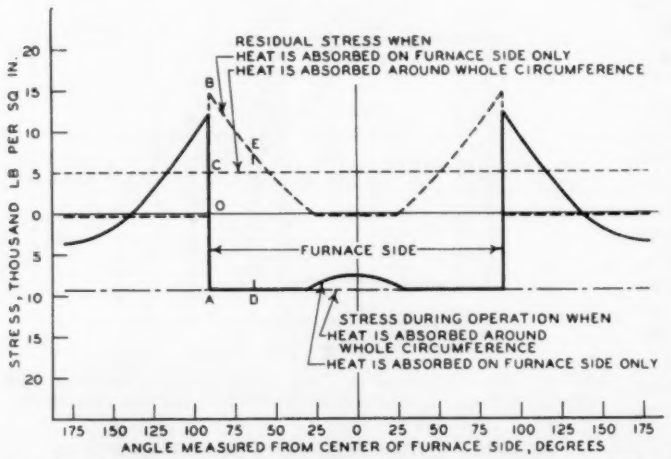


FIG. 24 DEVELOPMENT OF AXIAL STRESS AT OUTER SURFACE OF TUBE WALL FOR NON-UNIFORM HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT

COMPARISON OF AXIAL STRESSES IN A THICK BOILER TUBE HEATED ON ONE SIDE ONLY WITH THOSE FOR UNIFORM HEAT ABSORPTION

The maximum stresses in a boiler tube always occur at the inner and outer surfaces of the tube wall. Therefore plots of the stresses on developments of the inner and outer circumferences of the tube wall will show the maximum stresses in the tube. The axial stresses for the boiler tube heated on one side only are so plotted in Figs. 23 and 24 in comparison with the axial stresses for uniform heat absorption around the tube circumference.

Referring to Fig. 23, it is seen that when the tube is heated on one side only, the maximum axial stress occurs at the center of the furnace side. At this point the tensile stress during operation is 17,300 lb per sq in. and the residual compressive stress is 6900 lb per sq in., making a stress range of 24,200 lb per sq in. as shown by the distance AB. The distance AC shows that if the tube is heated uniformly around its circumference, the stress range is 22,800 lb per sq in. from a working tensile stress of 17,300 lb per sq in. to a residual compressive stress of 5500 lb per sq in. The working tensile stress is the same in both cases because this stress was assumed in both cases to reach the creep-limit stress of 17,300 lb per sq in. corresponding to the temperature of 662 F at the inner surface of the tube wall on the furnace side. When the tensile stress during operation tends to go much above the creep limit, the residual compressive stress will be correspondingly large. Therefore the stress range is a criterion of the stress conditions in the tube wall rather than the stress during operation only, and the metal would fail by reason of repeated reversals of stress rather than by simple tension or compression.

Referring to Fig. 24, at the outer surface of the tube, the maximum stress range for uniform heat absorption is AC = 14,400 lb per sq in. from a working compressive stress of 9400 lb per sq in. to a residual tensile stress of 5000 lb per sq in. For heat absorption on one side only, the maximum stress range is AB = 24,000 lb per sq in. from the same working compressive stress of 9400 lb per sq in. to a residual tensile stress of 14,600 lb per sq in. This stress range occurs at the point where the rate of heat transfer was assumed to decrease abruptly from 90,000 Btu per hr per sq ft to zero. As in practice the rate of heat transfer will occur gradually rather than abruptly, the maximum stress range for non-uniform heating will be less than 24,000 lb per sq in. The maximum stress range will occur where the rate of heat transfer begins to fall off as we pass around the tube from the furnace side. If this point is 65 deg from the middle point of the furnace side rather than 90 deg as above assumed, the maximum stress range will be reduced to 16,600 lb per sq in. As heat absorption becomes more nearly uniform around the circumference of the tube the maximum stress range at the outer surface of the tube wall will continue to decrease and will approach as a limit the stress range for uniform heat absorption.

When the heat absorption falls off gradually around the tube rather than abruptly, the moments of the resultant stresses about the vertical axis before bending will be less than previously calculated, thereby reducing the stress s_4 in Fig. 22. This will decrease the maximum stress range at the inner surface of the tube wall represented in Fig. 23. As the heat absorption becomes more nearly uniform, the maximum stress range AB will decrease and approach as a limit the stress range AC for uniform heat absorption.

FATIGUE STRENGTH OF THICK BOILER TUBES

That the mere stressing of a boiler tube beyond the elastic limit of the material near the inner and outer surfaces of the tube wall is not injurious to the tube becomes evident when we consider that stressing beyond the elastic limit is done purposely in many manufacturing operations of rolling, drawing, and spinning in order to improve the material by raising its elastic limit and ultimate strength. It is well known, for example, that cold-drawn wire has a higher elastic limit and a greater tensile strength than the ingot from which it was made. Helical steel springs are purposely overloaded in order to stress a portion of the material beyond the elastic limit and give it a permanent set so that the springs will thereafter bear a greater load without permanent deflection and without failure by fatigue than they would if they had not been so overloaded. In boiler

tubes subjected to high rates of heat absorption, this improvement in the material is automatically accomplished by the conditions of service rather than requiring a separate manufacturing operation to accomplish it.

Referring to Fig. 4, the portion of the mild steel in the tube wall which has been loaded beyond the elastic limit to the point H', now follows the line $H'J$ when the load is released. This portion of the material now has an elastic limit H' higher than the original elastic limit at G, and so long as the working stress does not exceed H', there will be elastic deformation of the material in proportion to the stress along the line IH'. Due to the permanent set acquired by certain portions of the tube wall being stressed beyond the elastic limit to H' and to other portions of the tube wall not being stressed beyond the elastic limit G, the stress in these certain portions will be reversed when heat transfer ceases and the internal fluid pressure is reduced to zero.

At the inner surface of the tube wall, the circumferential stress varies from 20,000 lb per sq in. tension to 6300 lb per sq in. compression before creep has occurred, and from 17,400 lb per sq in. tension to 9000 lb per sq in. compression after creep has occurred to the creep-limit stress for a heat-absorption rate of 90,000 Btu per hr per sq ft. These two ranges of circumferential stress variation are compared in Fig. 25 with test data on the stress ranges which may be repeated ten million times without failure of mild steel as explained in Appendix No. 2.

A similar comparison is made in Fig. 26 for the ranges in axial stress variation due to both uniform and non-uniform heat absorption around the tube at a rate of 90,000 Btu per hr per sq ft. With uniform heat absorption, the stress range at the inner surface of the tube wall is from 17,300 lb per sq in. tension to 5500 lb per sq in. compression, and the stress range at the outer surface of the tube wall is from 5000 lb per sq in. tension to 9400 lb per sq in. compression. When the heat absorption is non-uniform, the maximum possible ranges theoretically occur when the heat absorption is assumed to change abruptly from 90,000 Btu per hr per sq ft to zero, and these ranges are: On the inner surface of the tube wall, 17,300 lb per sq in. tension to 6900 lb per sq in. compression; on the outer surface of the tube wall, 14,600 lb per sq in. tension to 9400 lb per sq in. compression. Actually, these latter ranges will approach the former ranges with a gradual change in the rate of heat absorption around the tube.

By reference to Figs. 25 and 26, it is seen that the maximum stress ranges in a thick boiler tube subjected to a very high rate of heat absorption are all within ranges withstood ten million times without failure during test. If the boiler tube passed through one cycle per day, it would require

27,400 years to complete	10,000,000 cycles
2,740 years to complete	1,000,000 cycles
274 years to complete	100,000 cycles

Thus, even if at this slow rate of reversal, the number of reversals permissible should be much less than that found by test with a rapid rate of stress reversal, boiler tubes should last many years so far as any stresses due to a high rate of heat absorption are concerned.

Bursting of Thick Boiler Tubes by Internal Pressure

The foregoing discussion has shown that a boiler tube will not fail within any reasonable period of time by fatigue due to repeated reversals of the stresses set up by high rates of heat absorption either uniformly or non-uniformly disposed around the tube circumference. Before the tube can fail by bursting, all portions of the tube wall must be stressed to a point where more or less rapid creep will occur to rupture. Before this can take place, all heat-transfer stresses will be entirely neutralized by creep. Failure of the tube will then be caused by the stresses due to internal fluid pressure only. These facts may be shown by reference to Figs. 27, 28, and 29.

While the boiler tube in Figs. 27, 28, and 29 is of the same dimensions as in the previous discussion, the steam pressure is

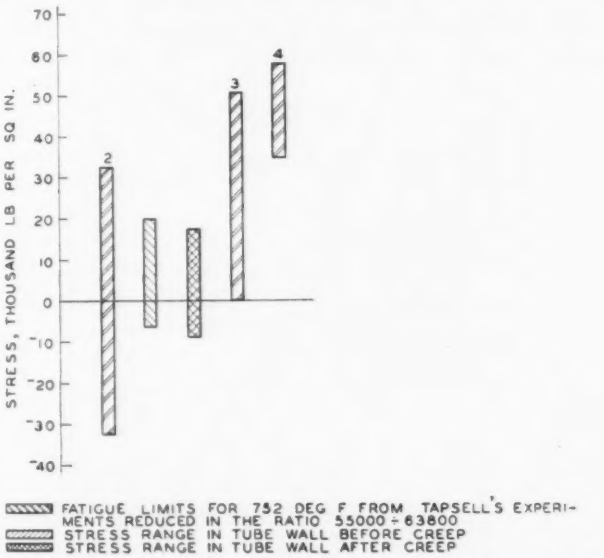


Fig. 25 Comparison of Circumferential-Stress Ranges for Heat Absorption of 90,000 Btu per Hr per Sq Ft With Fatigue-Test Data

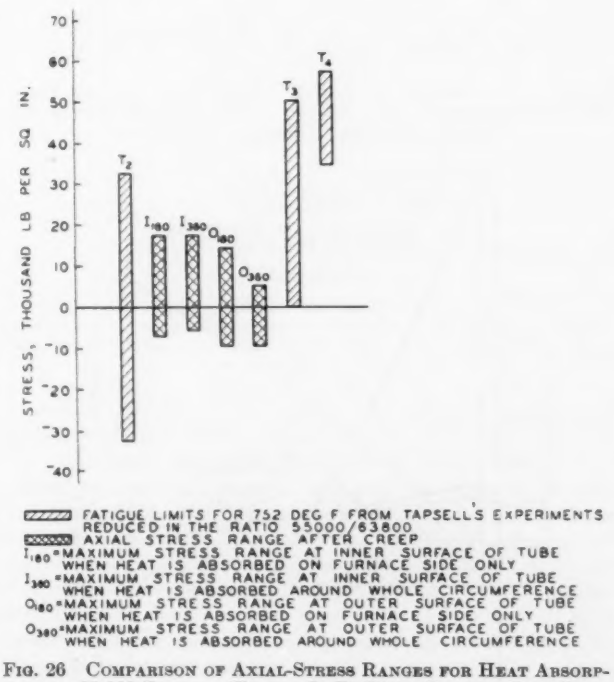


Fig. 26 Comparison of Axial-Stress Ranges for Heat Absorption of 90,000 Btu per Hr per Sq Ft With Fatigue-Test Data

1390 lb gage instead of 1840 lb gage for the reason that these figures were originally made for a study of a different installation than the preceding figures. The circumferential stresses only have been considered because, as shown previously, the axial stresses are less than the circumferential stresses except with an abnormally non-uniform heat absorption around the tube.

Referring to Figs. 27, 28, and 29, the difference of the areas

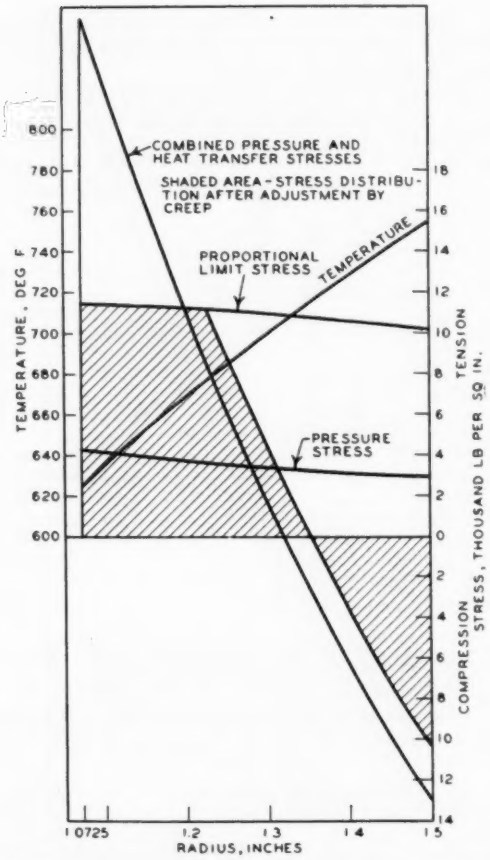


FIG. 27 CIRCUMFERENTIAL STRESS DISTRIBUTION IN TUBE WALL FOR HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT WITH 41.9 F TEMPERATURE DROP THROUGH WATER FILM

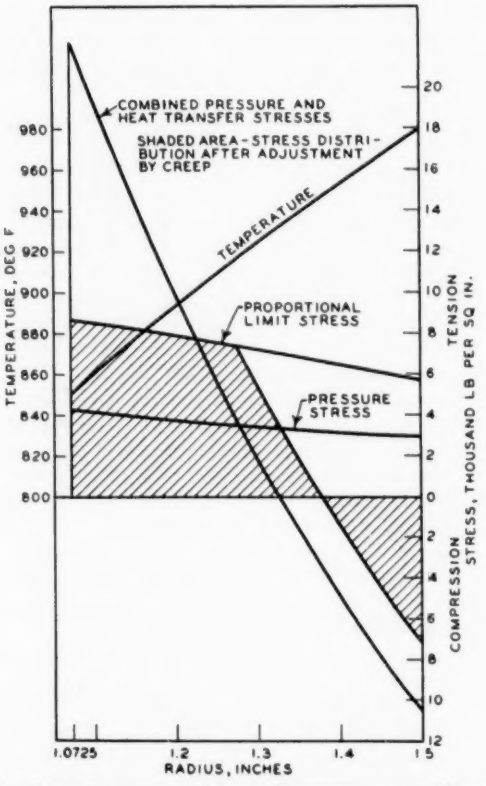


FIG. 28 CIRCUMFERENTIAL STRESS DISTRIBUTION IN TUBE WALL FOR HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT WITH 224.6 F TEMPERATURE DROP THROUGH WATER FILM

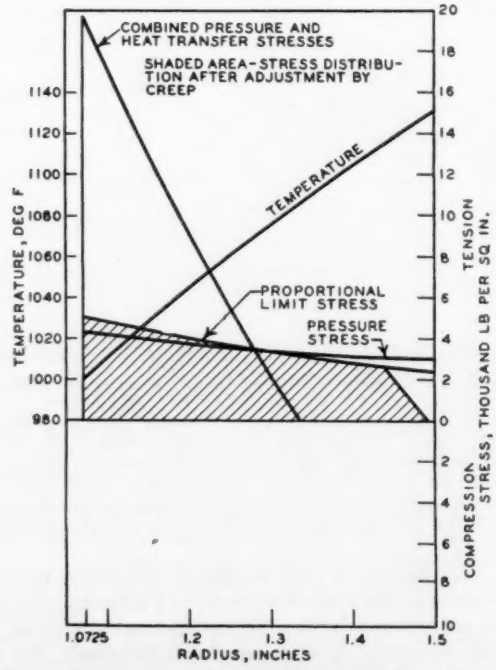


FIG. 29 CIRCUMFERENTIAL STRESS DISTRIBUTION IN TUBE WALL FOR HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT WITH 374.6 F TEMPERATURE DROP THROUGH WATER FILM

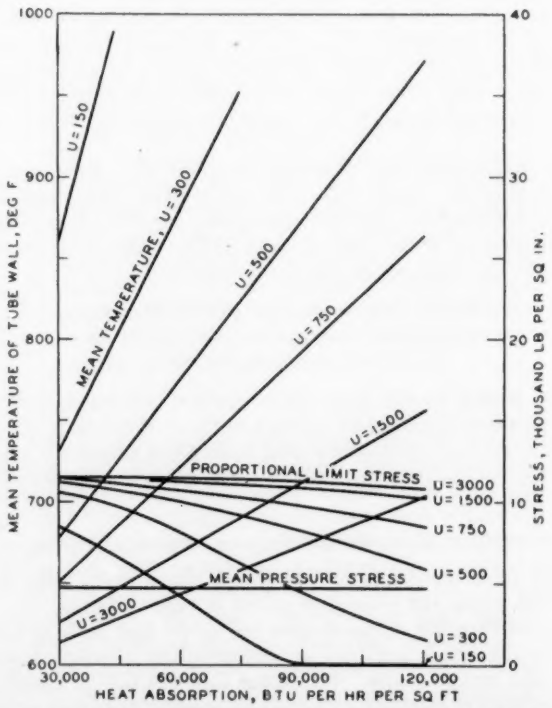


FIG. 30 VARIATION OF MEAN TEMPERATURE AND CORRESPONDING PROPORTIONAL-LIMIT STRESS WITH HEAT ABSORPTION IN A 3-IN. MILD-STEEL TUBE WITH WALL 0.344 IN. THICK

above and below the zero line to the curve for combined pressure and heat-transfer stresses must equal the area under the line for pressure stress only in accordance with the relations previously explained, these stress lines being calculated in accordance with the theory of elasticity of materials. Due, however, to the maximum values of these calculated stresses being above the elastic limit of the material, plastic flow will occur to reduce these stresses to much lower values, as previously explained. Eventually, these stresses will all be reduced to the creep-limit stress at the operating temperature at each point in the tube wall. A comparison of Figs. 36 and 39 in Appendix No. 2 shows that at low temperatures the creep-limit stress is above the proportional-limit stress of mild steel, while at high temperatures the reverse is true. For the purpose of this discussion, the proportional-limit stress has been taken as the stress to which all higher stresses are eventually reduced by plastic flow.

Curves representing the proportional-limit stress of mild steel at the varying temperature in the tube wall were drawn above and below the zero line to represent the limiting stresses in the tube wall. Wherever the calculated stress exceeded the proportional-limit stress it was reduced to that limiting value;

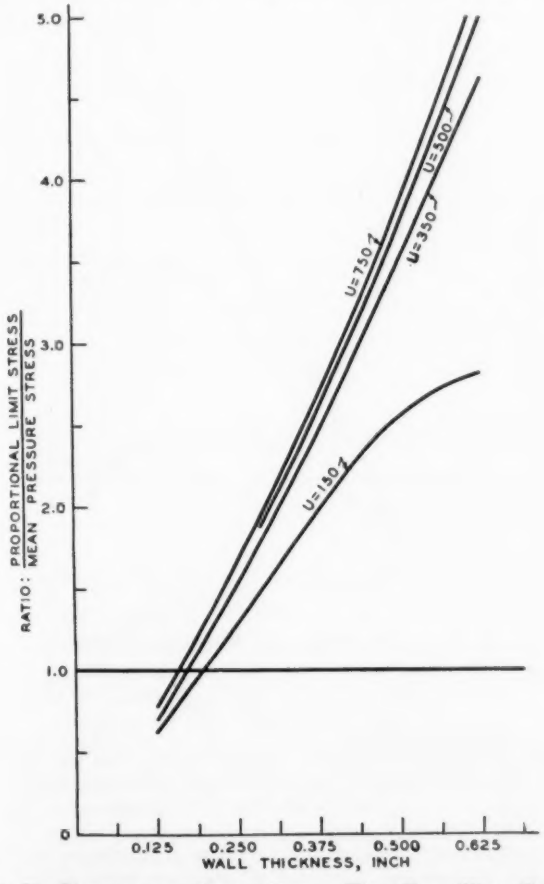


FIG. 31 VARIATION OF SAFETY FACTOR WITH TUBE-WALL THICKNESS FOR HEAT ABSORPTION OF 30,000 BTU PER HR PER SQ FT WITH DIFFERENT COEFFICIENTS OF HEAT TRANSFER THROUGH WATER FILM

but to compensate for this reduction in stress adjacent to the inner surface of tube wall, the stresses at other points within the tube wall were increased. Compensation is evidently complete when the combined stress line is moved up until the difference of the cross-sectioned areas above and below the zero line just equals the area under the pressure-stress line.

The temperature and corresponding proportional-limit curves in Figs. 27, 28, and 29 were drawn for three different coefficients of heat transfer from the inner surface of the tube wall to the water-steam mixture within the tube with the same rate of heat absorption, 90,000 Btu per hr per sq ft. The effect of an increase in temperature drop through the water film from 41.9 F to 224.6 F and finally to 374.6 F is to move the proportional-limit-stress curve toward the pressure-stress curve. Under the final

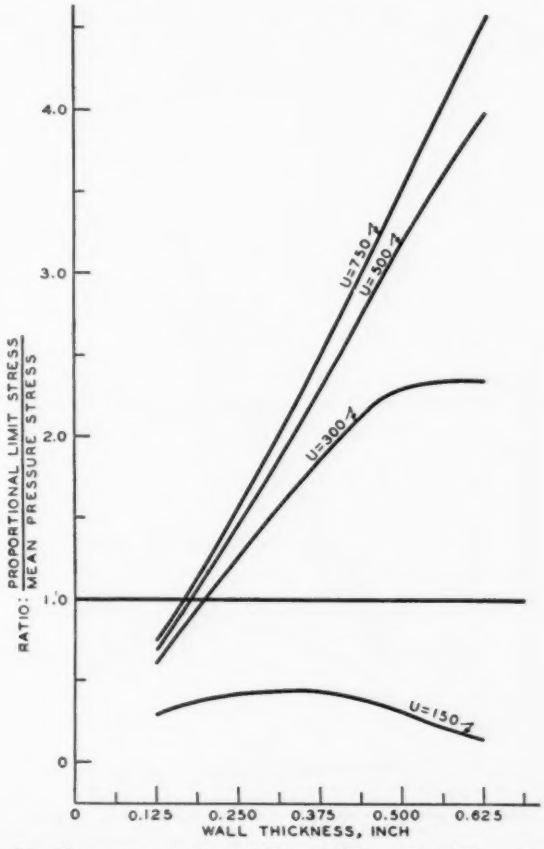


FIG. 32 VARIATION OF SAFETY FACTOR WITH TUBE-WALL THICKNESS FOR HEAT ABSORPTION OF 60,000 BTU PER HR PER SQ FT WITH DIFFERENT COEFFICIENTS OF HEAT TRANSFER THROUGH WATER FILM

condition, the tube is just about to fail because the tube-wall temperature has increased until the stress due to internal pressure exceeds the creep-limit stress of the material which is below the proportional-limit stress at these high temperatures. Before this condition has been reached, all heat-transfer stresses have evidently been neutralized by creep. A measure of safety of the boiler tube is the ratio of the proportional-limit stress (or creep-limit stress) of the material of the tube wall at the mean operating temperature to the mean stress due to internal fluid pressure. When this ratio equals unity, the tube is about to burst by its internal pressure.

SAFEST THICKNESS OF BOILER-TUBE WALL

In a thin-walled tube the stresses due to internal fluid pressure are high so that the ratio of the proportional-limit stress at the mean wall temperature to the mean-internal-pressure stress may be low and the tube unsafe. As the thickness is increased, the pressure stress is considerably decreased at first without much change in the proportional-limit stress. It is conceivable, however, that a large increase in thickness with a high rate of heat

transfer might cause the mean temperature of the tube wall to become so high that the safety of the tube would be decreased in spite of the decrease in internal-pressure stress. The safest thickness of tube wall is that thickness where the ratio of the proportional-limit stress to the internal-pressure stress is a maximum.

For a given tube material, the safest thickness will depend upon the coefficient of heat transfer from the inner surface of the tube wall to the water-steam mixture within the tube, as well as upon the rate of heat absorption. To determine how this safest thickness compares with that specified in the A.S.M.E. Boiler Code, calculations were made for wall thicknesses from $^1/_8$ in. to $^5/_8$ in. for a 3-in. outside-diameter mild-steel tube under an internal steam pressure of 1390 lb gage. Heat-absorption rates of 30,000, 60,000, 90,000, and 120,000 Btu per hr per sq ft and film coefficient of 3000, 1500, 750, 500, 300, and 150 Btu per hr per sq ft per deg F were assumed The thickness specified by the A.S.M.E. Boiler Code for an internal steam pressure of 1390 lb gage is 0.312 in.

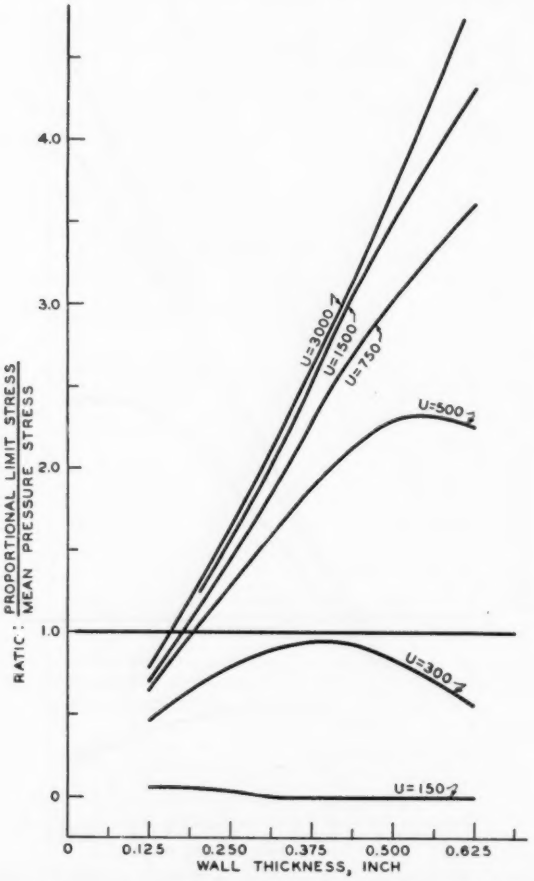


FIG. 33 VARIATION OF SAFETY FACTOR WITH TUBE-WALL THICKNESS FOR HEAT ABSORPTION OF 90,000 BTU PER HR PER SQ FT WITH DIFFERENT COEFFICIENTS OF HEAT TRANSFER THROUGH WATER FILM

For each tube-wall thickness, the mean wall temperature, corresponding proportional-limit stress, and mean-pressure stress were plotted as shown in Fig. 30 for a 0.344-in. wall. Then the ratio of proportional-limit stress to mean-pressure stress was plotted in Figs. 31–34, for the four assumed rates of heat absorption. In all cases the safest ratio is for a greater thickness of tube wall than specified by the A.S.M.E. Boiler Code. To make the tube thinner than called for would evidently be a step in the wrong direction.

The safety of a tube would be increased by using a material having a higher proportional-limit stress, such as a 0.45 per cent carbon steel instead of a 0.24 per cent as indicated by the proportional-limit curves in Fig. 36 of Appendix No. 2. Even with mild steel, however, the curves of Figs. 31–34 show that failure can occur only when the coefficient of heat transfer from the inner surface of the tube wall to the water-steam mixture within the tube is reduced to an abnormally low value by poor circulation within the tube or by scale accumulations on the tube wall unless the material is defective or corrosion occurs.

In closing, the author wishes to acknowledge the work of Messrs. R. H. Tilghman and G. H. Bierman, formerly members of the Technical Research Department of International Combustion Engineering Corporation, in preparing reports on various phases of the problem, data from these reports being utilized in the preparation of this paper.

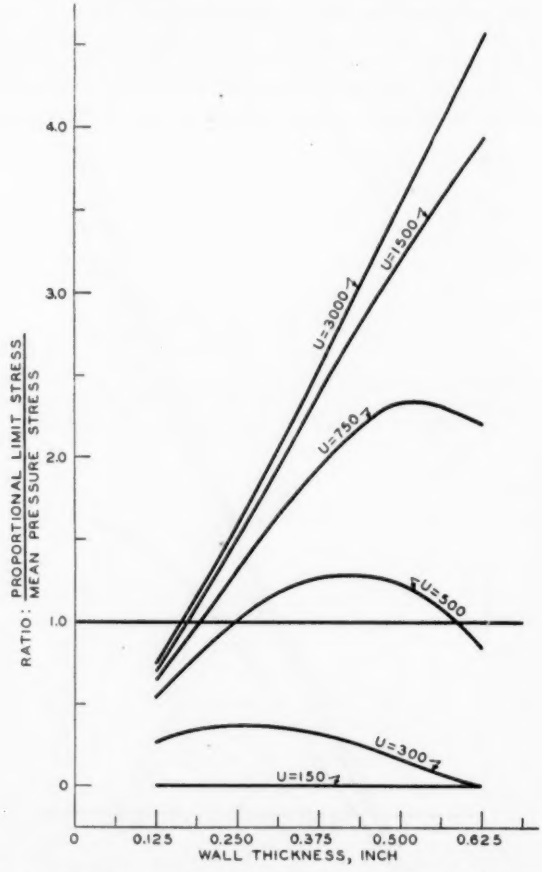


FIG. 34 VARIATION OF SAFETY FACTOR WITH TUBE-WALL THICKNESS FOR HEAT ABSORPTION OF 120,000 BTU PER HR PER SQ FT WITH DIFFERENT COEFFICIENTS OF HEAT TRANSFER THROUGH WATER FILM

Appendix No. 1

DERIVATION OF FORMULAS

TEMPERATURE DROP THROUGH CYLINDRICAL TUBE WALL

Consider a cylindrical tube in which heat is transferred radially through the tube wall to a fluid flowing within the tube. Let

r_1 = inside radius of tube wall, inches
r_2 = outside radius of tube wall, inches
T_1 = temperature at inside surface, deg F
T_2 = temperature at outside surface, deg F
Q = rate of heat absorption at outside surface, Btu per hr per sq ft
λ = thermal conductivity of material, Btu per hr per sq ft per deg F for 1 ft in thickness.

The heat absorbed by a 1-ft length of tube will be $2\pi r_2 Q/12$ Btu per hr. Let the temperature be T at a cylindrical annulus of radius r and thickness dr within the tube wall. The heat passing through a 1-ft length of this annulus will be $2\pi r\lambda dT/dr$ Btu per hr. Equating the two expressions for flow of heat,

$$2\pi r\lambda \frac{dT}{dr} = \frac{2\pi r_2 Q}{12} \quad \text{or} \quad dT = \frac{Qr_2}{12\lambda}\frac{dr}{r}$$

Integrating,

$$T = \frac{Qr_2}{12\lambda}\log_e r + C$$

Since $T = T_2$ when $r = r_2$,

$$C = T_2 - \frac{Qr_2}{12\lambda}\log_e r_2 \quad \text{and} \quad T = T_2 - \frac{Qr_2}{12\lambda}\log_e \frac{r_2}{r}$$

But $T = T_1$ when $r = r_1$, whence

$$C = T_1 - \frac{Qr_2}{12\lambda}\log_e r_1 \quad \text{and} \quad T = T_1 + \frac{Qr_2}{12\lambda}\log_e \frac{r}{r_1}$$

Eliminating $Qr_2/12\lambda$,

$$T = \frac{T_1 \log_e \frac{r_2}{r} + T_2 \log_e \frac{r}{r_1}}{\log_e \frac{r_2}{r_1}}$$

Equating the two expressions for T and solving for the temperature difference,

$$T_2 - T_1 = \frac{Qr_2}{12\lambda}\log_e \frac{r_2}{r_1}$$

Stresses in Cylindrical Tube Wall Due to Fluid Pressure

For the cylindrical tube, let

r_1 = inside radius of tube wall, inches
r_2 = outside radius of tube wall, inches
p_1 = internal fluid pressure, lb per sq in.
p_2 = external fluid press lb per sq in.

Then the radial compressive stress at the inside surface is p_1 lb per sq in. and at the outside surface is p_2 lb per sq in. Let p_1 be greater than p_2.

Consider a cylindrical annulus within the tube wall of radius r and thickness dr and let

p = radial compressive stress at radius r, lb per sq in.
t = circumferential tensile stress at radius r, lb per sq in.
s = axial tensile stress at radius r, lb per sq in.

The internal pressure acting on the cylindrical annulus is then p lb per sq in. Let the external pressure acting thereon be denoted by $p + dp$. Since the pressure difference tending to burst the cylindrical annulus is balanced by the circumferential stress therein, we have for unit length of annulus:

$$2pr - 2(p + dp)(r + dr) = 2tdr$$

Multiplying and neglecting products of small quantities,

$$pdr + rdp + tdr = 0$$

whence

$$t + p + r\frac{dp}{dr} = 0 \qquad [1]$$

Due to the greater internal pressure, let the circumferential length of the cylindrical annulus be increased $2\pi u$ inches. The tensile strain per unit length circumferentially will then be $2\pi u/2\pi r = u/r$. The strain radially will be du/dr. Let e be the tensile axial strain. Then for a coefficient of elasticity E and a Poisson's ratio $1/m$,

$$E\frac{du}{dr} = -p - \frac{1}{m}\left(t + s\right) \qquad [2]$$

$$E\frac{u}{r} = t - \frac{1}{m}\left(s - p\right) \qquad [3]$$

$$Ee = s - \frac{1}{m}\left(t - p\right) \qquad [4]$$

The radial stress p has been taken negative to represent compression in comparison with positive values of the circumferential tensile stress t and axial tensile stress s. From [3],

$$Eu = tr - \frac{sr}{m} + \frac{pr}{m}$$

Differentiating with respect to r,

$$E\frac{du}{dr} = t + r\frac{dt}{dr} - \frac{s}{m} - \frac{r}{m}\frac{ds}{dr} + \frac{p}{m} + \frac{r}{m}\frac{dp}{dr}$$

Subtracting from [2],

$$0 = -p - \frac{t}{m} - t - r\frac{dt}{dr} + \frac{r}{m}\frac{ds}{dr} - \frac{p}{m} - \frac{r}{m}\frac{dp}{dr} \qquad [5]$$

We shall assume that the axial strain e is uniform over the whole cross-section of the tube, which is evidently true for sections remote from the ends of the tube. Then, differentiating [4] with respect to r,

$$0 = \frac{ds}{dr} - \frac{1}{m}\frac{dt}{dr} + \frac{1}{m}\frac{dp}{dr} \qquad [6]$$

Substituting ds/dr from [6] in [5] and dividing through by $-[1 + (1/m)]$,

$$0 = p + t + \frac{r}{m}\frac{dp}{dr} + r\left(1 - \frac{1}{m}\right)\frac{dt}{dr}$$

Substituting $(p + t)$ from [1] and dividing through by $r[1 - (1/m)]$,

$$\frac{d}{dr}(t - p) = 0$$

Therefore

$$t - p = C, \text{ a constant} \qquad [7]$$

Subtracting [7] from [1],

$$2p + r\frac{dp}{dr} = -C$$

whence

$$\frac{1}{r}\frac{d}{dr}(r^2 p) = -C$$

or

$$d(r^2p) = -Crdr$$

Integrating,

$$r^2p = -C\frac{r^2}{2} + B$$

or

$$p = -\frac{C}{2} + \frac{B}{r^2} \quad \ldots\ldots [8]$$

Since $p = p_1$ at $r = r_1$ and $p = p_2$ at $r = r_2$, we get the values of the constants of integration,

$$C = \frac{2(r_1^2p_1 - r_2^2p_2)}{r_2^2 - r_1^2}$$

and

$$B = \frac{r_1^2r_2^2(p_1 - p_2)}{r_2^2 - r_1^2}$$

Substituting in [8], we have for the radial compressive stress

$$p = \frac{1}{r_2^2 - r_1^2}\left[r_2^2p_2 - r_1^2p_1 + \frac{r_1^2r_2^2}{r^2}\left(p_1 - p_2\right)\right] \ldots [9]$$

Substituting in [7], we have for the circumferential tensile stress

$$t = \frac{1}{r_2^2 - r_1^2}\left[r_1^2p_1 - r_2^2p_2 + \frac{r_1^2r_2^2}{r^2}\left(p_1 - p_2\right)\right] \ldots [10]$$

Since by [7], $(t - p)$ is constant if e is constant over the cross-section of the tube wall, we have from [4] that s is also constant over the cross-section of the tube wall. Hence

$$s = \frac{\pi r_1^2p_1 - \pi r_2^2p_2}{\pi r_2^2 - \pi r_1^2} = \frac{1}{r_2^2 - r_1^2}\left[r_1^2p_1 - r_2^2p_2\right] \ldots [11]$$

If p_2 is negligible in comparison with p_1, as when p_2 is atmospheric pressure and p_1 is a very high internal pressure, formulas [9], [10], and [11] reduce to

$$p = \frac{r_2^2 - r^2}{r_2^2 - r_1^2}\,\frac{r_1^2p_1}{r^2} \quad \ldots\ldots [12]$$

$$t = \frac{r_2^2 + r^2}{r_2^2 - r_1^2}\,\frac{r_1^2p_1}{r^2} \quad \ldots\ldots [13]$$

$$s = \frac{r_1^2p_1}{r_2^2 - r_1^2} \quad \ldots\ldots [14]$$

Stresses in Cylindrical Tube Wall Due to Fluid Pressure and to Heat Transfer Through Tube Wall

As before, let

r_1 = inside radius of tube wall, inches
r_2 = outside radius of tube wall, inches
p_1 = internal fluid pressure, lb per sq in.
p_2 = external fluid pressure, lb per sq in.
T_1 = temperature at inside surface, deg F
T_2 = temperature at outside surface, deg F, these temperatures being measured above some reference temperature T_0.

Consider a cylindrical annulus of radius r and thickness dr within the tube wall where the temperature is T deg F above the reference temperature T_0. Let

p = radial compressive stress at radius r, lb per sq in.
t = circumferential tensile stress at radius r, lb per sq in.
s = axial tensile stress at radius r, lb per sq in.

Then as previously derived,

$$t + p + r\frac{dp}{dr} = 0 \quad \ldots\ldots [1]$$

If α = coefficient of linear expansion per deg F, the strain due to the temperature rise T is αT. If the total strains are du/dr, u/r, and e in the radial, circumferential, and axial directions, respectively,

$$E\left(\frac{du}{dr} - \alpha T\right) = -p - \frac{1}{m}\left(t + s\right) \quad \ldots\ldots [2]$$

$$E\left(\frac{u}{r} - \alpha T\right) = t - \frac{1}{m}\left(s - p\right) \quad \ldots\ldots [3]$$

$$E(e - \alpha T) = s - \frac{1}{m}\left(t - p\right) \quad \ldots\ldots [4]$$

From [3],

$$Eu = E\alpha Tr + tr - \frac{sr}{m} + \frac{pr}{m}$$

Differentiating with respect to r,

$$E\frac{du}{dr} = E\alpha T + E\alpha r\frac{dT}{dr} + t + r\frac{dt}{dr} - \frac{s}{m} - \frac{r}{m}\frac{ds}{dr} + \frac{p}{m} + \frac{r}{m}\frac{dp}{dr}$$

Subtracting from [1],

$$0 = -p - \frac{t}{m} - E\alpha r\frac{dT}{dr} - t - r\frac{dt}{dr} + \frac{r}{m}\frac{ds}{dr} - \frac{p}{m} - \frac{r}{m}\frac{dp}{dr} \quad [5]$$

Assuming the axial strain e to be uniform over the whole cross-section, we have by differentiating [4],

$$\frac{ds}{dr} = -E\alpha\frac{dT}{dr} + \frac{1}{m}\frac{dt}{dr} - \frac{1}{m}\frac{dp}{dr} \quad \ldots\ldots [6]$$

Substituting ds/dr from [6] in [5] and dividing through by $-[1 + (1/m)]$,

$$0 = p + t + E\alpha r\frac{dT}{dr} + \frac{r}{m}\frac{dp}{dr} + r\left(1 - \frac{1}{m}\right)\frac{dt}{dr}$$

Substituting $p + t$ from [1] and dividing by $r[1 - (1/m)]$,

$$\frac{d}{dr}\left(t - p\right) + \frac{E\alpha m}{m - 1}\frac{dT}{dr} = 0$$

Therefore

$$t - p + \frac{E\alpha m}{m - 1}T = C, \text{ a constant} \quad \ldots\ldots [7]$$

Subtracting [7] from [1],

$$2p + r\frac{dp}{dr} - \frac{E\alpha m}{m - 1}T = -C$$

or

$$\frac{1}{r}\frac{d}{dr}\left(r^2p\right) - \frac{E\alpha m}{m-1}T = -C \quad \ldots\ldots [8]$$

But

$$T = \frac{T_1 \log_e \frac{r_2}{r} + T_2 \log_e \frac{r}{r_1}}{\log_e \frac{r_2}{r_1}} \quad \ldots\ldots [9]$$

Substituting [9] in [8] and integrating,

$$r^2p - \frac{E\alpha m}{(m-1)\log_e \frac{r_2}{r_1}}\left[(T_1 \log_e r_2 - T_2 \log_e r_1)\frac{r^2}{2} + (T_2 - T_1)\left(\frac{\log_e r}{2} - \frac{1}{4}\right)r^2\right] = -\frac{Cr^2}{2} + B$$

Whence

$$r^2p - \frac{E\alpha m}{m-1}\left[\frac{r^2T}{2} - \frac{r^2(T_2 - T_1)}{4\log_e \frac{r_2}{r_1}}\right] = -\frac{Cr^2}{2} + B \ldots [10]$$

where B is a constant of integration. Since $p = p_1$, and $T = T_1$ when $r = r_1$, and since $p = p_2$ and $T = T_2$ where $r = r_2$,

$$C = \frac{2(r_1^2p_1 - r_2^2p_2)}{r_2^2 - r_1^2} - \frac{E\alpha m}{m-1}\left[\frac{r_1^2T_1 - r_2^2T_2}{r_2^2 - r_1^2} + \frac{T_2 - T_1}{2\log_e \frac{r_2}{r_1}}\right] [11]$$

and

$$B = \frac{r_1^2r_2^2}{r_2^2 - r_1^2}\left[(p_1 - p_2) + \frac{E\alpha m}{m-1}\frac{T_2 - T_1}{2}\right] \ldots\ldots [12]$$

Substituting these constants in [10], the radial compressive stress is found to be

$$p = \frac{1}{r_2^2 - r_1^2}\left[r_2^2p_2 - r_1^2p_1 + \frac{r_1^2r_2^2}{r^2}\left(p_1 - p_2\right)\right] + \frac{E\alpha m}{2(m-1)}\left[\frac{r_1^2T_1 - r_2^2T_2}{r_2^2 - r_1^2} + \frac{r_1^2r_2^2}{r^2}\frac{T_2 - T_1}{r_2^2 - r_1^2} + T\right] \ldots [13]$$

Substituting the values of p and C in [7], the circumferential tensile stress is found to be

$$t = \frac{1}{r_2^2 - r_1^2}\left[r_1^2p_1 - r_2^2p_2 + \frac{r_1^2r_2^2}{r^2}\left(p_1 - p_2\right)\right] + \frac{E\alpha m}{2(m-1)}\left[\frac{r_2^2T_2 - r_1^2T_1}{r_2^2 - r_1^2} + \frac{r_1^2r_2^2}{r^2}\frac{T_2 - T_1}{r_2^2 - r_1^2} - \frac{T_2 - T_1}{\log_e \frac{r_2}{r_1}} - T\right] \ldots [14]$$

To find the axial tensile stress s, we utilize the relation

$$\int_{r_1}^{r_2} s \times 2\pi r dr = \pi r_1^2p_1 - \pi r_2^2p_2 \ldots\ldots [15]$$

From [4],

$$s = E(e - \alpha T) + \frac{1}{m}\left(t - p\right) \ldots\ldots [16]$$

The values of $(t - p)$ and C are obtained from [7] and [11], respectively. Whence

$$r_1^2p_1 - r_2^2p_2 = \int_{r_1}^{r_2} 2srdr$$

$$= \int_{r_1}^{r_2} 2\left[Ee - E\alpha T + \frac{1}{m}\left(t - p\right)\right]rdr$$

$$= \int_{r_1}^{r_2} 2\left[Ee - E\alpha T + \frac{1}{m}\left(C - \frac{E\alpha m}{m-1}T\right)\right]rdr$$

$$= \int_{r_1}^{r_2} 2\left[Ee - E\alpha T + \frac{2}{m}\frac{r_1^2p_1 - r_2^2p_2}{r_2^2 - r_1^2} - \frac{E\alpha}{m-1}\left(\frac{r_1^2T_1 - r_2^2T_2}{r_2^2 - r_1^2} + \frac{T_2 - T_1}{2\log_e \frac{r_2}{r_1}}\right) - \frac{E\alpha}{m-1} - T\right]rdr$$

Substituting T from [9] and integrating

$$Ee = \frac{m-2}{m}\frac{r_1^2p_1 - r_2^2p_2}{r_2^2 - r_1^2} - E\alpha\frac{r_1^2T_1 - r_2^2T_2}{r_2^2 - r_1^2} - E\alpha\frac{T_2 - T_1}{2\log_e \frac{r_2}{r_1}} \ldots\ldots [17]$$

Substituting in [16], we get for the axial tensile stress

$$s = \frac{r_1^2p_1 - r_2^2p_2}{r_2^2 - r_1^2} + \frac{E\alpha m}{m-1}\left[\frac{r_2^2T_2 - r_1^2T_1}{r_2^2 - r_1^2} - \frac{T_2 - T_1}{2\log_e \frac{r_2}{r_1}} - T\right] \ldots\ldots [18]$$

In order to eliminate T from expressions [13], [14], and [18], its value may be substituted from [9], thereby obtaining for the radial, circumferential, and axial stresses:

$$p = \left[-\frac{r_1^2p_1 - r_2^2p_2}{r_2^2 - r_1^2} + \frac{r_1^2r_2^2(p_1 - p_2)}{r^2(r_2^2 - r_1^2)}\right] + \frac{E\alpha m(T_2 - T_1)}{2(m-1)(r_2^2 - r_1^2)\log_e \frac{r_2}{r_1}}\left[-(r_2^2\log_e r_2 - r_1^2\log_e r_1) + \frac{r_1^2r_2^2}{r^2}\log_e \frac{r_2}{r_1} + (r_2^2 - r_1^2)\log_e r\right] \ldots\ldots [19]$$

$$t = \left[\frac{r_1^2p_1 - r_2^2p_2}{r_2^2 - r_1^2} + \frac{r_1^2r_2^2(p_1 - p_2)}{r^2(r_2^2 - r_1^2)}\right] + \frac{E\alpha m(T_2 - T_1)}{2(m-1)(r_2^2 - r_1^2)\log_e \frac{r_2}{r_1}}\left[(r_2^2\log_e r_2 - r_1^2\log_e r_1) - (r_2^2 - r_1^2) + \frac{r_1^2r_2^2}{r^2}\log_e \frac{r_2}{r_1} - (r_2^2 - r_1^2)\log_e r\right] \ldots\ldots [20]$$

$$s = \left[\frac{r_1^2p_1 - r_2^2p_2}{r_2^2 - r_1^2}\right] + \frac{E\alpha m(T_2 - T_1)}{2(m-1)(r_2^2 - r_1^2)\log_e \frac{r_2}{r_1}}\left[2(r_2^2\log_e r_2 - r_1^2\log_e r_1) - (r_2^2 - r_1^2) - 2(r_2^2 - r_1^2)\log_e r\right] \ldots\ldots [21]$$

Stresses in Cylindrical Tube Wall Due to Heat Transfer Only Through Tube Wall

By a derivation similar to the preceding one, it may be shown that if

r_1 = inside radius of tube wall, inches
r_2 = outside radius of tube wall, inches
T_1 = temperature at inside surface, deg F, and
T_2 = temperature at outside surface, deg F, above some reference temperature T_0
p = radial compressive stress at radius r, lb per sq in.
t = circumferential tensile stress at radius r, lb per sq in.
s = axial tensile stress at radius r, lb per sq in.

$$p = \frac{E\alpha m(T_2 - T_1)}{2(m-1)(r_2^2 - r_1^2)\log_e \frac{r_2}{r_1}}\left[-(r_2^2 \log_e r_2 - r_1^2 \log_e r_1) + \frac{r_1^2 r_2^2}{r^2}\log_e \frac{r_2}{r_1} + (r_2^2 - r_1^2)\log_e r\right] \quad [1]$$

$$t = \frac{E\alpha m(T_2 - T_1)}{2(m-1)(r_2^2 - r_1^2)\log_e \frac{r_2}{r_1}}\left[(r_2^2 \log_e r_2 - r_1^2 \log_e r_1) - (r_2^2 - r_1^2) + \frac{r_1^2 r_2^2}{r^2}\log_e \frac{r_2}{r_1} - (r_2^2 - r_1^2)\log_e r\right] \quad [2]$$

$$s = \frac{E\alpha M(T_2 - T_1)}{2(m-1)(r_2^2 - r_1^2)\log_e \frac{r_2}{r_1}}\left[2(r_2^2 \log_e r_2 - r_1^2 \log_e r_1) - (r_2^2 - r_1^2) - 2(r_2^2 - r_1^2)\log_e r\right] \quad [3]$$

By comparing the expressions derived for fluid pressure only and for heat transfer only with the expressions derived for a combination of both effects, it will be seen that in each case the total stress is equal to the sum of the stresses due to fluid pressure and to heat transfer. Hence the stresses due to fluid pressure and to heat transfer may be calculated separately and then added to obtain the total stress in the tube wall. The complete derivation of the total stress has been included to show this fact. For the derivation of the formulas for heat transfer only, see the treatise on "Strength of Materials," by John Case.

Axial Stresses in Cylindrical Tube Wall Before Bending Due to Heat Transfer on One Side Only

Consider a cylindrical tube in which heat is transferred radially through the tube wall around 180 deg only of the circumference to a fluid flowing within the tube, the remaining 180 deg of the tube wall being at the same temperature as the fluid within. Let

r_1 = inside radius of tube wall, inches
r_2 = outside radius of tube wall, inches
p_1 = internal fluid pressure, lb per sq in. (outside fluid pressure being zero)
T_1 = temperature at inside surface of tube wall on furnace side, deg F
T_2 = temperature at outside surface of tube wall on furnace side, deg F
T_3 = temperature of tube wall on opposite to furnace side of tube, deg F
E_1 = modulus of elasticity corresponding to T_1, lb per sq in.
E_2 = modulus of elasticity corresponding to T_2, lb per sq in.
E = modulus of elasticity corresponding to $(T_1 + T_2)/2$, lb per sq in.
α_1 = coefficient of thermal expansion corresponding to T_1, inch per inch per deg F
α_2 = coefficient of thermal expansion corresponding to T_2, inch per inch per deg F
α = coefficient of thermal expansion corresponding to $(T_1 + T_2)/2$, inch per inch per deg F
$1/m$ = Poisson's ratio
s_1 = axial compressive stress at r_1 on furnace side, lb per sq in.
s_2 = axial compressive stress at r_2 on furnace side, lb per sq in.
s_3 = axial tensile stress in tube wall on opposite to furnace side, lb per sq in.

On the furnace side of the tube, the axial stress in the tube wall will vary from a maximum of s_2 lb per sq in. in compression at the outside surface to a minimum of s_1 lb per sq in. at the inside surface. The latter stress may be tension. This variation in stress is due primarily to heat transfer on the furnace side of the tube. On the opposite side there will be a nearly uniform axial tensile stress across the tube wall of s_3 lb per sq in. The summation of the compressive stresses on the furnace side of the tube is given by

$$\int_{r_1}^{r_2} s\pi r dr \quad [1]$$

where πr is the semicircumference of an elemental area at radius r. Assuming a straight-line relation for the stress in the tube wall between the stresses at the inside and outside surfaces,

$$\frac{s - s_1}{s_2 - s_1} = \frac{r - r_1}{r_2 - r_1} \quad [2]$$

Substituting s from [2] in [1] and integrating, we get for the summation of the stresses on the furnace side,

$$\frac{\pi}{6}(r_2 - r_1)\left[s_1(r_2 + 2r_1) + s_2(2r_2 + r_1)\right] \quad [3]$$

The summation over one-half of the cross-sectional area of the tube wall of the nearly constant stress s_3 in the wall on the opposite to the furnace side, is

$$\frac{\pi}{2} s_3(r_2^2 - r_1^2) \quad [4]$$

Since the summation of the axial tensile stresses will exceed the summation of the axial compressive stresses by the total axial internal pressure,

$$\pi r_1^2 p_1 = \frac{\pi}{2} s_3(r_2^2 - r_1^2) - \frac{\pi}{6}\left(r_2 - r_1\right)\left[s_1(r_2 + 2r_1) + s_2(2r_2 + r_1)\right] \quad [5]$$

In Equation [5] there are three unknown quantities, namely, s_1, s_2, and s_3. Let $s_1 + s_3 = A$ and $s_2 + s_3 = B$. Then, substituting $s_1 = A - s_3$ and $s_2 = B - s_3$ in [5] and solving for s_3,

$$s_3 = \frac{r_1^2 p_1}{r_2^2 - r_1^2} + \frac{A}{6}\frac{r_2 + 2r_1}{r_2 + r_1} + \frac{B}{6}\frac{2r_2 + r_1}{r_2 + r_1} \quad [6]$$

Now, A is the compressive stress at the inside surface and B is the compressive stress at the outside surface on the furnace side of the tube relative to the constant stress in the tube wall

on the opposite side of the tube due to the suppressed relative axial expansion of the different parts of the tube before bending. These relative expansions are due mainly to differences in temperature, but are also affected by the circumferential stresses in the tube wall. The radial stresses are so small as to be negligible. Hence we may write

$$A = (T_1 - T_3)\alpha_1 E_1 - \frac{t_1}{m} \quad [7]$$

and

$$B = (T_2 - T_3)\alpha_2 E_2 + \frac{t_2}{m} \quad [8]$$

The reason for the above expressions for A and B may be explained by considering a cube which is expanded equally in all directions by a rise in temperature. If a tensile stress t is applied in one direction, the expansion at right angles to this tensile stress will be reduced by an amount t/mE, where E is the modulus of elasticity of the material and $1/m$ is Poisson's ratio. Hence the compressive stress required to return the cube to its original length at right angles to the tensile stress is less than that corresponding to the temperature expansion by t/m. The circumferential stress t_1 at the inside surface is a tensile stress, while the circumferential stress t_2 at the outside surface is a compressive stress.

For the difference between the circumferential stresses at the outer and inner surfaces of the tube wall on the furnace side of the tube, the following approximate relation will be used, namely:

$$t_2 + t_1 = (T_2 - T_1)\alpha E$$

$$= 2t_1 = -2t_2$$

$$t_1 = (T_2 - T_1)\alpha E/2 \quad [9]$$

$$t_2 = -(T_2 - T_1)\alpha E/2 \quad [10]$$

These various relations were not combined, but were used separately to work out the several values.

Moments of Axial Stresses in Cylindrical Tube Wall Due to Heat Transfer on One Side Only

Consider a cylindrical tube in which for 180 deg of the circumference on the furnace side the stress varies in a straight-line relation from a maximum compressive stress in the outside surface to a minimum compressive stress at the inside surface while for the remaining 180 deg a constant tensile stress exists in the tube wall. Let

- r_1 = inside radius of tube wall, inches
- r_2 = outside radius of tube wall, inches
- s_1 = axial compressive stress before bending at r_1 on furnace side, lb per sq in.
- s_2 = axial compressive stress before bending at r_2 on furnace side, lb per sq in.
- s_3 = axial tensile stress before bending in tube wall on opposite to furnace side, lb per sq in.
- s_4 = axial stress in outermost fiber superimposed by bending, lb per sq in.

Let s be the stress on an elemental area $dr \times rd\theta$ distant r from the center of the tube cross-section. Then the elemental force is $srdrd\theta$ and the elemental moment about the neutral axis passing through the center of the tube cross-section is $r \sin \theta \times srdrd\theta$, where θ is the angle between the radius to the elemental area and the neutral axis. The total moment for one-half of the tube cross-section on one side of the neutral axis is

$$M = \int_{r=r_1}^{r=r_2} \int_{\theta=0}^{\theta=\pi} sr^2 \sin\theta \, drd\theta \quad [1]$$

On the furnace side of the tube,

$$s = s_1 + \left(s_2 - s_1\right)\frac{r - r_1}{r_2 - r_1} \quad [2]$$

as previously derived. Substituting in [1] and integrating, we get for the total moment of the stresses on the furnace side of the tube before bending:

$$M_1 = 2\left[\frac{s_1(r_2^3 - r_1^3)}{3} + \frac{(s_2 - s_1)(r_2^4 - r_1^4)}{4(r_2 - r_1)} - \frac{(s_2 - s_1)r_1(r_2^3 - r_1^3)}{3(r_2 - r_1)}\right] \quad [3]$$

For the opposite side of the tube, the stress

$$s = s_3 \text{ (constant)} \quad [4]$$

Substituting this in [1] and integrating, we get for the total moment of the stresses on the opposite to the furnace side of the tube before bending:

$$M_2 = \frac{2s_3(r_2^3 - r_1^3)}{3} \quad [5]$$

The stress superimposed on the tube at the outermost fiber by bending is given by the well-known relation

$$M = \frac{s_4 I}{c} \quad [6]$$

where $I = \frac{\pi}{4}(r_2^4 - r_1^4)$, section modulus, and $c = r_2$, distance from neutral axis to outermost fiber. If the tube is unrestrained, it will bend until

$$M = M_1 + M_2 \quad [7]$$

Hence

$$s_4 = \frac{4(M_1 + M_2)r_2}{\pi(r_2^4 - r_1^4)} \quad [8]$$

Appendix No. 2

PHYSICAL PROPERTIES OF MILD STEEL

When this investigation of the effects of heat absorption upon the stresses in boiler tubes was started, the best data available on the properties of mild steel at elevated temperatures were those by French, Cross, and Peterson in Technologic Paper No. 362 of the U. S. Bureau of Standards, entitled "Creep in Five Steels at Different Temperatures." The data on 0.24 per cent carbon steel in this publication checked very closely the data on 0.17 per cent carbon steel by Spring and Kanter in *Valve World* for August, 1928, in their article on "Long-Time or Flow Tests of Carbon Steels at Various Temperatures, With Particular Reference to Stresses Below the Proportional Limit." As stress-strain curves at several temperatures were given in the former publication but not in the latter, it was thought best to use the Bureau of Standards data almost exclusively in order to obtain a consistent group of physical properties.

For the modulus of elasticity of mild steel at elevated temperatures, the modified Sutherland formula was used as given

by Orrok in his paper on "High-Pressure Steam Boilers" in Trans. A.S.M.E. for 1928 (FSP-50-32). The curve representing this formula is given in Fig. 35 with points plotted for the experimental data on 0.24 per cent carbon steel from the Bureau of Standards publication. Although the points are somewhat scattered, it is believed that they justify the use of this formula in connection with the Bureau of Standards data.

The proportional limit of 0.24 per cent carbon steel is given by the curve in Fig. 36, plotted from data in the above-mentioned Bureau of Standards publication. There is also plotted in this figure a curve for 0.45 per cent carbon steel from data received from the U. S. Bureau of Standards.

In Figs. 37 and 38 are drawn the stress-strain diagrams used for correcting the calculated stresses in a thick boiler tube where the calculated stresses exceeded the elastic limit of mild steel. Fig. 37 contains the stress-strain diagrams for the mean wall temperatures with uniform heat absorption around the tube and also the experimental stress-strain diagrams from the Bureau of Standards publication. Fig. 38 contains the stress-strain diagrams for the mean tube-wall temperatures with non-uniform heat absorption. In constructing each of these stress-strain diagrams, a straight line was first drawn from the origin up to the proportional limit shown by the curve in Fig. 36 and at the slope given by the curve in Fig. 35 corresponding to the mean temperature of the tube wall. The portion of the curve above the proportional limit was then drawn to correspond with the experimental curves.

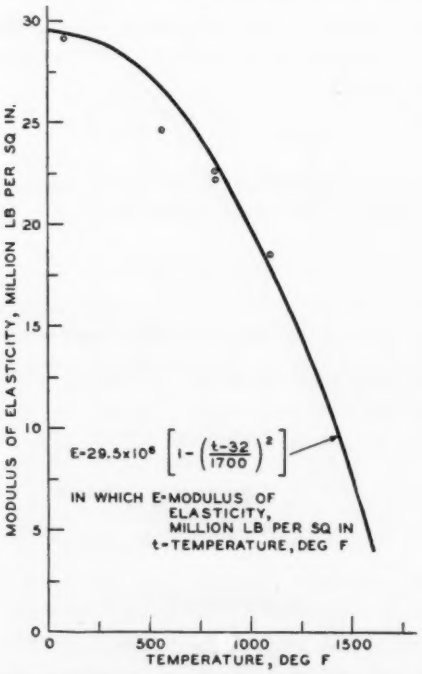


FIG. 35 MODULUS OF ELASTICITY OF MILD CARBON STEEL AT ELEVATED TEMPERATURES

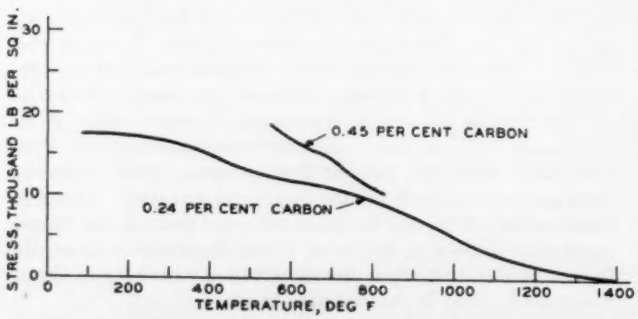


FIG. 36 PROPORTIONAL-LIMIT STRESS OF MILD AND MEDIUM CARBON STEEL AT ELEVATED TEMPERATURES

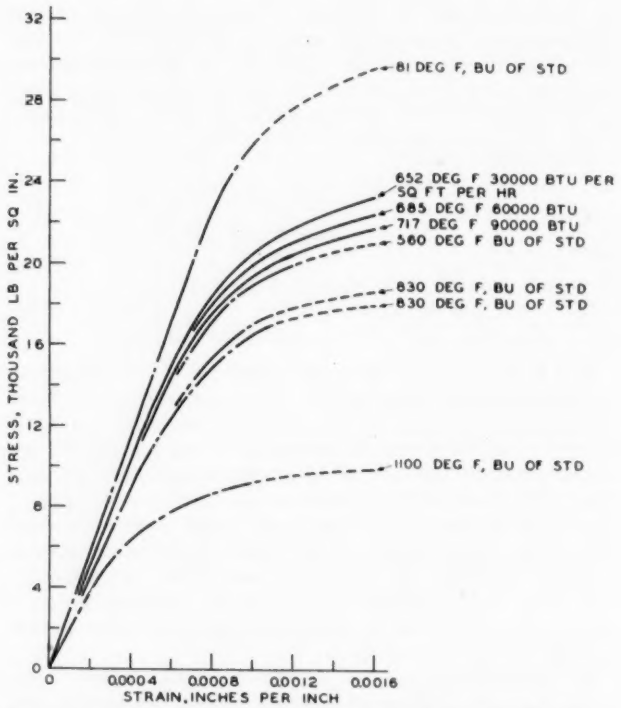


FIG. 37 STRESS-STRAIN DIAGRAMS FOR MILD CARBON STEEL AT MEAN TUBE-WALL TEMPERATURES FOR UNIFORM HEAT ABSORPTION

The curves in Fig. 39 for the rate of creep of 0.24 per cent carbon steel were obtained by cross-plotting the data in the Bureau of Standards publication where curves are given for three temperatures, 560, 830, and 1100 F.

The coefficient of thermal expansion of mild steel as given by the curve in Fig. 40 was based on an investigation made by R. H. Luebbers, formerly of the Technical Research Department of International Combustion Engineering Corporation.

The coefficient of thermal conductivity given in Fig. 41 was based on studies of Messrs. R. F. Nielsen and R. H. Luebbers, formerly of the Technical Research Department of International Combustion Engineering Corporation.

The value of Poisson's ratio was taken as 0.30.

The most comprehensive data found on the fatigue strength of mild steel at elevated temperatures were those in the paper by Tapsell published in the *Journal of the Iron and Steel Institute* for 1928, entitled "The Fatigue-Resisting Properties of 0.17 Per Cent Carbon Steel at Different Temperatures and at Different Mean Tensile Stresses." The steel was normalized and the experiments were conducted in a Haigh machine which gave direct pulsating stresses at the rate of 2400 cycles per min. The results of the experiments are given in Table 4. Failure did not occur during ten million stress cycles.

The fatigue limit was determined for three different stress ranges at each temperature. In column 2, the mean of the stress range was 0 lb per sq in.; for 70 F, the fatigue limit was reached when the stress was reversed from 28,000 lb per sq in. in tension to 28,000 lb per sq in. in compression. In column 3, the stress varied from zero to a maximum value in tension; for 70 F, the machine was set for a mean stress of 24,600 lb per sq in.

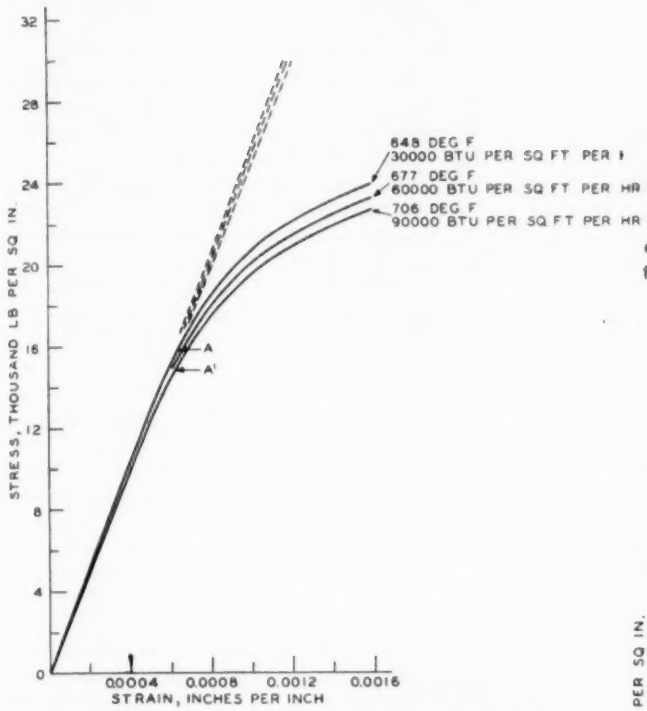


FIG. 38 STRESS-STRAIN DIAGRAMS FOR MILD CARBON STEEL AT MEAN TUBE-WALL TEMPERATURES FOR NON-UNIFORM HEAT ABSORPTION

and the stress varied 24,600 lb per sq in. above and below this mean value, that is, from 0 to 49,200 lb per sq in. in tension. Column 4 shows that at 70 F the machine was set for a mean stress of 49,300 lb per sq in. and the stress varied 16,800 lb per sq in. below and above this value, or from 32,500 to 66,100 lb per sq in. By comparison of columns 2, 3, and 4 with column 5, it is seen that, particularly at elevated temperatures, the fatigue strength of the material is above its proportional limit stress.

TABLE 4

Temperature, deg F	Fatigue-limit stress, 1000 lb per sq in.			Proportional limit, short-time test, 1000 lb. per sq in.	Tensile strength, short-time test, 1000 lb per sq in.
(1)	(2)	(3)	(4)	(5)	(6)
70	0 ± 28.0	24.6 ± 24.6	49.3 ± 16.8	35.8	63.8
212	0 ± 27.5	23.5 ± 23.5	47.1 ± 17.9	40.7	68.2
392	0 ± 27.5	25.8 ± 25.8	51.5 ± 24.6	30.2	84.2
572	0 ± 35.8	33.6 ± 33.6	67.2 ± 17.9	19.3	76.8
752	0 ± 37.6	29.1 ± 29.1	53.7 ± 13.4	14.6	62.2
932	0 ± 26.2	19.0 ± 19.0	31.3 ± 13.4	11.2	43.4

The proportional limits given by Tapsell, however, appear high when compared with the values in Fig. 36 from the experiments at the U. S. Bureau of Standards. The lower values obtained at the Bureau of Standards may be due to the extreme accuracy with which the proportional limit was determined at the first slight deviation of the stress-strain curve from a straight line. Usually, higher values of the proportional limit are obtained by reason of less accurate measurements.

It may be, however, that the material tested by Tapsell was superior to the average grade of mild steel, as is indicated by the high tensile strength in column 6. At 70 F, Tapsell obtained a tensile strength of 63,800 lb per sq in., while an average for ordinary mild steel would be 55,000 lb per sq in. To obtain safe values for the fatigue limits, all of Tapsell's values were therefore multiplied by the ratio 55,000/63,800, obtaining:

Temperature, deg F	Fatigue-limit stress, 1000 lb per sq in.			Proportional limit, short-time test, 1000 lb per sq in.	Tensile strength, short-time test, 1000 lb per sq in.
(1)	(2)	(3)	(4)	(5)	(6)
70	0 ± 24.1	21.2 ± 21.2	42.5 ± 14.5	30.9	55.0
212	0 ± 23.7	20.3 ± 20.3	40.6 ± 15.4	35.1	58.8
392	0 ± 23.7	22.3 ± 22.3	44.4 ± 21.2	26.0	72.6
572	0 ± 30.9	29.0 ± 29.0	57.9 ± 15.4	16.6	66.2
752	0 ± 32.4	25.1 ± 25.1	46.3 ± 11.6	12.6	53.6
932	0 ± 22.6	16.4 ± 16.4	27.0 ± 11.6	9.7	37.4

It may be mentioned that the proportional-limit stress and creep-limit stress were taken the same in compression as in tension.

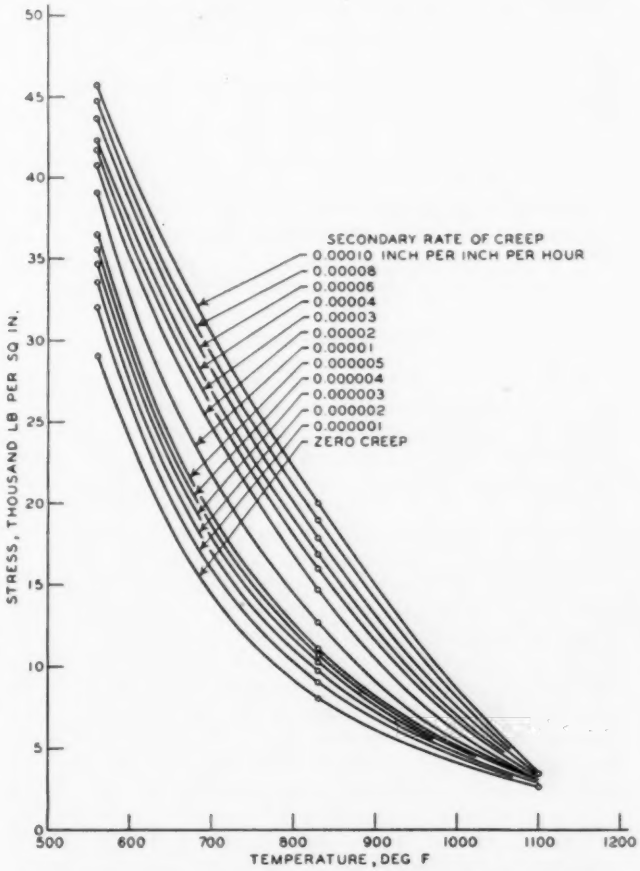


FIG. 39 SECONDARY RATE OF CREEP OF MILD CARBON STEEL AT ELEVATED TEMPERATURES

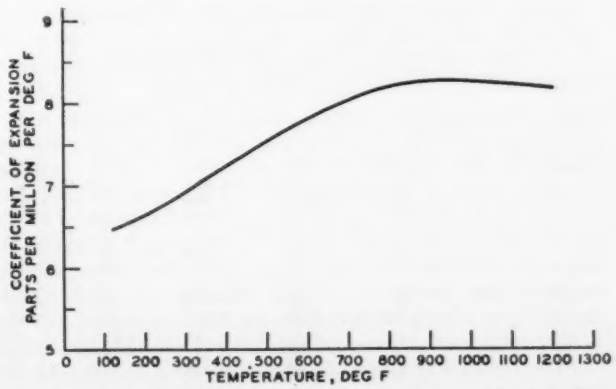


FIG. 40 COEFFICIENT OF THERMAL EXPANSION OF MILD CARBON STEEL AT ELEVATED TEMPERATURES

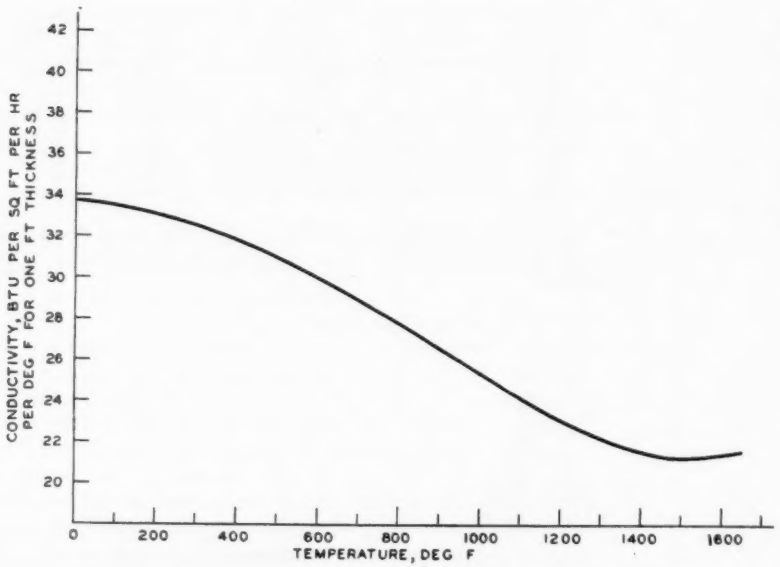


FIG. 41 COEFFICIENT OF THERMAL CONDUCTIVITY OF MILD CARBON STEEL AT ELEVATED TEMPERATURES

criterion, however, is the results secured in long-time service, which should be the governing factor in establishing any set of rules.

The author says that the investigations show that an increase in tube safety is secured by using thicker rather than thinner tubes as called for by the A.S.M.E. Boiler Code.

The new German rules, enacted by the Imperial Minister of Public Works, on July 13, 1931, sanction the use of thinner tubes than those specified in the old German rules or in the A.S.M.E. Code. This was done with the idea of securing improved results, especially at high rates of heat transfer. The A.S.M.E. and new German rules follow:

$$\text{(A)}\quad P = \frac{(t - 0.039)}{D} \times 18{,}000 - 250$$

$$\text{(B)}\quad P = \frac{(t - 0.0394)}{d} \times 18{,}200$$

$$\text{(C)}\quad P = \frac{(t - 0.0394)}{d} \times 22{,}750$$

Discussion

D. S. JACOBUS.[2] Analyses have been made to determine the effect of stresses induced by transmitting heat through the tubes, but so far as is known this is the first attempt to show in a quantitative way the modifying effect of plastic flow. That plastic flow is an important element was brought out in a paper which the writer read at the Annual Meeting of the Society in 1929.[3] It is well known that the effect of creep is to make the stress more uniform than indicated by the elastic theory.

By far the greatest number of tube failures are caused by deposits on the inside of the tubes, and there is no means of making tubes immune to such failures if they are not kept clean. Failures which occur through the accumulation of matter inside of tubes usually result from local overheating, and the consequent bulging of the tubes gives a much different distribution of forces than assumed in the usual analyses. The conduction of heat away from the part which is locally overheated to cooler parts of the tube is facilitated through using thicker rather than thinner tubes. The author has simplified the analysis by assuming that the internal surfaces of the tubes are clean and computed the magnitude of the effect of plastic flow under these conditions in reducing the stresses. Analyses of this sort are of great value in setting forth the elements entering the problem. The final

[2] Advisory Engineer, The Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

[3] "Working Stresses for Steel at High Temperatures," by D. S. Jacobus, Trans. A.S.M.E., FSP-52-35, p. 295.

where

P = maximum allowable working pressure in lb per sq in.
t = thickness of tube wall, in.
D = outside diameter of tube, in.
d = inside diameter of tube, in.

Formula (A) is the A.S.M.E. Boiler Code formula for seamless steel tubes in which the tensile strength may be as low as 45,000 lb per sq in. It is also for lap-welded steel tubes for pressures below 875 lb per sq in.

Formula (B) is the new German formula for tubes of a mild steel having a tensile strength of from 49,800 to 64,000 lb per sq in.

Formula (C) is the new German formula for tubes of steel having a tensile strength of from 64,000 to 78,000 lb per sq in.

A comparison of the tube thickness as required by the A.S.M.E. Code rules and the new German rules is given in Table 5 and of the pressures for given thicknesses in Table 6.

The figures in the tables show that the new German rules sanction the use of much lighter weight tubes than called for by the A.S.M.E. Code.

The new German rules were prepared by a subcommittee of the German Steam Boiler Code Committee of the Vereinigung der Grosskesselbezitzer e.V. (Association of Boiler Proprietors, Inc.). An abstract of an article by E. Lupberger entitled "Discussion of Heated Tubes in High-Pressure Boilers," appeared in the December, 1931, issue of *Mechanical Engineering*.

TABLE 5 COMPARISON OF TUBE THICKNESSES REQUIRED BY THE A.S.M.E. BOILER CODE RULES AND THE NEW GERMAN RULES

Outside diam of tubes in in.	Formula used[a]	Maximum allowable pressure, lb per sq in. 200	400	600	800	1000	1200	1400	1600	1800	2000	2500
1½	A	0.077	0.093	0.110	0.127	0.143	0.160	0.177	0.193	0.210	0.227	0.268
	B	0.055	0.069	0.083	0.097	0.110	0.122	0.134	0.146	0.157	0.167	0.193
	C	0.052	0.064	0.075	0.086	0.097	0.107	0.117	0.127	0.136	0.146	0.167
2	A	0.089	0.111	0.133	0.156	0.178	0.200	0.222	0.245	0.267	0.289	0.345
	B	0.060	0.080	0.099	0.117	0.135	0.151	0.167	0.183	0.198	0.213	0.246
	C	0.056	0.072	0.088	0.102	0.117	0.131	0.145	0.158	0.171	0.183	0.212
3	A	0.114	0.147	0.181	0.214	0.247	0.281	0.314	0.347	0.381	0.414	0.497
	B	0.071	0.101	0.130	0.157	0.184	0.210	0.234	0.258	0.281	0.303	0.354
	C	0.066	0.089	0.113	0.135	0.157	0.179	0.199	0.219	0.239	0.258	0.303
	A	0.139	0.183	0.228	0.272	0.317	0.361	0.406	0.450	0.495	0.539	0.650
	B	0.082	0.122	0.161	0.198	0.234	0.268	0.301	0.333	0.363	0.393	0.462
	C	0.073	0.106	0.138	0.168	0.198	0.226	0.254	0.281	0.307	0.333	0.393

[a] Formula A is from the A.S.M.E. Boiler Code; B and C are from the new German rules, B being for tubes of steel having a tensile strength of 49,800 to 64,000 lb per sq in. and C for steel having a tensile strength of 64,000 to 78,000 lb per sq in.

TABLE 6 COMPARISON OF ALLOWABLE WORKING PRESSURES FOR VARIOUS TUBE SIZES AND GAGE THICKNESSES CALCULATED BY THE A.S.M.E. BOILER CODE RULES AND THE NEW GERMAN RULES

Outside diam of tubes in in.	Formula used[a]	Minimum gage, B.W.G. 12 t = 0.109	10 t = 0.134	8 t = 0.165	6 t = 0.203	4 t = 0.238	2 t = 0.284	1 t = 0.300	0 t = 0.340	00 t = 0.380	000 t = 0.425
1½	A	590	890	1262	1718	2138					
	B	988	1398	1954	2721	3530					
	C	1235	1747	2442	3404	4410					
2	A	380	605	884	1226	1541	1955	2099	2459		
	B	711	994	1369	1868	2371	3110	3389	4144		
	C	889	1242	1711	2335	2964	3887	4235	5180		
3	A	170	320	506	734	944	1220	1316	1556	1796	2066
	B	455	630	856	1148	1432	1831	1976	2358	2767	3262
	C	569	788	1070	1435	1790	2289	2470	2947	3460	4080
4	A	65	177	317	488	645	853	924	1104	1284	1487
	B	335	461	623	828	1026	1297	1395	1648	1913	2228
	C	419	576	779	1035	1283	1621	1744	2060	2391	2784

[a] Formula A is from A.S.M.E. Boiler Code; B and C are from the new German rules, B being for tubes of steel having a tensile strength of 49,800 to 64,000 lb per sq in. and C for steel having a tensile strength of 64,000 to 78,000 lb per sq in.

This article contained diagrams (Fig. 13) which showed the stresses in a tube 83 mm (3.27 in.) outside diameter when subjected to radiant heat. The diagrams gave the stresses for various wall thicknesses and for three rates of heat transfer. The stresses for the different rates of heat transfer were computed on the basis of the elastic theory, and the results secured by using the old German rule and the new German rule for the milder steel (formula (B) of this paper) were shown in the diagrams. On these the writer marked by crosses the thickness required by the A.S.M.E. Boiler Code rules for a tube 3.27 in. outside diameter at the different working pressures which are given in atmospheres on the curves. For example, in the curves marked 60 metric atmospheres (853 lb per sq in.) the thickness of a 3.27 in. diameter tube required by the A.S.M.E. Code would be 0.239 in. (6.08 mm), and a cross is placed on each of the three curves for 60 atmospheres, corresponding to 6.08 mm. The diagram as marked showed that the A.S.M.E. Code thicknesses approach the most advantageous thickness given by the curves more nearly than those obtained by the new German rule for heat transmission of 36,880 and 18,440 Btu per hr per sq ft, whereas at the higher rate of heat transmission of 73,750 Btu per hr per sq ft the new German rule approaches more nearly to the theoretical figure than the A.S.M.E. rule. They also showed that the A.S.M.E. Code rules call for somewhat thinner tubes in most cases than the old German rules.

The heat-transfer rates given in Mr. Lupberger's paper are much lower than those which exist in power-plant practice, where 100,000 Btu per sq ft and considerably over must often be provided for. It should be borne in mind that the most highly heated portion of a tube must be considered. The modifying results of plastic flow would reduce the maximum stresses below those shown by the elastic theory and the stresses for the lower transfer rates by the elastic theory would approach the maximum stresses which would actually exist at some higher rate of heat transfer.

Mr. De Baufre's analysis shows that for a tube 3 in. outside diameter, having a wall 0.425 in. thick and subjected to a working pressure of 1840 lb gage, or 125 atm, the maximum stress for a rate of heat transfer of 90,000 Btu per hr per sq ft is about 17,000 lb per sq in.

It will be seen on comparing this stress with the stresses given in the diagrams (Fig. 13) of Mr. Lupberger's paper that it is considerably less than would be obtained by an analysis where the effect of creep is not included. If Mr. De Baufre will add to his paper curves similar to those given in Fig. 13, it would be a valuable contribution. It would also add greatly to the value of the paper to give the stresses for heat-transfer rates of 125,000 and 150,000 Btu per hr per sq ft.

The Boiler Code Committee has sanctioned the use of higher carbon-steel tubes than specified in the code, with an upper carbon limit of 0.35 per cent and having a minimum tensile strength of 60,000 lb per sq in., in order that these tubes may be used under conditions where they are subjected to radiant heat and the rate of creep is an important element. These higher carbon-steel tubes have been sanctioned for use at pressures not to exceed those allowed by the code for lower carbon-steel tubes of the same thickness. This limitation does not count against the advantages secured in using the tubes under conditions where they are subjected to radiant heat and a high heat transfer.

Summarizing, it seems to the writer that for ordinary operating conditions the tube thicknesses specified in the A.S.M.E. Boiler Code are as nearly correct as can be shown by any theory. Extreme conditions of heat transfer should be met by employing tubes of a higher carbon content or of special alloy steel of the same tube thicknesses as given in the code for tubes having a lower carbon content. For small-diameter tubes of forced circulation boilers it is advantageous under certain conditions to reduce the thickness below that called for by the A.S.M.E. Code.

GEO. A. ORROK.[4] The mathematics of the paper is clear and logical, and the picture of the processes taking place in the stressed metal as presented is clear and convincing. The author gives ample mathematical formulas for the calculation of any design problem of boiler-tube surfaces, presenting also those situations where the lessons of experience are indicated. The writer is not going to discuss the mathematical side of the paper, but will call attention to the question of temperature margin with some suggestions from operating experience.

The cost of maintaining boiler surface in operating condition includes the regular cleaning and renewals of deteriorated surface at stated periods, as well as those much more expensive and annoying forced-maintenance periods caused by the appearance of blistered, burned, or burst tubes. If an increased thickness of boiler-tube metal would reduce the second part of the expense, then the additional cost of metal would be amply repaid. But unfortunately the greater thickness of tube metal is not a guarantee against the effects of scale accumulations, poor circulation, flame impingement, and their consequent troubles.

A boiler tube exposed to the radiant heat of the furnace operates at a temperature margin to softening of perhaps 200 to 300 F, and the thicker the tube the lower must be this temperature margin. The gradual increasing of tube thickness above the code value is equivalent to wrapping the tube with successive layers of more and more plastic metal, which tends to increase the stress and strain on the inner layers until the latter also become plastic and creep sets in. Thus a thick boiler tube wall will have some internal layer truly elastic covered both inside and out by plastic layers. It is evident that the greater the thickness of the tube with the same steam temperature the higher will be the temperature of the outside of the tube and consequently the

[4] Consulting Engineer, Orrok, Myers & Shoudy, New York, N. Y. Mem. A.S.M.E.

less temperature margin for meeting the uncontrollable exigencies of operation. Mellanby and Kerr in England were the first to point out that the temperature margin is the only real factor of safety. We know that in high-pressure boilers a very small amount of scale decreases this temperature margin to a large degree, usually resulting in a bulge or a blowout.

There has been a great deal of discussion about the embrittlement of overstressed metal in riveted joints. We are not sure that the metal of a thick tube overstressed by heat transfer would not develop a widespread embrittlement in certain conditions. Moreover, a comparison of service given by boiler and superheater tubes when exposed to radiation shows that, on the average, the latter have a shorter life. If a worn-out radiant superheater tube is inspected, a substantial loss of weight on the side exposed to the heat can be usually found which is due mainly to outside oxidation. Boiler tubes with a cooler outside surface stand much better. Thus thick tubes will be reduced to thin tubes on many square feet of surface toward the middle and end of their useful life. This difficulty is aggravated when high-carbon and alloyed steels are used with their lower heat conduction characteristics.

On the assumption of one stress cycle per day, the author considers thick tubes as immune to fatigue cracks, but there are many locations in the water walls and in the convection zone where partial stress cycles will be occurring every second or so due to peculiarities of circulation of both water and gas, and the writer is not so sure that fatigue cracks would not develop under such circumstances. Certain difficulties of this type have been manifest even in low-pressure boilers. It is needless to say that the higher the outside tube-metal temperature the more the situation is aggravated.

Our many experiences with boiler tubes which have been deformed by pressure and heat lead also to the conclusion that it is not only the mean temperature of the tube wall which is of importance, and again following Mellanby and Kerr, the main criterion is the temperature of the outside surfaces of the tube. When this temperature exceeds the creep limit, deformation invariably takes place, even though the mean temperature of the wall is higher than the creep-limit temperature figuring on the whole section. Our efforts should be directed toward keeping the temperature of the outside tube surface exposed to radiant heat within proper bounds, in which case the mean temperature of the tube metal will take care of itself and be safe enough for all purposes.

Those of us who have seen tube and header walls of thicknesses exceeding $^3/_8$ or $^1/_2$ in. after exposure to radiant heat can testify to the extensive deterioration which invariably takes place.

All the properties of steel defining its stability under stress in boiler operation are adversely affected by temperature, and the writer's preference is always for the coolest tube under the operating conditions.

The author considers that the safest thickness for boiler-tube walls is that thickness in which the ratio of the internal fluid pressures to the proportional limit stress at the mean wall temperature is the highest, which statement, carried to its logical conclusion, predicts an infinitely thick tube. The writer is inclined to believe that this statement requires further argument. As a corollary he also suggests that the safety of a tube may be increased by using materials with a higher proportional limit stress. This would without doubt be true if the heat conductivity of the material were of the same order. But it appears also to be true that as the carbon content or the alloy percentage of the steel increases, the coefficient of heat conductivity is cut down to about one-third of the figures for ordinary mild steel, thus increasing the outside temperature of the tube by many degrees for the same heat conductance. From data taken from the International Critical Tables, the coefficient of heat transfer for ordinary tube steels is of the order of 45 to 48, decreasing to as low as 16 with the increase of carbon content or of alloying materials. The data, however, are fragmentary, and much more actual experimental work is required with various samples of steel before judgment should be passed.

His conclusions regarding the coefficient of heat transfer from the inner surface of the tube wall to the water-steam mixture are most interesting and show well the dangers of poor circulation and scale accumulation.

The appendixes will be valuable to all engineers who are concerned with boiler design, and it is hoped that with the aid of such tools as these we may get a much better understanding of the physical processes lying behind boiler-tube failures.

N. Arstay.[5] Many authors have treated this subject of boiler-tube stresses, but this is the first paper in which a concrete example has been treated exhaustively and all factors have been taken into account. Moreover, it gives a clear picture of the successive states in the tube metal. The case in its general aspect may be stated briefly thus: A tube subject to inner pressure and high heat input from the outside develops tension stresses at the inner surface which are beyond the creep limit, and hence the tube will deform, releasing the stress. Successive operating runs of the boiler will effect a residual compression of the inside fibers which crept during the operation. Thus on each starting of the boiler before stressing the inner fibers to full tension, the compression would be relieved, and hence the maximum operating tension strain and stress due to pressure and heat transfer will be less than that calculated for a perfectly elastic tube in the initial state. In this sense the tube will become very much like any big gun built of cylinders shrunk one on the other with pressure decreasing outward. It is clear that the full effect of creep in the most stressed fibers will ultimately amount to the reduction of total stress at the inner fibers to the zero creep stress, or a very slow creep, according to the other viewpoint, at the temperature of metal. In other words, the thermal stresses are of secondary or less importance, being mostly relieved by creep.

The question now is which tube will be safer—the thick one preset by the creep or the thin one with low thermal stresses. This question is still unanswered. We know that, in general, tubes of A.S.M.E. Code thickness fail only in locations of scale accumulations or in locations where flame impingement produces heat inputs well above the 90,000 Btu per hr per sq ft used in the paper as maximum. The general tube distortion has been observed only in overheated superheater tubes, which, by the way, usually are relatively thicker.

The useful temperature range for the metal is narrow, let us say from 550 to 1100 F, being defined below by the saturation temperature and above by the serious softening of the metal. The cleanliness of the inner surface will affect the metal temperature greatly, and in the hot spot due to scale there will be creeping and a bulge, irrespective of the initial thickness of the tube. It will take a somewhat longer time for a thick tube to develop the bulge, but the result will be identical with the case of a thin tube. It is obvious that if the scale settles in the tubes, the process of bulging in both the thick and the thin tube will be the same, separated only by a time interval which is insignificant compared with the usual operating runs. Both tubes will suffer equally, and there is no marked advantage in the thick tube. Recently, the German boiler code has been revised and thinner tubes specified.

In the case of flame impingement we must distinguish between the continuous impingement due to an unfortunate furnace and

[5] Principal assistant engineer of Geo. A. Orrok, Orrok, Myers & Shoudy, New York, N. Y. Mem. A.S.M.E.

burner design and the occasional impingement, when the flame licks a tube now and then, here and there. For the continuous impingement with heat inputs upward of 120,000 Btu per hr per sq ft, allowing for creep, the tube will be stable in a continuous operation when the pressure stress in the inner fibers is under the zero creep stress at the temperature, irrespective of the thermal stress. When shutting down the boiler, the thermal stresses in the tubes will be relieved. If the metal temperature is below the stress-relieving point, the permanent set will maintain the inner compression and outer tension, but if the tube is kept several hours at a temperature in which these forces are relieved, the tube will return to the initial stress state, but not to the initial shape, and thus the creep deformations will be cumulative with every lighting up. Banking periods with tubes not quite clean will be very favorable for this release of residuary stresses. It is very difficult to say without experimental data just in which direction the accumulation of deformation will go. For the intermittent impingement the question is more indefinite; on one hand, we have a more frequent change of stress and release of residuary stress favoring residual deformation, but on the other hand, the time for creeping is reduced.

Some years ago the writer made a study of stresses in the connections of longitudinal fins with tubes in the usual water-wall construction. The stresses are enormous if the metal of the fins does not yield, and the writer has found that the fins and the welds are entirely plastic, closely following, without serious stresses, the thermal changes. That was the only possible explanation of the usually good service performance. It should be noted here, however, that the creep in this case is almost wholly the compression yield in the fin, when the boiler is lighted. The high temperature of the fin's metal allows a fast releasing of the residual tension stress when the boiler is being shut down, so that the metal of the tube is not affected adversely. We have many instances where fin tubes have given several years of splendid service, and this shows that a plastic state in certain portions of the tube metal has no inherent danger.

It becomes more and more obvious that the creep depends upon many factors. Unfortunately, very little is being done to ascertain the effect of compound stresses. The shear creep tests have just begun. The change in density of the metal which crept, if such changes occur, has not been compared with the change of density in metal worked cold to the same residual deformation. It would be very interesting to rupture in tension a few specimens of tube steel which were subjected to various extensions by creep and to see how a degree of extension by creep affects the tensile properties. The writer does not know of any series of experiments in which boiler tubes were tested to destruction, and the need for such experiments is apparent at present when the high steam pressures are becoming common.

This question of boiler-tube thickness is sufficiently important to warrant a series of experiments, which might be as follows:

(1) Selection of a hard-driven high-pressure boiler which is losing tubes and insertion of a few tubes of varying thickness in the water walls of first rows. The comparative service given by the tubes of various thickness will be the indication by which to judge the economy of greater metal thickness.

(2) A series of experiments with a small forged-steel drum and several U-tubes of various wall thicknesses connected to the drum and subjected on some definite portion to a very high rate of heat transfer by radiation—for instance, by placing them in a carborundum cavity heated intensely and to the direct flame impingement of gas burners, the observer to be protected of course. If this program should be carried out, then the processes of tube failures due to thermal and pressure stresses might be ascertained. So many factors usually enter into the common tube failures that the effect of tube thickness is never observed nor ascertained.

C. F. Hirshfeld.[6] One of our findings in Detroit may offer a means of advancing knowledge in the field under discussion. We deal here, among other things, with the question of plastic flow of steel. This is a subject that is rather difficult to study in the laboratory because of the difficulties imposed by the necessity of working with the metal at relatively high temperatures.

In studying the behavior of lead cable sheaths at duct temperatures we found ourselves in the field of plastic flow. It occurred to some of us that there might be a resemblance between the behavior of lead and of steel, except that corresponding phenomena occur at different temperatures. To test the idea we had an investigation conducted at the University of Michigan. This showed that lead at low temperatures becomes quite an elastic material in the sense that it gives the same type of stress-strain curve that steel does at ordinary temperatures. This finding suggests that we might use lead at ordinary temperatures to indicate qualitatively what may be expected of steel at high temperatures. The writer suspects that experiment along such lines would prove fruitful in advancing our knowledge of the type of phenomena concerned in the use of steel at elevated temperatures.

Clarence H. Kent.[7] The problem of the thermal stresses in a tube subjected to heat flow on one side only is entirely new. We have known that creep must relieve thermal stresses in such applications as the tubes of oil stills, and the author's application of the known data on creep to this problem is interesting.

In the past boiler tubes have been treated as thin cylinders, but the increase in thickness necessitated by high pressures has rendered this method less and less exact. When tubes are treated as thin cylinders, the temperature distribution becomes a linear function of the radius, as in Equation [2] under the heading, Axial Stresses in Cylindrical Tube Wall Before Bending, instead of the actual logarithmic function as given in the first part of Appendix No. 1. In calculating stresses set up by temperature difference or by external forces the radial stress is neglected in the case of thin cylinders. In the examples which the author has considered, the radial stresses have been found to be too low for practical consideration.

The axial and circumferential thermal stresses of Fig. 3 would have shown a maximum variation of approximately 11 per cent from the curves shown had the cylinders been treated as thin. Thus no very large error is introduced even with the cylinders considered in using the very simple relations for these stresses developed for thin cylinders: $\max t_t = s_t = \pm \frac{E\alpha m}{2(m-1)}(T_2 - T_1)$ in the author's notation. If the use of the more complex equations given on the third page is desirable, one is referred to *Engineering*, vol. 124, 1927, p. 443, in which Mr. L. H. Barker presented some very useful alignment charts for the solution of these equations.

It will be noticed that the sum of the radial and tangential thermal stresses equals the axial stress, or $p_t + t_t = s_t$, as shown by the author's equations or curves. This is true for all kinds of thermal stresses in cylinders when the distribution is symmetrical about the axis of the cylinder, and forms a quick check on the accuracy of numerical results.

Attention should probably be called to the fact that the equations for thermal stresses presented in the paper may be applied only to that portion of the tube not near the ends. Obviously the thermal axial stress must reduce to zero at the free ends, and the circumferential stress may increase as much as 35 per cent

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[7] Professor of Mechanical Engineering, Pennsylvania State College, State College, Pa. Mem. A.S.M.E.

above that calculated for the middle section. The exact distribution will depend, of course, on the end conditions. The length of this end section in which the equations developed do not apply is in turn dependent upon the cylinder dimensions and probably would not exceed 12 inches in this case. The writer gave solutions of this problem for various end conditions for thin cylinders which could be applied with reasonable accuracy in this case.[8]

At the top of the sixth page the author is apparently justifying the use of the maximum-stress theory in determining the point at which the material reaches its elastic limit. On the page preceding he has determined the circumferential strain by dividing the circumferential stress by the modulus of elasticity, instead of by the more usual formula: $e = \frac{1}{E}\left(t - \frac{s}{m}\right)$, if the radial stress is neglected and e represents the unit circumferential strain. If one applies the strain-energy theory for ductile materials, one finds that it would require a single stress of 29,100 lb per sq in. to equal the maximum applied circumferential stress of 26,344 lb per sq in. and axial stress of 22,561 lb. If the proportional limit in Fig. 4 is 14,000 for a single stress, the same strain energy will be produced by two equal principal stresses of 11,830 lb per sq in.

The choice of one or other of these theories would not affect the method used by the author to determine the probable circumferential stress, but the use of the latter theory, for instance, would increase the spread between probable and calculated values in Fig. 9. Lack of data prevents general agreement on any particular theory of combined stresses, but it is usually considered that the proportional limit is less for combined stresses of this character than for single stresses.

Referring to the first part of Appendix No. 1, the writer questions whether it is necessary to repeat here the derivation of standard formulas given in the textbooks. The principle of superposition shows at once that the total stress in the tube wall is the sum of those produced by internal pressure and those produced by temperature difference.

In the second part of Appendix No. 1 the author has developed expressions for stresses in a tube subjected to heat flow on one side only. The writer believes it is possible to obtain more exact values for the stresses involved, still treating the tube as a thin cylinder, as the author has done. If we consider the tube split into two semi-cylinders, that part subjected to heat flow will experience a change in curvature in two directions—one parallel to the axis and the other at right angles to it. The change in radius of curvature is given by $r = \frac{t}{\alpha(T_2 - T_1)}$; this change of curvature may be removed by the application of moments in the two planes considered whose value is $M = -\frac{EIm}{m-1}\frac{\alpha(T_2 - T_1)}{t}$. The stresses produced by these moments in each of the two directions are $\max s_l = t_l = \pm \frac{E\alpha m}{2(m-1)}(T_2 - T_1)$, which is the same as the stresses in a thin plate subjected to the above moments in two planes. This compares with the author's formulas [9] and [10] under the heading referred to, in which he has omitted the factor $m/(m-1)$, as he has done also in Equations [7] and [8].

By the application of these moments, that half of the tube subjected to heat flow is now straightened in the axial direction and restored to its original curvature in the circumferential direction. It must then be joined to the half in which there is no heat flow and which is subjected to no moments. This will produce bending in a plane through the axis until the moments in the two halves are equal, as the author has considered. But there must also be both moments and shear applied to the edges of the two halves to bring the edges together and then to equalize the slope of their walls. These the author has not considered as affecting the circumferential stresses and through these the axial stresses. Upon these thermal stresses we must superpose those due to internal pressure.

A. E. R. DE JONGE.[9] Dr. Lohrisch has shown[10] that the distribution of heat absorption in a boiler tube is qualitatively about as indicated in Fig. 42. If a front-row tube is considered, the heat absorbed by radiation has to be superposed so

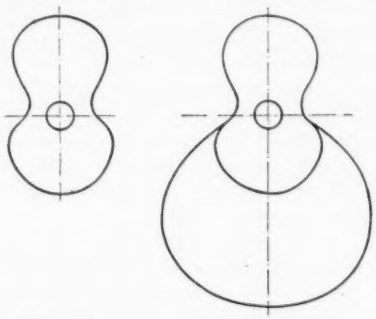


FIG. 42 FIG. 43

that the heat absorption of the boiler tube would be changed somewhat as indicated in Fig. 43. The writer would like to ask the author how this distribution of heat absorption will modify the stresses from his assumptions—i.e., how the longitudinal and the circumferential stresses will be distributed. It seems that the longitudinal stresses are somewhat eased, while the circumferential stresses become worse and probably set up a bending over the transverse cross-section of the tube. It would be interesting to hear from the author what their effect will be on the tube—i.e., both in longitudinal and transverse direction.

AUTHOR'S CLOSURE

Referring to the suggestion of Dr. Jacobus that curves be added to the paper similar to those given by Mr. Lupberger and that stresses be given for heat-transfer rates of 125,000 and 150,000 Btu per hr per sq ft, time has not provided the opportunity to do so. It might be added that while the curves of Mr. Lupberger do not show the maximum stresses to which the tube wall is subjected by reason of creep with high rates of heat absorption, they do show the ranges in stress; for to the extent that tension is relieved by creep, a compressive residual stress is set up in the material. If temperature stresses contribute to the failure of a boiler tube, the failure must be due to fatigue at the inside surface, which will be a function of the stress range.

Referring to Mr. Orrok's comparison of boiler tubes and superheater tubes, it may be mentioned that the paper is limited to boiler tubes. The conditions in superheater tubes are different from boiler tubes by reason of the higher temperature of the fluid within the tube and the lower rate of heat transmission from the inner tube wall to this fluid.

Mr. Orrok also refers to the lower heat conductivity of high-carbon and alloy steels as compared with the mild carbon steel usually used in boiler tubes. For mild steel boiler tubes, the heat conductivity of the metal is around 30 Btu per hr per sq ft per deg fahr for 1 ft thickness under ordinary temperature conditions, as shown by Fig. 41, instead of 45 to 48 as mentioned by Mr. Orrok. For medium carbon steel, 0.35 per cent

[8] "Thermal Stresses in Thin-Walled Cylinders," Trans. A.S.M.E., 1931, APM-53-13, p. 167.

[9] The Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

[10] *V.D.I. Forschungsarbeiten*, heft 322, pp. 46–68 (1930).

carbon, the heat conductivity will be reduced only to about 28 instead of to 16 as mentioned by Mr. Orrok. Even 3.5 per cent nickel will only reduce the heat conductivity of steel to 23 to 25, depending upon the carbon percentage. The slightly higher metal temperatures due to these decreases in heat conductivity would not counterbalance the advantages of the higher creep limit with medium carbon steel and with 3.5 per cent nickel steel as compared with mild carbon steel in boiler tubes.

In answer to Mr. de Jonge, the conditions shown by him are less severe than assumed in the paper for a tube receiving heat on one side only. The absorption of any heat by the rear of the tube would reduce the axial stresses as calculated in the paper. The discussion of Professor Kent, however, shows a better method of attack than used in the paper for calculating the stresses in a tube wall with non-uniform heat absorption, and it would be interesting to apply this method to the heat-absorption distribution shown by Mr. de Jonge.

The use of thinner tubes than specified by the A.S.M.E. Boiler Code would have a number of advantages, as listed in the article referred to by Dr. Jacobus; and the practical results of the German experiment in using such tubes will undoubtedly be watched with interest by all manufacturers and users of high-pressure steam boilers. One practical disadvantage to the use of thin tubes, however, is the absence of any margin of safety where corrosion is likely to occur, either externally as mentioned by Mr. Orrok or internally. Unfortunately, a thick tube wall would not prevent failure of boiler tubes by poor circulation of the water-steam mixture within the tubes, by accumulation of scale on the inner surface of the tube wall, by flame impingement on the outside of the tube wall, by corrosion of either the inner or outer surfaces, or by defective material.

The author agrees with Dr. Jacobus that the tube thicknesses specified in the A.S.M.E. Boiler Code are as nearly correct as can be shown by any theory. It is questioned, however, whether thinner tubes should be used for small diameter tubes of forced circulation boilers.

FSP-55-7

Pulsation of Air Flow From Fans and Its Effect on Test Procedure

By HAROLD F. HAGEN,[1] BOSTON, MASS.

Error that results from velocity-pressure determinations of volume when a pulsating velocity exists is discussed theoretically. The paper also shows experimentally that such pulsation is of frequent occurrence, and gives quantitatively the amount of the error. It further suggests necessary changes in fan-testing procedure.



THE method of testing fans that has been in use for the past decade is based on a volume determination obtained from a weighted average of velocity pressures. The procedure has been coded, and the resultant standardized testing has proved of great help to both manufacturer and user. However, certain inconsistencies have appeared in the development work of the industry from time to time until the weight of evidence that something is wrong with the accepted method of testing can no longer be ignored. The most important of these inconsistencies may be described briefly.

A few years ago an authoritative laboratory reported that the total efficiency of a certain fan could be increased by the use of an expanding chimney on the fan outlet, the so-called evasée stack. As no ultimate work can be done in a stationary passage, the reported increase of efficiency violated the general law of the conservation of energy, and Bernoulli's law in particular. The author secured confirmation of the reported result when using standard test procedure, but a simultaneous measurement of the volume with a flow nozzle showed 18 per cent less volume than was indicated by the pitot-tube traverse in the large duct. With this reduction in volume, the effect of the chimney was found to be a slight decrease in total efficiency, which of course was in obedience to natural laws and the effect of friction. At the time, poor distribution of air velocities in the large duct was accepted as the explanation of the error. The explanation was disquieting, however, because there was no provision in standard test methods to guard against such a false result.

Recently, propeller fans with only two blades were introduced, which on test showed efficiencies unreasonably high. Tests on multi-blade propellers of the same design showed much lower efficiencies. This decrease in efficiency with increasing number of blades seemed to require explanation.

A third element came from the apparently unexplainable agreements and differences encountered in attempting to check pitot-tube traverse determinations with flow-nozzle volume. In many cases agreement between the two would occur with great exactness; in other cases, even when the extenuation of poor distribution did not apply, differences of a magnitude beyond any reasonable test inaccuracy would appear.

If there were a pulsation in the air, all these inconsistencies would become understandable. In the large duct from the expanding chimney a pulsation could produce a large error. Fewer blades, as in the case of the propeller fans, might tend to create a more violent pulsation and falsely indicate a higher efficiency. And the differences and agreements between nozzle and pitot-tube volumes would be explained by the presence or absence of pulsation. The hypothesis appeared worthy of experimental investigation, particularly when it was realized that with the instruments used for air measurement a high-frequency pulsation would escape detection.

The existence of more or less cyclic variations of air velocity had been noted by various investigators. It had been observed in wind-tunnel experimentation. Also, in connection with some measurements on the intake flow to internal-combustion engines, the possibility of pulsating flow had been realized and the resultant error in pitot-tube indications had been calculated. Experimenters, working with a special hot-wire anemometer, had taken oscillographs of pulsation in both air and water currents.

A velocity wave is accepted, and accepted without question, when the actuator is a displacement engine. The code for testing reciprocating compressors specifies equipment designed to eliminate velocity pulsation. But it has been assumed that a fan rotor gives a steady flow, and all fan testing has been based unquestioningly on this assumption.

The error introduced by unsuspected pulsation is somewhat analogous to the difference between the average and the effective values of a rectified alternating-current wave. In the electrical measurement the effective value wanted is the square root of the average of the instantaneous squares. The usual direct-current ammeter gives, instead of this desired value, the average of the first power of the current. With air measurements we want the average of the first power of the air velocity and we get with our instruments the average of the instantaneous squares.

Wave Errors

The possible error may readily be seen from a consideration of the following simple wave forms. Consider an ideal square wave rising instantaneously to a velocity of 133.3 fps, continuing at this velocity for a definite time interval, then dropping instantaneously to zero, and remaining so for the same time interval. Fig. 1 shows this wave graphically. The average velocity is 66.67 fps, and the volume flowing is correctly given by multiplying this velocity by the cross-sectional area of the duct.

The customary procedure with pitot tube and manometers would give a quite different result. With an air density of 0.075 lb per cu ft, the 133.3 fps would be equivalent to 4 in. of water. The average velocity pressure of the pressure wave is 2 in., and this would be registered by the U-tubes. The velocity calculated from this indicated 2 in. is 94.2 fps. We know from our assumption that the average velocity was only 66.67 fps, so that a volume 41 per cent high would be reported. This square

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[1] Vice-President in Charge of Research, B. F. Sturtevant Co. Mr. Hagen was born in Brooklyn, N. Y., on November 14, 1884. He received the degree of M.E. from the Stevens Institute of Technology in 1907. He has been engaged in research work in the fan industry since 1911. Many of the large ventilating and mechanical-draft installations throughout the country, of which the Holland Tunnel is an example, are equipped with fans of his designing.

Contributed by the Power Test Codes Committee and presented at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

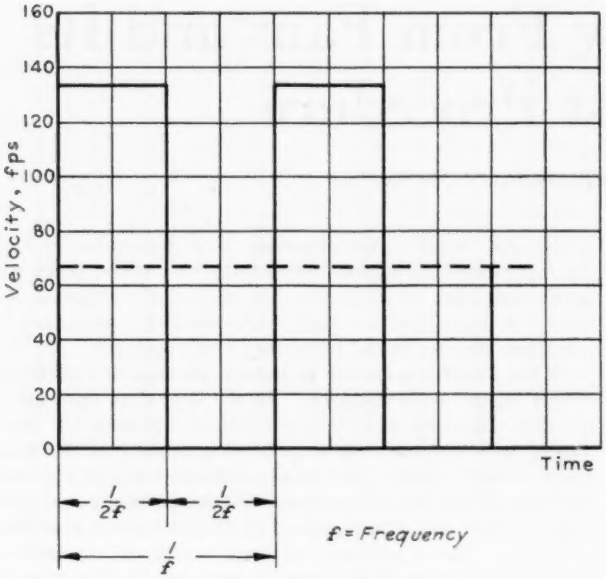


FIG. 1 SQUARE WAVE

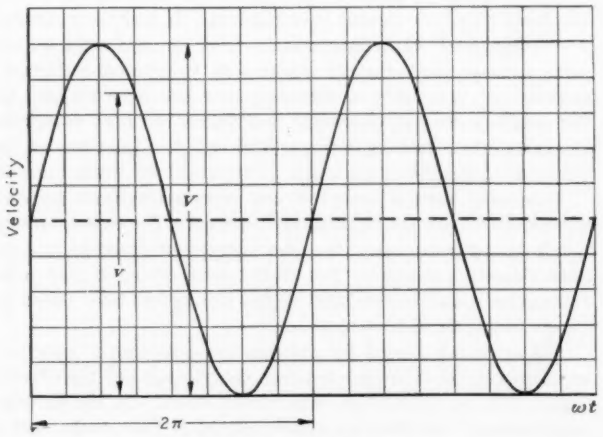


FIG. 2 SINE WAVE TO ZERO

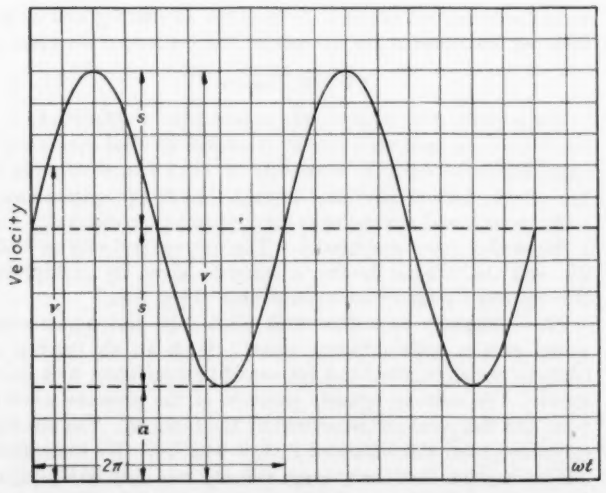


FIG. 3 SINE WAVE TO MINIMUM a AND WITH AMPLITUDE $2S$

wave is of course an impossible condition, but it brings out most clearly the theoretical error under discussion.

A sine wave would more closely approximate true conditions of flow. The sine wave in Fig. 2 represents a pulsation in velocity from zero to a maximum of V and back to zero. The instantaneous velocity v of this wave is

$$v = \frac{V}{2}(1 + \sin \omega t)$$

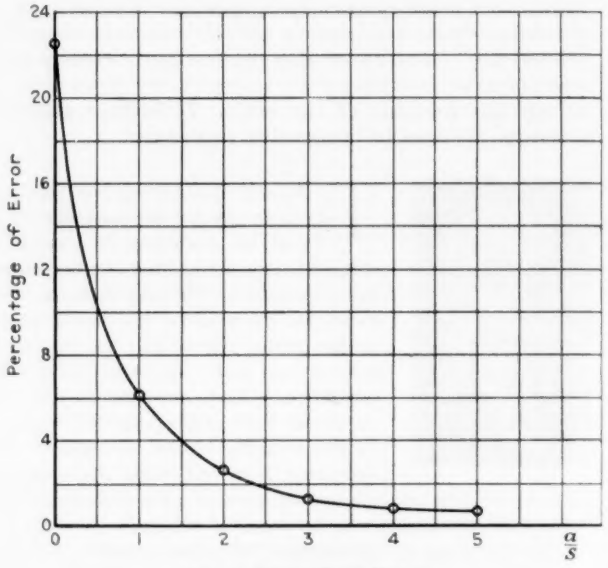


FIG. 4 PITOT-TUBE ERROR
(With sine wave amplitude $2S$ and average $a + S$.)

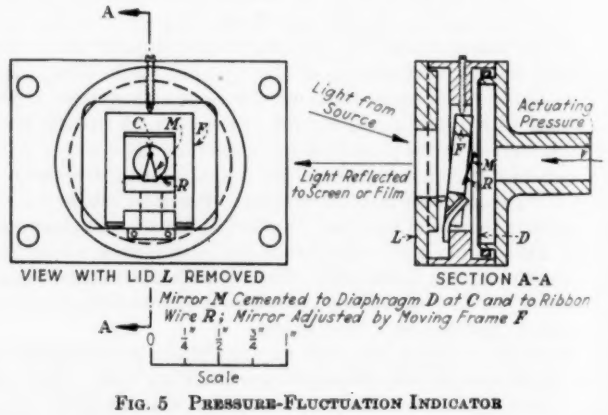


FIG. 5 PRESSURE-FLUCTUATION INDICATOR

and H, the average of $v^2/2g$, is

$$H = \frac{1}{2\pi}\int_0^{2\pi} \frac{v^2}{2g}\, d\omega t = \frac{1}{8\pi}\int_0^{2\pi} \frac{V^2}{2g}(1 + \sin \omega t)^2\, d\omega t = \frac{3}{8}\left(\frac{V^2}{2g}\right)$$

This value would determine the reading on the gages, and the resultant calculated average velocity would be $V\sqrt{(3/8)}$ instead of the correct $V/2$.

The reported volume would be in error in the percentage of

$$\frac{V\sqrt{(3/8)}}{V/2} = 1.225, \text{ or } 22^1/_2 \text{ per cent}$$

A more general case for analysis is shown in Fig. 3, where the sine wave does not drop to zero but to a minimum value a. In this case the maximum velocity $V = 2S + a$, where S is the half-

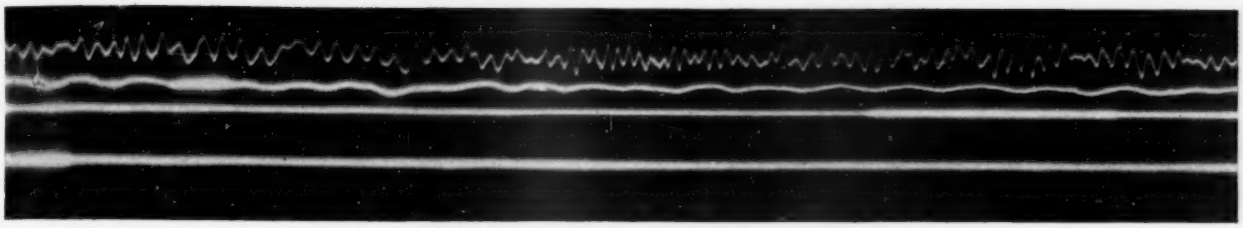


FIG. 6 SPECIAL TEST OF CENTRIFUGAL FAN

(Diagram taken at pitot-tube station. Bottom line, atmospheric; second, calibration line, 0.454 in.; third, average static; top line, impact. Corresponding U-tube readings: Impact, 0.79 in.; static, 0.6 in.; average velocity pressure, 0.2005 in. Film speed, 3.9 in. per sec.)

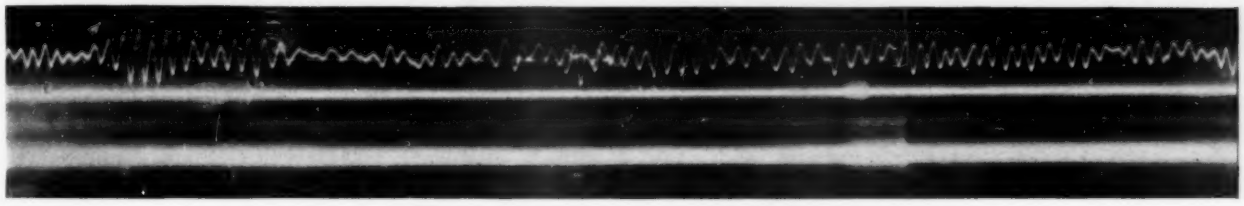


FIG. 7 SPECIAL TEST OF CENTRIFUGAL FAN

(Diagram taken at mouth of nozzle. Bottom line, atmospheric; second, heavy calibration line, 0.54 in.; top line, impact, which in this case is all velocity pressure. Film speed, 3.6 in. per sec.)

amplitude of the wave. The instantaneous value $v = a + S(1 + \sin \omega t)$; and, with H as before,

$$H = \frac{1}{2\pi} \int_0^{2\pi} \frac{[a + S(1 + \sin \omega t)]^2}{2g} \, d\omega t = \frac{1}{2g} \left[(a + S)^2 + \frac{S^2}{2} \right]$$

The true average velocity is v avg $= a + S$, so that the error becomes

$$\frac{\sqrt{\left((a + S)^2 + \frac{S^2}{2}\right)} - (a + S)}{a + S}$$

$$\sqrt{\left(1 + \frac{S^2}{2(a + S)^2}\right)} - 1$$

When $a = 0$, we have the special case of Fig. 2, and the error is $\sqrt{(3/2)} - 1 = 0.225$, or $22\frac{1}{2}$ per cent, as before.

The curve of Fig. 4 shows the percentage of error with varying values of a from 0 to $5S$, or five times the wave half-amplitude. If the error is to be 1 per cent,

$$\sqrt{\left(1 + \frac{S^2}{2(a + S)^2}\right)} = 1.01$$

$$\left(\frac{S}{a + S}\right)^2 = 0.04$$

$$S = 0.2a + 0.2S$$

$$a = 4S$$

Apparently the way to correctness lies in a high a/S. This condition is most readily secured by the use of a flow nozzle at the mouth of which all the head is changed into velocity. In the tests that have been made, the amplitude of the wave appears to be substantially the same at the mouth of the nozzle as it is in the duct, but in the duct the average velocity is small, giving a very small value to a and producing, as a consequence, a small value of a/S and large percentage of error. At the mouth of the nozzle the average velocity is high; a likewise takes on a high value, and the controlling ratio a/S being large, produces a small and perhaps negligible error. Here, then, is the explanation for the differences between the nozzle volume and the pitot-tube volume that, as previously mentioned, have been found in many tests. Further, the peculiar results observed in connection with the expanding chimney fall into line, as in this case, due to the large area of the test duct, the value of a/S became very small, greatly exaggerating the error.

TEST OF PULSATING FAN

To provide a concrete picture of flow vibrations the instrument shown in Fig. 5 was constructed. The figure is almost self-explanatory. A shallow, cylindrical chamber, subjected to the pressure in the duct, is covered with a thin, taut rubber diaphragm D. An adjustable frame F carries a thin metal strip R. (A piece of galvanometer suspension gold ribbon was used.) A small, triangular piece of mirror glass M is cemented near its base to the strip R and at its vertex to the diaphragm D at C. This gives a moving system of small inertia that has a natural frequency very much higher than the frequency of the pulsations of the air in the duct. Figs. 6 and 7 show vibragraphs taken with this instrument.

It will be of value at this point to follow through the actual test results of the fan under test when the pictures of Fig. 6 and 7 were secured. The diagrams of Fig. 6 were taken at the standard pitot station. The lowest straight line was obtained by rotating the film with the diaphragm chamber at atmospheric pressure. The next straight line is a calibration line. The slightly wavy line was secured by connecting the chamber to the static side of the pitot tube. There is often a vibration of considerable amplitude in the static pressure, but the pinholes of the static tube act as an effective throttle. This static line is therefore to be considered only as an average, not as a record of the wave. The usual pitot tube and rubber tubing have passages small enough on both static and impact sides to throttle vibrations, and further have an air column of such length that its natural period of vibration is too slow to follow the high-frequency vibration in the duct. To secure the top line, which shows the wave of the impact pressure, a special short, large-bore impact tube with an air column period less than half the period of the recorded wave was used.

Fig. 7 is similar to Fig. 6, but shows results secured at the nozzle mouth. The atmospheric and calibration lines are located as before. The static line is omitted, as in this case it is identical

with the atmospheric line. The wave is the impact pressure, which becomes at the mouth of the nozzle a pressure due to velocity only. The similarity between the two impact waves of Figs. 6 and 7, both as to form and amplitude, is notable.

Fig. 8 shows the results of test on this same fan, giving both the nozzle results and those determined in accordance with the standard methods of the duct-traverse test. These two sets of curves were obtained by simultaneous measurement, and the difference in volume, approximately 7 per cent, is represented by points on the two curves at the same static pressures.

The test set-up used is shown in Fig. 9. To the fan outlet was connected a transformation leading to the standard cylindrical test duct, with all areas identical. At the far end of this duct an expanding cone with an internal throttle led to a larger duct and a final flow nozzle. The introduction of the large duct permitted

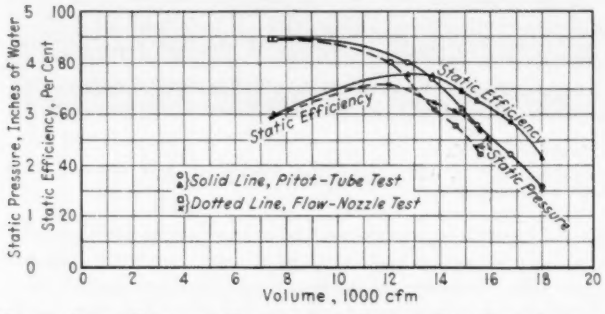


FIG. 8 PITOT-TUBE AND FLOW-NOZZLE TEST OF FAN WITH PULSATING DISCHARGE

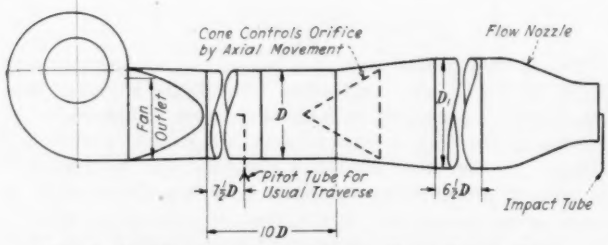


FIG. 9 SET-UP FOR PITOT-TUBE AND FLOW-NOZZLE TEST

the use of a larger nozzle without violating the rule of nozzle proportion—a diameter of mouth not more than half the diameter of the preceding duct—and made possible accurate nozzle determinations of volume throughout the useful range of fan performance. In Fig. 8 this range is from 12,000 to 16,000 cfm. A nozzle connected directly to the end of the smaller test duct would have a maximum area one-fourth of the fan outlet area, and would therefore limit results to the low-volume portion of the fan characteristic, approximately 10,000 cfm, on the curves of Fig. 8. There seems to be no way at present to avoid the increased cost of the test set-up due to the required additional duct.

A nozzle with a mouth diameter approximately seven-tenths of the diameter of the small test duct seems to be most satisfactory where a test is to be made with a single nozzle. The velocity from this nozzle is substantially twice the velocity in the small duct, and the delivery area is sufficiently large to carry the test beyond the useful range of the fan. Where still larger volumes are required, the nozzle mouth may be carried up to the full size of the small duct diameter. In such cases, however, extreme caution should be used and the points secured checked by extrapolation from the points determined from the smaller nozzle.

It is highly desirable, in dealing with a fan that produces a pulsating flow, to use at least one additional nozzle. By means of the conical throttle the same volume point may be secured from both nozzles and the test points compared. An additional nozzle approximately six-tenths of the small-duct diameter is extremely useful in making certain of correct results.

ADDITIONAL TEST DATA

Many tests of fans of different types have been made. Some fans show close agreement between pitot-tube traverse and nozzle; others show discrepancies as high as 18 per cent. Table 1 gives some of these results.

TABLE 1

Fan No.	Type[a]	Blades	Static pressure	Volume—Pitot	Volume—Nozzle	Percentage of error
1	B-S	8	1.5	9,200	9,210	None
2	B-S	8	0.79	11,430	9,825	16.5
3	B-D	8	1.8	14,290	14,370	None
4	B-S	12	1.9	12,680	12,000	5.7
5	B-S	16	0.88	11,520	10,460	10.1
6	B-D	16	2.42	11,720	11,120	5.4
7	B-D	16	2.06	15,250	15,400	None
8	R-S	12	7.25	4,750	4,780	None
9	R-S	12	8.85	5,250	5,030	4.4
10	R-S	12	6.24	4,825	4,460	8.2
11	R-D	12	0.56	25,250	25,900	None
12	R-D	24	1.82	15,100	14,960	None
13	F-S	48	2.1	13,300	11,250	18.3
14	F-S	60	6.0	11,000	10,600	3.8

[a] B denotes backwardly curved blade; R, radial tip blade; F, forwardly curved blade; S, single inlet, D, double inlet.

No specific conclusions can be drawn from these results as to the cause of pulsation. Actual pulsation seems to exist in all fans, but in some the amplitude of the wave is sufficiently small so that errors in measurement do not occur. In other cases, the amplitude is sufficiently large to make wide differences. An error of 10 per cent is not unusual, and in fan No. 13, in Table 1, the error ran as high as 18 per cent. This was the fan, referred to before, that was provided with an extreme expanding chimney. The table appears convincing, however, in the matter of test procedure. It is not safe on any fan to assume the absence of serious pulsation, and when violent pulsations are present, theoretically even a nozzle measurement may be appreciably in error.

Test procedure therefore ought to result in a determination of the presence or absence of pulsation error. With the instruments at present available to the industry, this information can best be secured by making two volume determinations—one by the usual pitot-tube traverse method and one by a nozzle determination at higher velocity. Many engineers dislike the pitot-tube traverse, but it seems to the author from the work in connection with this investigation that the pitot-tube traverse is an extremely accurate method of determining the average velocity pressure in a duct. If, as the result of pulsation, this velocity pressure is incorrectly interpreted as corresponding to the average velocity, the error is with the interpretation and not with the instrument.

It would seem, therefore, that an agreement between the volume determined by the pitot-tube traverse and the volume measured at the mouth of the flow nozzle should be the criterion of correctness of result. A disagreement between these two volumes could safely be accepted as an indication of a pulsation that must be considered.

In extreme cases of difference between pitot-tube traverse and nozzle volumes and where a high degree of accuracy is required (as, for example, when noting the effect of small changes in fan proportions), an additional run of pipe and an additional small nozzle may be used, employing an auxiliary fan, if necessary, to get the required volume through the system. Agreement between the two nozzle volumes determined at different velocities could serve as the required indication of a correct measurement.

It has been noticeable in all the work that the pulsation error is greatest with the larger outputs and that close agreements

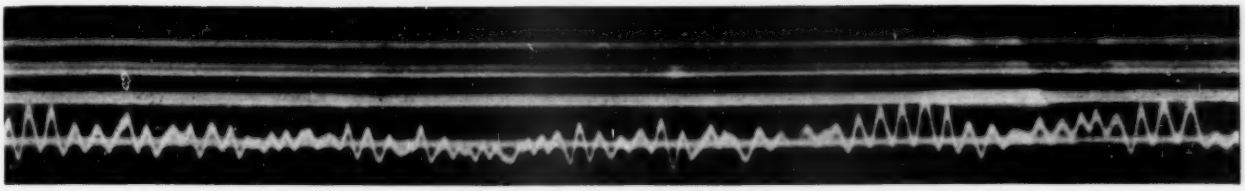


FIG. 10 STANDARD SUCTION-PIPE AND DRUM TEST OF PROPELLER FAN
(Impact tube at pitot-tube station in suction duct. Calibration lines: Top, atmospheric; second 0.2 in.; third 0.298 in.; fourth, 0.714 in. Film speed, 5.24 in. per sec.)

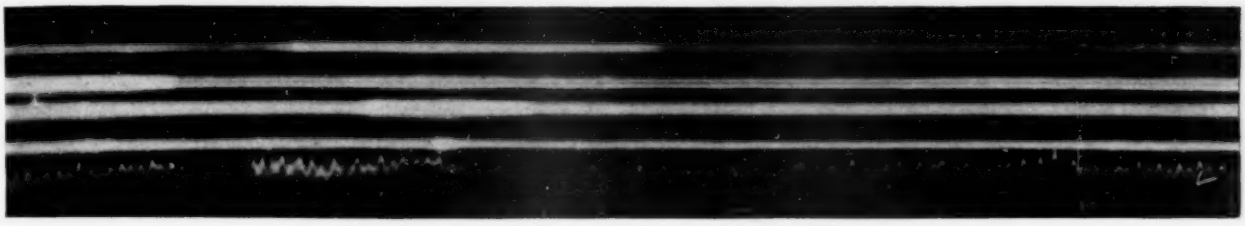


FIG. 11 STANDARD SUCTION-PIPE AND DRUM TEST OF PROPELLER FAN
(Impact tube in drum. Calibration lines: Top, atmospheric; second, 0.266 in.; third, 0.444 in.; fourth, 0.698 in. Film speed, 5.24 in. per sec.)

between pitot tube and nozzle are readily secured at the smaller volumes. This tendency is clearly shown in the curves of Fig. 8 and is typical of practically all the fans tested.

PROPELLER FANS

With propeller fans tested on the outlet side, there seems to be little hope for any consistency. Tests indicate errors of the same general order of magnitude as given in the table for centrifugal fans, but results are widely varying and cannot readily be checked. Contrary to expectation, no difference could be observed in the results of pulsation between two-bladed fans and six-bladed fans. Careful investigation and test reduced the reported efficiencies of these propeller fans to a reasonable point and indicated that the apparently extreme efficiencies were due to a combination of pulsation error and the almost classical mistake of crediting the velocity pressure in the fan intake to static.

The standard procedure for testing propellers provides for readings on the inlet side and the use of a centrifugal fan to overcome the resistances of the duct and drum. Obviously, the results may contain pulsation error, due not only to the action of the propeller fan, but also to the working of the auxiliary, so that propeller fans under customary test methods will give results that vary with the particular centrifugal fan that is connected to the test duct. Even on the inlet side, before the air has passed through the propeller fan, the pulsation appears. Figs. 10 and 11 show vibragraphs taken on the suction side of a two-bladed propeller fan. In this case there was no centrifugal fan used. Fig. 10 shows impact in the inlet duct; Fig. 11 shows impact in the large drum between the inlet duct and the propeller fan. The change of frequency shown in Figs. 10 and 11 is noteworthy. A careful measurement of volume by steam-heater condensation and air-temperature rise indicated an error of 10 per cent in the volume by pitot-tube traverse in the inlet duct corresponding to these diagrams.

It is proposed, therefore, that for the determination of the volumes handled by propeller fans the same general procedure be followed as has been advocated for centrifugal fans—namely, a pitot-tube traverse in the inlet duct and a nozzle determination from a nozzle half the diameter of the duct and discharging directly into the drum from which the propeller fan exhausts. The same requirement for an agreement between the two volumes thus determined would apply.

CONCLUSIONS

From these analyses and tests the following conclusions seem allowable:

1 A variation of velocity can exist in the flow of air past a fixed point.

2 This changing velocity results in an average velocity pressure higher than that corresponding to the average velocity.

3 There can be no certainty of accurate volume determination if the fluctuations are extreme, and consequently accurate testing requires controlling evidence of the magnitude of fluctuation. This evidence may be secured by comparing two volume determinations made at different velocities. A close agreement implies small pulsation and correct results.

4 In most cases, and as customarily employed, the flow nozzle will give volumes more nearly correct than those determined by pitot-tube traverse.

Discussion

[NOTE: The following discussion has been drastically condensed for purposes of economical publication. The complete discussion is on file with the P.T.C. Committee No. 10 on Centrifugal and Turbo-Compressors and Blowers of The American Society of Mechanical Engineers, 29 West 39th Street, New York, N. Y.—Editor.]

J. E. WATSON.[2] Mention is made of the well-known fact that an evasée stack at the outlet of a fan does not increase the total efficiency. As a discussion of the evasée or streamline fan outlet was not relevant, the author did not cover this fully, and it should be pointed out that the total efficiency referred to is that of the fan based upon the fan-test code and not the total efficiency of the entire fan circuit. Varying with the type of fan used and depending upon the outlet velocity, the actual static efficiency of the entire system may be improved considerably by the use of the expanding or streamline chimney on the fan outlet, due to the efficient conversion of dynamic pressure to static pressure.

SANFORD A. MOSS.[3] The computation which the author gives of a hypothetical square wave, as shown in Fig. 1, is well known, but the rest of his mathematical analysis and all of his practical

[2] Chief Engineer, Prat-Daniel Corporation, New York, N. Y. Jun. A.S.M.E.

[3] Mechanical Engineer, Thomson Research Laboratory, General Electric Company, West Lynn, Mass. Mem. A.S.M.E.

analysis are novel as applied to air flow and seem quite accurate. The writer has had two striking illustrations of the phenomenon discussed.

A liquid tachometer was used for measurement of speed of a 500-kw turbo-generator, with a long helical-spring flexible-drive shaft. By pressing a lead pencil against the tachometer shaft while the turbo-generator was at full speed, the speed of the turbo-generator was shown by the tachometer reading to be increased about 10 per cent. Of course, the friction of a lead pencil could produce no such effect on the rotor of the turbo-generator. What happened was that the friction on the tachometer shaft made the helical-spring flexible shaft wind up and then let go periodically. This meant that the tachometer itself was operating with a pulsating speed. The tachometer reads the pressure produced by its impeller, which varies with the square of the speed, so that the tachometer reading gave the square root of the mean square of the speed, instead of the mean speed. The tachometer reading was thus greater than the actual shaft speed for exactly the same reason that an impact tube gives the error with pulsating flow which the author cites.

About 25 years ago, the writer made a test on a displacement blower with perceptible pulsations, with a short length of pipe and a nozzle, at the end of which was an impact tube. The object of the test was to measure the actual delivery of the displacement blower, so as to find by comparison with the theoretical displacement the volumetric efficiency, with the idea of demonstrating the desirability of substituting a centrifugal blower which would not have a volumetric efficiency loss. Previous experience had indicated that the volumetric efficiency was around 80 per cent. However, the test showed a volumetric efficiency of 105 per cent, and the substitution of a centrifugal blower for the displacement blower was not made. It was years before it was realized that the flow measurement was subject to the error which the author discusses.

C. J. Fechheimer.[4] That errors may arise in measurements of air volumes when pulsations are present is well known, and it is timely that false conclusions in fan testing be called to attention. It is believed that when the frequency of pulsation is fairly rapid, as indicated by Figs. 10 and 11, accurate measurements may be made with an electric thermal volume meter. This device, also known as a Thomas meter, consists essentially of an electrically heated element that warms the air and electric thermometers which measure the temperature rise of the air. The air volume can then be computed from the measured loss in the heater, the air temperature rise, and the specific heat and density of the air. There are certain precautions which should be observed with this device which are covered in another paper.[5] There are many advantages which this device possesses, such as the small pressure drop therein, and one meter is suitable for a complete curve of fan performance.

A number of years ago, while making a study of ventilation of alternating-current electrical machinery, the writer had a device built which in principle is similar to Fig. 5. A brief description is given in another paper.[6] In Fig. 10 in that paper a record of the pulsations of pressure is shown, giving the main pulsations and harmonics due to the presence of teeth and slots in the stator of a salient pole alternator. No direct use was made of the data, as means for calculating pressures, air volumes, and their distribution in a machine were obtained by mathematical deduction and empirical constants, but the records assisted toward securing a physical picture of the air-flow phenomenon in that type of machine.

R. D. Madison.[7] Some of the conclusions of the paper are not sufficiently proved by the data offered. Too few data are given and they are not sufficiently correlated. In Figs. 6 and 7 the relative point of rating is not given. One must assume that the fan speed is the same as for test curve of Fig. 8; then for 0.6 in. static pressure, which is beyond the plotted results, the inference would be that the fan is operating near free delivery. Table 1 is decidedly incomplete for a paper of scientific value. No fan speeds are given, no velocity pressures, no indication as to the relative points of rating or even if they are on the same basis, and no values of a/S to prove that where the percentage of error was small the value of a/S was likewise small.

Fig. 6 is no doubt given to show how large the wave amplitude is in comparison with the average velocity pressure. There is no actual proof that there is a velocity pulsation. The wave shown is of the total pressure, and one must subtract from it the static pressure—a pressure which the author himself says is dampened by the small pitot-tube holes and "therefore to be considered only as an average, not as a record of the wave." The author should have used a type of pulsometer similar in principle to that used by Judd and Pheley,[8] in which the diaphragm separates two air-tight chambers and the true velocity-pressure wave is given by the simultaneous difference of the static and total pressures on opposing sides of the diaphragm. If the static and velocity-pressure pulsations are of the same frequency, they may be additive or subtractive depending on the phase angle, or they may not be synchronized at all. In either case a simultaneous differential reading would eliminate this chance of error. Although from general physics one would expect the velocity-pressure amplitude to be only slightly higher at the nozzle than at the pitot-tube station in the duct when the velocity of air is small compared with the velocity of sound, Figs. 6 and 7, for the reason given, do not provide data from which "the amplitude of the wave appears to be substantially the same."

The author infers that the big discrepancies between the two methods of test and reported in Table 1 are due solely to pulsation. Fan No. 13 shows 18.3 per cent error, which according to the curve of Fig. 4 corresponds to a value of $a/S = 1/8$. This would represent a condition where the maximum velocity is $17/9$ths and the minimum $1/9$th of the average velocity—a rather severe type of flow leaving a centrifugal fan.

Again, in referring to Table 1 the author states that "no specific conclusions can be drawn from these results as to the cause of pulsation." It would seem that this very fact would lead the author to look for some cause of vibration other than velocity pressure, which, as has been mentioned, may not have been correctly interpreted by the pulsometer used in the tests. Should not static pulsations be considered? Could not noise have complicated the results, particularly as the special pitot tube used had a large bore with short connections?

The author is correct in saying that the ordinary gage does not correctly interpret pulsating flow. The gage is actuated by pressure which is proportional to the square of the velocity. If there is pulsation, the average pressure as read by a gage will always be greater than that corresponding to the true mean velocity. If the frequency is low, the gage will probably show the fluctuation and the observer will estimate the average pressure. If the frequency is high, the gage will likely not fluctuate. The error is only great when the value of the amplitude S is large

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[5] "The Thermal Volume Meter," by G. W. Penny and C. J. Fechheimer, A.I.E.E. Quart. Trans., April, 1928, p. 537.

[6] "Ventilation of Revolving Field Salient Pole Alternators," by C. J. Fechheimer, A.I.E.E. Quar. Trans., April, 1930, p. 548.

[7] Research Engineer, Buffalo Forge Company, Buffalo, N. Y. Mem. A.S.M.E.

[8] "Effect of Pulsations on Flow of Gases," by Horace Judd and Donal B. Pheley, Trans. A.S.M.E., vol. 44, 1922, p. 870.

compared with a, the minimum pressure, as the author points out.

Fig. 9 shows a recommended test set-up. One cannot see how in a delicate test such as this the author can suggest a cone-throttling device which is sure to cause a high-velocity region to be set up in the large drum preceding the flow nozzle. A more uniform throttling device would seem to be more in order. As the set-up would be expensive for general approval tests and as these are becoming more and more common, any new code should simplify such procedure rather than complicate it.

To sum up, the author's conclusion 1, that a variation in velocity can exist, is valid. Conclusion 2, that the average velocity pressure is higher than the corresponding average velocity, is also true if a velocity pulsation does exist. Conclusions 3 and 4 are not substantiated by the data presented. The author may believe these conditions to exist, but the data presented are insufficient to cause any one to arrive at the same conclusions as the author. If the author has data not submitted which will show that velocity pulsation can be definitely measured and that it accounts for the discrepancy between pitot-tube traverse and nozzle tests, it should be included. The subject is of general interest and should be printed in more complete form so that it can be more intelligently criticized or approved. It does not seem that it would be a difficult matter to construct a pulsometer to be used in obtaining a correction factor for the standard pitot-tube traverse and save the tremendous expense of duplicate tests on commercial fans. At least, it should be further investigated before a radically different code is passed upon.

P. H. HARDIE.[9] This paper brings to the front again the discussion of the relative accuracy of pitot-tube and nozzle measurements of air quantities in connection with fan tests. The writer thinks that any one who is at all familiar with these two methods will concede that the nozzle measurement is more reliable and therefore preferable. The paper, however, shows the pitot tube in a more adverse light than is actually the case. If there was no explanation, other than pulsating flow, for differences in volume measurements as great as 16 or 18 per cent (shown in Table 1), the pitot tube would be considered so unreliable that it would be discarded. Due to the lower cost of the pitot-tube test, many engineers will consider carefully the relative accuracy of the two methods before eliminating the pitot tube as a test instrument.

The vibragraphs shown are interesting, but do not seem to prove the author's contention that large errors in pitot-tube volume measurements are due to pulsating flow. From measurements of the velocity pressure wave in the vibragraph, Fig. 6, the average value of a/S for the 87 cycles shown was found to be 6.3. This ratio gives an error due to pulsating flow of $^1/_2$ per cent instead of 7 per cent as reported for this test and shown in Fig. 8. The relative amplitudes of the pulsations in the other vibragraphs shown are even smaller.

In order to illustrate the extremely large amplitude of the velocity pressure wave necessary to produce an error of 7 per cent, graphs were made of a velocity sine wave with a ratio of $a/S = 0.8$ and of the resulting velocity-pressure wave. If the velocity pressure wave were sinusoidal, the error for the same amplitude would be about 6 per cent, and if it were triangular, the error would be only 4.5 per cent. The error for a triangular wave is expressed by the following equation:

$$\frac{\text{Per cent error}}{100} = \frac{3S'\sqrt{a' + S'}}{\sqrt{(a' + 2S')^3} - \sqrt{(a')^3}} - 1$$

The primed symbols refer to the dimensions of the velocity pressure wave corresponding to those for the velocity wave.

The author, while reporting the fact that the static pressure fluctuates as well as the impact pressure, apparently gives no consideration in this paper to the resulting higher measured static pressure. Since the velocity pressure is the difference between impact and static pressures, this higher static reading would at least tend to reduce the apparent error due to pulsating flow and might even cancel it. By "apparent error" the writer means the error computed from vibragraphs which give only the true wave shape for the impact pressure. The slightly higher static pressure itself would have but little effect upon the efficiency.

Poor distribution of the air flow at the pitot traverse station and duct velocities too low to be accurately measured are the causes of 95 per cent of the errors obtained with pitot tubes. It should be evident to those making a test that accurate results cannot be expected under such conditions, and therefore the difference obtained between the pitot and nozzle measurements is no criterion of their relative accuracy when so measured.

The writer wishes to disagree with the author in his contention that propeller fans cannot be accurately tested on the outlet end. A test was made on a two-blade propeller fan with a pitot tube and nozzles of three different sizes. Good agreement between the two methods was obtained, the nozzle measurement giving only slightly higher flows. A thorough investigation was made to determine if any peculiar conditions of flow existed, such as spin, that would cause erroneous results, but none was found. It was discovered, however, that the standard pitot tube used would read a slight positive static pressure in a stream of air 4 ft. from the outlet of a nozzle, and that the higher the velocity, the greater the error. This characteristic of the tube may account for the lower flows obtained with the pitot tube.

E. L. ANDERSON.[10] A review of investigations published covering the study of pulsation in air flow reveals a considerable collection of data upon reciprocating compressors. The published work on pulsation flow from low-pressure centrifugal fans rests in this paper, in which several of the phenomena remain unexplained. Application of Hodgson's formula of the sine wave in explaining several noted excess readings in pitot velocity is excellent, with the reservation that sound waves are complex, and, further, rarely are true sine forms, and that the "upstream cavity is larger than the downstream cavity, with the result that when a change of pressure occurs the pressure will rise and fall most quickly in the downstream cavity, thus exaggerating the readings" (Hodgson).

The basis for the major deductions responsible for the proposed code lies in the work performed upon reciprocating compressors. A further study was made by us upon a system of approximately the same size of ducts, and design analogous to the proposed code, substituting a centrifugal fan in place of the reciprocating compressors. Means were provided for controlling both amplitude and frequency of pulsations, and measurements of waves were made acoustically. Curves were plotted giving the pitot-tube pressure variations as related to frequency variations with constant total pressure at nozzle and giving the pitot variations as functions of amplitude at three different selected frequencies. The results fail in confirming results on centrifugal fans as duplicating those obtained in reciprocating compressors. The curves of the tests made indicate:

(*a*) That all pressures are affected by frequency as well as amplitude.

(*b*) Errors of static pressure readings due to pulsation.

(*c*) Relative effects of pressure variations with frequency and amplitude may be compensating or cumulative.

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(*d*) Error in all pressures may be either greater or less than smooth flow conditions.

(*e*) There is an indication of reflected system resonance.

Other curves plotted show the relative pressure readings at pitot and nozzle referred to a fixed nozzle velocity from a large-volume chamber which has been very heavily treated acoustically. The flow-nozzle static pressure smooth flow for upstream was 0.60 in. water gage and for downstream was 0.116 in. water gage.

Following the suggestion of reflected system resonance a test was run on a pipe of estimated period. Indications are of such a definite nature as to warrant further determinations regarding this condition.

The accuracy of a system of fan test is dependent upon three factors of pressure: total, static, and velocity. Providing the velocity pressure is accurately determined, it is essential that either the total or static be as accurately determined in order to compute the air-horsepower consumption of the fan, and, further, the relative position of the fan operation on a given system or systems. The proposed code is based upon the following:

1 That data compiled relating to reciprocating high-pressure, low-frequency pulsations may be applied to centrifugal low-pressure, high-frequency apparatus.

2 That the pulsation is substantially a sine wave.

3 That pitot-tube error is a function of wave amplitude only.

4 That static pressure be considered as an average and true reading due to throttle effect of pinhole orifices.

5 That pressure discrepancies are a function of wave amplitude only.

6 That pressure discrepancies are always in excess of pulsationless flow.

7 That nozzle flow may be corrected by applying the formula as given by Hodgson, and as further indicated by Andrae and Diedrich, for amplitude only.

8 That acoustic period of system has no effect on pulsation.

9 That the accuracy of nozzle measurements exist in Hodgson's suggestion of conversion of static to velocity pressure at nozzle, and throttling upstream from nozzle.

As related to the suggested physical design of proposed test system, the following is presented:

1 The pressure frequency amplitude of such data produces such dissimilar conditions as to preclude its application without thorough search for actual similitude.

2 Pulsation wave is not a true sine wave. Any correction factor based upon the true sine wave will be in error.

3 It is shown experimentally that pitot-tube error is caused not only by amplitude, but also by frequency, and that the effect of frequency may equal the amplitude error. The error may be either compensating or cumulative.

4 It is further shown experimentally that static pressure reading is rarely if ever accurate under pulsation flow, and that under some possible conditions, the error may transcend that of the impact tube.

5 All pressure discrepancies are a function of wave amplitude and frequency.

6 Pulsation error may be either above or below the pulsationless flow.

7 Nozzle flow is affected by amplitude and frequency.

8 Period of system may affect pulsation.

9 This assumption is confirmed as stated by Hodgson.

HENRY F. SCHMIDT.[11] The author has called attention to an important possible source of error in the testing of fans and blowers. Further study of the pulsations produced by such apparatus should be conducted, if possible, to determine their source.

The writer wonders, however, if for practical purposes the errors observed do not exist principally in the pitot tube and the usual method of determining static pressure. A number of years ago, in attempting to study the velocity distribution and determine the losses in a free vortex or whirl chamber of a blower where the velocities were of the order of 600 ft per sec, various forms of pitot tubes were employed, but the results obtained were so obviously in error that all forms of pitot tubes were abandoned and the readings were taken with an impact tube.

At this time the writer found that the Anaconda Copper Company had had a very extensive investigation made of pitot tubes in connection with determining the discharge from the big stack of the smelter at Great Falls, Mont. This investigation, which included the usual forms of pitot tube, also covered experimental work on static ends of the plate type, having holes of various sizes and distribution, as well as slots. This investigation showed that none of the static ends, either of the conventional pitot tubes or of the special forms investigated, indicates the true static pressure, all of them having errors which are a function of the velocity of the stream in which the tube is inserted.

More recently the writer had occasion to demonstrate the inconsistency of the static readings obtained with the standard pitot tube recommended by the American Society of Heating and Ventilating Engineers. By inserting the pitot tube in a jet issuing from a Moss nozzle with a quite smooth flow, it was observed that whereas the pressure in the jet must be atmospheric, the static end showed either a plus or a minus error, depending upon the velocity of the jet. This might be attributed to standing waves in the jet. This assumption, however, is precluded by the fact that with a constant jet velocity, the pitot tube was moved slowly along the axis of the jet and the error remained substantially constant, decreasing very slightly with increased distance from the nozzle.

A Prandtl tube was tested in the same jet at the same velocities and showed a fairly uniform error independent of the velocity and of the position along the axis of the jet. For this reason the Prandtl tube is superior to the A.S.H.V.E. tube, even though it does not give the correct static pressure. Because of these experiences, the sensitiveness to disturbances in flow and alignment with the stream lines, and the discrepancies between nozzle and pitot-tube measurements, which are mentioned in his paper, the writer believes that the pitot tube should be discarded as a test instrument for the measurement of volume or static pressure.

Since 1909 the writer has employed only nozzles for determining volume, measurements being made either at the entering or discharge side of the blower, and in some instances both at the inlet and discharge. In such cases, the writer has not observed differences exceeding 1 per cent in the two measurements. In making nozzle measurements, the writer has always employed a piezometer ring having a large number of small holes, as this undoubtedly gives more reliable results than a single opening in the side of the duct. If for some reason it is impossible to use a piezometer ring, at least four openings in one plane of the duct should be connected to a single manometer.

While the nozzle developed and recommended by Dr. Moss is unquestionably the ideal form for small nozzles used in testing high-pressure blowers or compressors, this form of nozzle is prohibitive because of cost for the testing of large blowers.

While the writer agrees with the author that pulsations do exist and may in some cases be the cause of serious error, based on broad experience with both pumps and blowers of exactly similar design and the use of piezometer rings for measuring pressures, the writer wonders whether the discrepancies do not possibly exist almost entirely in the pitot-tube method of measurement.

In regard to the suggested method of using steam coils for

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heating air as a check method of determining volume, the writer would expect far more serious errors by this method than in the case of a pitot tube, as it would be necessary to accurately determine the quality of steam entering the coil in order to be able to calculate the heat given up by the condensate, which is weighed. The heating method is open to many objections, and there are many possible sources of error such as radiation, accuracy of measurement of the heat input, and the correct measurement of the temperature rise of the air. If any heating method of determining the volume is to be employed, the writer believes it preferable to employ electric heating elements, which should be in the form of copper strips properly braced to prevent vibration, and the thermocouple elements must also be so supported to prevent vibration, since it has been found in the laboratory that serious errors in the temperature measurements are observed if the wires forming the thermocouples vibrate in the air stream. While possibly such measurements can be made with a high degree of accuracy under laboratory conditions by people skilled in this method, it has many pitfalls for the casual user.

The author has stated that, contrary to expectations, six-bladed propellers do not show less pulsation than those with two blades, thus indicating that the decrease of efficiency with increasing number of blades is not attributable to pulsations. Further evidence to this effect is shown by the writer's tests on $2^3/_8$-in. propellers pumping water and on 12-in. propellers in air. While the efficiencies are not the same, the percentage differences in efficiency for each additional blade are nearly alike.

The writer attributes the decrease of efficiency to the increased number of edges and the fact that the friction on a surface decreases with increased distance from the leading edge. Thus the surface friction on two surfaces each 2 in. wide is greater than that of one surface 4 in. wide. According to the hydrodynamical theory for flow in an ideal fluid, an entering edge causes no resistance, but the writer does not believe this is true for air or liquids with which we have to deal. For this reason, the writer has termed the observed efficiency differences "the edge loss," whatever that may actually mean.

Though it may not be possible to design blowers which do not produce pressure pulsations of appreciable magnitude, such pulsations as do exist can be damped almost completely by the use of a duct constructed of perforated plate surrounded by a space of 2 or 3 in. radial depth filled with cotton waste, enclosed in a non-perforated shell similar to the "Burgess" muffler, the perforated section of duct being 10 diam or more long, depending upon the amplitude of the disturbances which exist at the blower outlet. Some research work along this line should be undertaken to fix some rules for determining the length of muffler required to damp pulsations to a negligible value. Such mufflers are also quite effective in reducing noise.

Possibly such a dampening means should be employed in testing to avoid any effect of pulsations on the readings.

J. M. Mayoral.[12] The author presents an interesting explanation of the inconsistencies sometimes experienced in the testing of centrifugal fans. His use of the pressure-fluctuation indicator opens up a method of investigation that may eventually furnish a solution of the problem. The paper shows that in measuring the output of a centrifugal fan much more reliable results can be obtained by using a nozzle exhausting to the atmosphere at a relatively high velocity than by using a pitot-tube traverse in the duct. It also shows that if the results of the pitot-tube traverse and the nozzle determination of volume agree, it is an indication that any velocity pulsations in the flow are negligible.

The vibragraphs demonstrating that the wave form of the impact pressure as measured at the pitot-tube station is similar in shape and amplitude to that measured at the mouth of the nozzle are interesting. In order to get a reliable indication of what is happening in the duct, however, it will not be sufficient to have vibragraphs of the impact pressure only. Study must be made of the variation of the static as well as the impact pressure in order to determine the form of the velocity-pressure wave. The author admits that the static pressure in the duct is variable and that the wavy static pressure line in Fig. 6 does not show the true wave form.

C. E. Peck.[13] This paper has material of especial significance for any one concerned with methods of accurately determining fan performance. In all probability, the presence of pulsation as a factor which seriously affects fan-test procedure has been neglected to a great extent in the past and explains in many cases why investigators may not independently check the performance of the same fan when two different methods of measuring volume are used. It also is of special importance to note that the standard test code in its present form is not complete enough to provide for elimination of excessive errors in fan testing when the pulsations produced by the fan are of sufficient magnitude to make the results of the pitot-tube method difficult to interpret correctly. As mentioned in the paper, the presence or absence of errors due to pulsation can at the present time be best determined by making two volume determinations—one by the usual pitot-tube traverse method and one by a nozzle determination at a higher velocity, close agreement implying that only small errors are present. This method uses the nozzle as the standard of reference. In some extreme cases, the nozzle result may give considerable error due to pulsation. The most desirable solution would be the development of a simple, compact indicating instrument which could be placed in the duct and would measure the amplitude of the pulsation. An approximate knowledge of the amplitude and the average velocity would be enough to determine the error due to pulsation, as in Fig. 4 of the paper. This would simplify and make accurate the testing of the fan in systems outside of the laboratory and the well-equipped test floor. The advantage of the pitot-tube traverse method is the ease with which it can be adapted to all types and sizes of fan testing set-ups, and at the same time provide a means of determining velocity and static pressure simultaneously. This advantage is lost when flow nozzles are used because of the large number needed to cover the size and volume range. Since in any fan test the pitot-tube readings cannot be interpreted correctly until the magnitude of the error due to pulsation is determined, it is important that a simple auxiliary instrument be devised to measure pulsation amplitudes if the pitot-tube method is to survive in the standard method of testing fans.

Another more fundamental approach to the problem of pulsation magnitude, but probably much more difficult to execute, would be to make a study of the amplitude and energy in the pulsating wave as a function of the fan design (type, speed, capacity, size, etc.) and its accompanying duct work. Some general conclusions can be arrived at as more fans are checked experimentally for pulsation errors. The observation noted by the author, that the pulsation error is greater with larger outputs, has been checked recently by some tests at the Westinghouse Elec. & Mfg. Co. in connection with the ventilation of railway motors.

A single-inlet centrifugal-type blower was used to supply air through a 2-ft-diam duct to a railway motor, the duct being about 70 ft in length. Volumes were measured simultaneously by means of a thermal volume meter (which measures mass rate of flow) and two types of pitot tube—the standard type used in the

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fan test code and the spherical-head Prandtl type. For those types of motors requiring low static pressures (2 to 4 in. of water) for about 10,000 cu ft per min volume, the two methods of volume measurement checked consistently within $1^1/_2$ per cent. The two different type pitot tubes checked within 1 per cent. For these tests, the thermal volume meter was located about 22 duct diameters from the fan and the pitot-tube station about 26 diameters from the fan. For the same locations and the same volume, but for higher static pressures (7 to 9 in. of water) obtained by increasing the speed of the blower, both pitot tubes checked each other, but differed from the thermal volume-meter measurement by giving 10 per cent higher volume. For the same volume and pressure, another pitot-tube traverse taken $7^1/_2$ duct diameters from the fan (standard distance) gave a volume 13 per cent higher than at the station 26 diam from the fan, or 23 per cent higher than that obtained by the thermal volume meter. Although these tests are not as yet complete, some tentative conclusions are:

(1) The variation in pitot-tube readings compared to the thermal volume meter from the lower to the higher static pressures indicates that the pulsation has become a serious factor at the higher pressures.

(2) The upward variation of the pitot-tube reading at the two stations indicates higher pulsation errors at a point nearer the blower. An exceedingly long duct would be needed to eliminate pulsation errors. In these tests, assuming the thermal volume meter as a standard of reference, even 26 diam was not enough to eliminate excessive pulsation errors.

(3) The thermal volume meter does not seem sensitive to the pulsation magnitudes which affect the pitot tube. More tests will have to be made before this conclusion is entirely justified. The results obtained by placing the volume meter closer to the blower ($7^1/_2$ diam) have not yet been made at the time this is being written. It seems reasonable to expect that the volume-meter readings (energy input to heater and air temperature rise) would be little affected by pulsations of the order encountered in fan work. The temperature rise of an air particle going through the heater coils varies approximately inversely as the velocity to the first power; hence, the average temperature rise of the particle is the same whether the air is pulsating or not. Only in cases where the pulsation would be of enough magnitude and vibrate in such a manner as to produce non-uniform distribution and "back flow" would the volume meter accuracy be affected. Further experimental evidence as to the accuracy of the volume meter under pulsating conditions must be obtained before definite conclusions as to its correctness can be made.

Figs. 6 and 7 of the paper bring up several questions. Is not the measurement of static pressure by means of the static holes in the pitot tube subject to errors of the same order as those encountered in the measurement of the total pressure (velocity pressure plus static pressure) when such measurements are made within a duct? Fig. 6 shows that a vibration of appreciable magnitude exists at the static pressure holes. This probably gives higher average static pressure readings than the actual average static pressure. Since the velocity head is not measured directly in this case, but can only be determined by the difference between the total pressure and the static pressure, the error in the average velocity head is affected by errors in measurement of the two components by which it is determined. It seems to the writer that a high value of a/S is not the only criterion of correctness. The flow-nozzle method, as described in Fig. 9, enables a pitot determination at the exit of the nozzle to be made, at which the static pressure is zero, and in which case the impact pressure corresponds to the velocity head directly; hence, no error is introduced due to possible inaccuracies arising from measurements of static pressure. It seems reasonable to assume that for this reason, as well as for a higher value of a/S, the flow-nozzle determinations of volumes are more accurate and consistent when the flow is pulsating. The question also comes up as to how the static pressure is to be determined accurately under pulsating flow conditions if it is necessary to determine the volume independently by means of a flow nozzle.

Some evidence as to whether the measurement of static pressure is a contributing cause to the different results obtained with the flow nozzle versus pitot-tube traverse inside a duct could be obtained as follows:

A pitot-tube traverse could be obtained inside a duct containing pulsating flow with appreciable values of a/S and static pressure. A similar traverse could be obtained at the exit of the same duct where the static pressure is zero, thus maintaining a constant value of a/S, but varying the static pressure for the same average velocity-pressure determination. If the average velocity pressure at the duct exit checked a nozzle determination more closely than that obtained inside the duct, then part of the error is due to static-pressure measurements.

If such results were found, then the standard test code would have to be revised still further, and the testing rig made such that static pressures could be measured more accurately. This would probably mean larger inlet or outlet boxes adjacent to the fan constructed in such a way as to damp out pulsations at the point where static pressures are to be measured.

Author's Closure

Several of the discussions mention a possible difficulty in measuring static pressure with a pulsating flow. Detailed comment on such a possibility would be highly controversial and at the present time fruitless and indeterminate.

The use of the free nozzle and impact tube to determine fan volume, which for accuracy seems to meet with general approval, permits basing of test results on the readings of impact tubes only. Another impact tube is used in the fan duct to measure total pressure. The fan static, if wanted, is obtained by calculation from the nozzle volume and the fan-outlet area.

The static tube is used only for the differential velocity pressure in the fan duct, and the resulting volume as calculated provides a check on the nozzle volume. In the event of large differences in the two volumes, it is recognized that even the nozzle volume may be too high, and certain methods of approximation are suggested to approach a more nearly correct result. In this method of test the function of the static tube is solely to indicate the presence of pulsation. Apparently none of the commentators objects to its use for that purpose.

It is well known that flow pulsation increases the loss in head due to friction. The amount of this increase is directly proportional to the increase of the indicated velocity pressure above the velocity pressure for uniform flow. It is important, therefore, to know whether pulsation is present, particularly if pulsation can persist through long lengths of duct.

Mr. Peck's contribution contains new and valuable information showing that serious pulsation has carried for at least 26 diam of duct length.

The confirmation from their own work, some of it apparently antedating that of the author, by Dr. Moss and Messrs. Fechheimer and Peck is valuable and necessary.

In Mr. Madison's opinion the author's presentation is incomplete. The paper proceeds from inference based on known facts to a rational hypothesis which test results confirm with reasonable quantitative exactness. All pertinent data are given. The theory is explained, and a suggested method for avoiding the test error is presented, and with sufficient completeness to enable any one familiar with the art to undertake a similar investigation. Such procedure seems to accord with the accepted scientific method. Fig. 6 was taken at the largest nozzle-

volume point of Fig. 8. A speed conversion, with which Mr. Madison is unquestionably familiar, to comply with certain of our laboratory rules of standardization of curves was introduced in the curve.

Mr. Madison's opinion that measurement of a fluctuating velocity pressure is a simple matter must be answered at length to avoid the misleading of other experimenters. Any one interested should read report No. 320 of the National Advisory Committee for Aeronautics, which describes experiments leading to a successful quantitative diagram of a pulsating static. The work done by Professor Judd and Mr. Pheley was at the low frequencies of reciprocating compressors. The ingenious instrument they used is not applicable to the high frequencies encountered in a fan discharge.

The symmetrical throttling cone, the distance to the nozzle, and the diameter of the nozzle mouth are all in accordance with accepted practice for measurement with free nozzles. The results have been checked by nozzles directly on the end of 10 diam of straight duct.

Mr. Madison and Mr. Hardie both find difficulty in attempting to conceive of a kind of flow that would account for the large errors found by comparative tests. The author believes that the evidence of the quantitative results of Table 1 in the paper is the best answer to their opinions.

Mr. Hardie's quantitative analysis of Fig. 6 is due to a misunderstanding of the diagram. The paper clearly states that the static line is an average, and so there are no instantaneous velocities to be evaluated. The author disclaims any quantitative value for even the impact line. The air column leading to the instrument will decrease materially the recorded wave amplitude. Considered in this light, the diagrams seem not unconvincing.

The suggestion made by both Mr. Hardie and Mr. Schmidt that inaccuracies of the standard pitot tube are responsible for the errors is controverted by the results reported by Mr. Peck, whose findings are in accordance with those of other experimenters.

Mr. Anderson apparently considers precautions necessary to avoid pulsation error, but is concerned also about static-pressure difficulties, which were dealt with in the opening paragraph of this closure. The remainder of his discussion deals chiefly with certain suggested methods of approximation to be used when severe pulsation is indicated and which are more applicable to the proposed new code in which these suggestions appear than to the paper itself.

Referring again to Mr. Peck's discussion, the author hopes that Mr. Peck will continue his investigation. A 70-ft duct, 35 diam, will enable him to search out various matters that are not apparent in the usual length of test duct. The author wishes further that Mr. Peck might find it possible to use a flow nozzle at the end of the duct as an additional measurement. This volume would be of great help in deciding on the reliability of the electrical method and might further, in combination with the pitot tube and the electrical method, give experimental information as to nozzle-volume approximations when extreme pulsation is indicated.

FSP-55-8

Performance of Modern Steam-Generating Units

By C. F. HIRSHFELD[1] and G. U. MORAN,[2] DETROIT, MICH.



C. F. Hirshfeld

This paper is the final report on a study made of the performance of 250 large steam boilers over a period of three years. The results of the first year's study were reported at the 1929 Annual Meeting of the A.S.M.E., and a summary of the full three-year period was presented at the 1931 Annual Meeting. The study was requested by the Central Station Power Committee of the Power Division. The primary object was to study boiler outages and their causes and the effect of such factors as severity of use, design, fuels, and methods of firing on boiler reliability, in the hope that a yardstick might be obtained by which the individual operator could measure the performance of his own units and predict the probable availability of new designs. The data obtained from each unit are presented, together with some comparisons of types and classes of unit. There also is a discussion of primary causes of boiler shutdown, which has not appeared in either of the previous papers.



G. U. Moran

STEAM-GENERATING EQUIPMENT has received its share of attention in technical literature and discussion, with respect to most of the factors affecting its design and operation. The quite considerable increase in thermal economy, which has been attained during the past two decades, has been widely discussed. The former "boiler" has evolved into that complicated structure we now know as the steam-generating unit, but the development was by no means made blindly. The effect on thermal efficiency of each addition to and refinement of the "boiler" has been determined to such a degree that the designer may predict with confidence the thermal performance of his creation. This knowledge has made it possible to weigh investment cost against probable savings and to define limits beyond which an increase in efficiency would be overbalanced by a disproportionately large increase in initial cost.

But low investment is not of itself a complete criterion on the dollar side of the picture. A large investment which can be used for, say, 80 per cent of each complete year of the life of a plant may represent a better economic solution than a smaller investment made to accomplish the same result, but usable only during, say, 50 per cent of the entire year. In other words, the extent to which equipment can be kept in as continuous service as existing conditions indicate to be ideally desirable is also an important measure.

Thermal efficiency is comparatively easy of proof. Investment made is merely a question of fact. The performance factor, however, is a question of experience, and any one individual's intimate experience must necessarily be limited to a comparatively small number of installations. It results that while there is available a wealth of information concerning thermal efficiencies and first costs of steam-generating plants, there is very little information about the extent to which various features of design influence the ability of those plants to produce when and as required.

This fact was appreciated by the Central Stations Committee of the Power Division, A.S.M.E., when it requested the authors to make this investigation and to provide if possible a yardstick of normal values by which the individual operator might determine whether his performance was good, bad, or normal.

The study was started in July, 1928, with the collection of daily operating data from 186 boiler units located in 50 plants well scattered throughout the United States. The analysis of one year's records, which was presented as a paper[3] before the Power Division at the Annual Meeting, December, 1929, showed that, if significant results were to be obtained, the study should be continued over a longer period and a larger number of units should be included. The present paper therefore represents an investigation of 250 units in 61 stations, covering the three-year period between July 1, 1928, and July 1, 1931. A progress report was made at the Annual Meeting in December, 1931. It contained a general summary of the data at hand, but due to lack of sufficient time for preparation, the more detailed report was postponed and constitutes this, the final paper.

The procedure followed in obtaining the data has been set forth in the 1929 paper, together with definitions used. To conserve space the method will not be repeated here in detail. Suffice it to say that the data were collected by means of printed log sheets which were filled out by the operators themselves and

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] "The Performance of Modern Steam-Generating Units," Hirshfeld and Moran, Trans. A.S.M.E. (1930), paper FSP-52-34, p. 265.

TABLE 1 DESCRIPTIVE DATA AND DESIGN FACTORS

Unit No.	Steam pressure, lb per sq in. gage	Steam temperature, F	Heating surface in sq ft: Boiler	Superheater	Reheater	Water-cooled walls	Hearth screen	Air-cooled walls	Economizer	Air preheater	Total saturated heating surface	Volume of combustion chamber, cu ft	Projected grate area, sq ft	Ratio economizer to saturated surface	Ratio furnace volume to saturated surface	Ratio grate area to saturated surface	Heat value of fuel, Btu per lb	Method of firing[a]	Type of dust collector[b]	Combustion control[c]	Unit No.
Stoker-Fired Units, 500 to 700 Lb per Sq In. Gage Pressure																					
S-1	625	725	14086	2009	...	...	..	..	8503	9544	14086	4395	365	0.60	0.37	0.026	10460	TG	...	MI	S-1
S-2	600	725	21551	3200	...	1597	..	..	15772	51000	23148	8256	528	0.68	0.36	0.023	11240	TG	...	MI	S-2
S-3	625	735	14086	2225	...	..	..	..	8503	9544	14086	4970	356	0.60	0.37	0.026	10460	TG	...	MI	S-3
S-4	625	725	14086	2257	...	..	..	..	8503	9544	14086	4970	356	0.60	0.37	0.026	10460	TG	...	MI	S-4
S-5	625	725	14086	2171	...	..	..	..	8503	9544	14086	4970	356	0.60	0.37	0.026	10460	TG	...	MI	S-5
S-6	600	725	21551	3200	...	1597	..	..	15772	51000	23148	8256	528	0.68	0.36	0.026	11240	TG	...	MI	S-6
S-7	600	725	21551	3200	...	1597	..	..	15772	51000	23148	8256	528	0.68	0.36	0.023	11240	TG	...	MI	S-7
S-8	600	725	21551	3200	...	1597	..	..	15772	51000	23148	8256	528	0.68	0.36	0.023	11240	TG	...	MI	S-8
S-9	625	725	14086	2255	...	..	..	..	8503	9544	14086	4395	365	0.60	0.37	0.026	10460	TG	...	MI	S-9
S-10	625	725	14086	2170	...	..	..	..	8503	9544	14086	4395	365	0.60	0.37	0.026	10460	TG	...	MI	S-10
S-11	575	750	16615	2872	...	..	..	..	10250	9948	16615	6000	444	0 60	0 36	0 027	10215	TG	...	MI	S-11
S-12	650	725	14086	2485	...	...	..	..	8837	...	14086	5760	356	0.63	0.41	0.026	11240	TG	...	MI	S-12
S-13	575	750	17280	3850	...	1253	..	..	7376	15913	18533	5750	444	0.86	0.31	0.025	10215	TG	...	MI	S-13
S-14	650	725	14086	2485	...	..	..	..	8837	...	14086	5760	356	0.63	0.41	0.026	11240	TG	...	MI	S-14
S-15	675	725	17962	2800	...	..	..	..	3011	81000	17962	7790	528	0.17	0.44	0.030	10834	TG	CT	MI	S-15
S-16	675	725	17962	2800	...	..	..	..	3011	81000	17962	8500	528	0.17	0.44	0.030	10834	TG	CT	MI	S-16
S-17	650	725	14086	2485	...	..	..	..	8837	...	14086	5760	356	0.63	0.41	0.026	11240	TG	...	MI	S-17
S-18	650	725	14086	2485	...	..	..	..	8837	12000	14086	5760	356	0.63	0.41	0.026	11240	TG	...	MI	S-18
S-19	675	725	17962	2800	...	..	..	..	3011	81000	17962	7790	528	0.17	0.44	0.030	10834	TG	CT	MI	S-19
S-20	650	725	14086	2485	...	..	..	..	8837	...	14086	5760	356	0.63	0 41	0 026	11240	TG	...	MI	S-20
S-21	650	725	14086	2485	...	..	..	..	8837	...	14086	5760	356	0.63	0.41	0.026	11240	TG	...	MI	S-21
S-22	675	725	17962	2800	...	..	..	..	3011	81000	17962	7790	528	0.17	0.44	0.030	10834	TG	CT	MI	S-22
S-23	675	725	17962	2800	...	..	..	..	3011	81000	17962	7790	528	0.17	0.44	0.030	10834	TG	CT	MI	S-23
S-24	675	725	17962	2800	...	..	..	..	3011	81000	17962	7790	528	0.17	0.44	0.030	10834	TG	CT	MI	S-24
Stoker-Fired Units, 400 to 499 Lb per Sq In. Gage Pressure																					
S-25	420	700	10438	2925	...	...	..	..	6480	...	10438	5400	270	0.62	0.52	0.026	11570	Und	...	MI	S-25
S-26	400	707	15697	2822	...	597	..	..	7515	22072	16294	7780	337	0.46	0.48	0.020	13841	Und	...	SA	S-26
S-27	420	700	10438	2925	...	..	..	..	6480	...	10438	5400	270	0 62	0 52	0.026	11570	Und	...	MI	S-27
S-28	420	700	10438	2925	...	..	..	..	6480	...	10438	5400	270	0.62	0.52	0.026	11570	Und	...	MI	S-28
S-29	420	700	10438	2925	...	...	..	..	6480	...	10438	5400	270	0.62	0.52	0.026	11570	Und	...	MI	S-29
S-30	400	700	23764	3037	...	1322	..	..	14970	...	25086	8900	524	0.60	0.35	0.020	14054	Und	Dry	Ce	S-30
S-31	400	700	23764	3037	...	1322	..	..	14970	...	25086	8900	524	0.60	0.35	0.021	14054	Und	Dry	Ce	S-31
S-32	400	720	22900	4350	...	5336	..	..	...	90000	28236	8530	500	..	0.30	0.020	14098	Und	...	MI	S-32
S-33	400	700	23764	3037	...	1322	..	..	14970	...	25086	8900	524	0.60	0.35	0.021	14054	Und	Dry	Ce	S-33
S-34	400	700	23764	3037	...	1322	..	..	14970	...	25086	8900	524	0.60	0.35	0.021	14054	Und	Dry	Ce	S-34
S-35	400	720	22900	4350	...	5336	..	..	...	90000	28236	8530	500	..	0.30	0.020	14098	Und	...	MI	S-35
S-36	400	720	22900	4350	...	5336	..	..	...	90000	28236	8530	500	..	0.30	0.020	14098	Und	...	MI	S-36
Stoker-Fired Units, 300 to 399 Lb per Sq In. Gage Pressure																					
S-37	325	725	19069	4920	...	419	..	..	18018	...	19488	10400	546	0.93	0.53	0.028	11286	TG	...	MI	S-37
S-38	310	600	13850	2735	...	1060	..	..	...	...	14910	6900	280	..	0.46	0.019	13890	Und	...	Au	S-38
S-39	350	700	16396	3625	...	..	..	..	10703	...	16396	6736	257	0.67	0.41	0.016	13038	Und	...	Au	S-39
S-40	350	700	16396	3625	...	..	..	..	10703	...	16396	6736	258	0.65	0.41	0.016	13080	Und	...	Au	S-40
S-41	350	700	16396	3625	...	..	..	..	10703	...	16396	6736	257	0.65	0.41	0.016	13080	Und	...	Au	S-41
S-42	350	700	16396	3625	...	..	..	..	10703	...	16396	6736	257	0.65	0.41	0.016	13080	Und	...	Au	S-42
S-43	350	700	16396	3625	...	..	..	..	10703	...	16396	6736	257	0.65	0 41	0.016	13080	Und	...	Au	S-43
S-44	350	700	16396	3625	...	...	..	..	10703	...	16396	6736	257	0.65	0.41	0.016	13080	Und	...	Au	S-44
S-45	350	680	16396	3625	...	250	..	..	10703	...	16646	6736	257	0.66	0.40	0.015	13080	Und	...	Au	S-45
S-46	350	630	14579	4220	...	...	..	..	7728	...	14579	4050	370	0.53	0.28	0.026	14199	Und	...	MI	S-46
S-47	385	692	10636	5000	...	575	..	..	11340	...	11211	5200	324	1.00	0.47	0.029	13128	Und	...	MI	S-47
S-48	385	692	10636	5000	...	575	..	..	11340	...	11211	5200	324	1.00	0.47	0.029	13128	Und	...	MI	S-48
S-49	385	700	12340	5704	...	749	..	..	4800	56000	13089	5500	385	0.37	0.42	0.030	13128	Und	...	MI	S-49
S-50	360	725	11959	4500	...	850	..	..	5880	...	12809	6850	361	0.46	0.53	0.028	11225	Und	...	Au	S-50
S-51	360	725	11959	4500	...	850	..	..	5880	...	12809	6850	361	0.46	0.54	0.028	11225	Und	...	Au	S-51
S-52	360	725	11959	4500	...	850	..	..	5880	...	12809	6850	361	0.46	0.53	0.028	11225	Und	...	Au	S-52
S-53	325	650	15089	4052	...	..	..	..	12250	8954	15089	4800	410	0.81	0.32	0.027	10395	TG	...	MI	S-53
S-54	385	700	12337	5527	...	760	..	..	4800	56000	13097	5500	385	0.37	0.42	0.030	13128	Und	...	MI	S-54
S-55	325	700	13828	..	...	...	..	..	8400	...	13828	5400	182	0.61	0.39	0.013	11286	TG	...	MI	S-55
S-56	385	700	12340	5704	...	749	..	..	4800	56000	13089	5500	385	0.37	0.42	0.030	13128	Und	...	MI	S-56
S-57	375	700	19743	2938	...	..	..	190	11091	...	19743	9000	374	0.56	0.46	0.019	13965	Und	CC	Au	S-57
S-58	325	700	13828	..	...	..	..	..	8400	...	13828	5400	182	0.61	0.39	0.013	11286	TG	...	MI	S-58
S-59	385	720	23600	4130	...	..	..	..	...	...	23600	8960	394	..	0.38	0.017	14098	Und	...	MI	S-59
S-60	385	720	23600	4130	...	..	..	..	...	...	23600	9510	345	..	0.40	0.015	14098	Und	...	MI	S-60
S-61	325	700	13828	..	...	..	..	..	8400	...	13828	4160	182	0.61	0.30	0.013	11286	TG	...	MI	S-61
S-62	325	700	18933	5620	...	..	..	..	11464	...	18933	8400	494	0.60	0.45	0.026	11286	TG	...	MI	S-62
S-63	385	720	23600	4130	...	..	..	..	...	...	23600	9510	345	..	0.40	0.015	14098	Und	...	MI	S-63
S-64	350	630	12322	3174	...	..	..	..	7728	...	12322	3800	240	0.63	0.31	0.020	14199	Und	...	MI	S-64
S-65	385	720	23600	4130	...	..	..	..	...	...	23600	9510	345	..	0.40	0.015	14098	Und	...	MI	S-65

Stoker-Fired Units, 300 to 399 Lb per Sq In. Gage Pressure (Continued)

S-66	385	720	23600	4130	...	..	..	..	...	...	23600	8960	394	..	0.38	0.017	14098	Und	...	Ml	S-66
S-67	385	720	23600	4130	...	..	..	..	...	...	23600	8960	394	..	0.38	0.017	14098	Und	...	Ml	S-67
S-68	385	720	23600	4130	...	..	..	..	...	...	23600	8960	394	..	0.38	0.017	14098	Und	...	Ml	S-68
S-69	385	720	23600	4130	...	..	..	..	...	...	23600	9510	345	..	0.40	0.015	14098	Und	...	Ml	S-69
S-70	385	720	23600	4130	...	..	..	..	...	...	23600	9510	345	..	0.40	0.015	14098	Und	...	Ml	S-70
S-71	385	720	23600	4130	...	..	..	..	...	...	23600	8960	394	..	0.38	0.017	14098	Und	...	Ml	S-71
S-72	300	700	18772	4238	...	..	..	..	...	...	18772	5318	258	..	0.28	0.014	13968	Und	...	Ml	S-72
S-73	350	650	17257	2424	...	..	..	..	...	...	17257	4014	202	..	0.23	0.012	10939	TG	...	Au	S-73
S-74	310	650	13850	2735	...	1060	..	..	...	...	14910	6900	280	..	0.46	0.019	13890	Und	...	Au	S-74
S-75	350	700	11959	2010	...	..	..	..	3245	...	11959	6358	272	0.27	0.53	0.023	11225	Und	...	Au	S-75
S-76	385	720	23600	4130	...	..	..	..	...	...	23600	9510	345	..	0.40	0.015	14098	Und	...	Ml	S-76
S-77	384	720	23600	4130	...	..	..	..	...	...	23600	8960	394	..	0.38	0.017	14098	Und	...	Ml	S-77
S-78	350	700	11959	2010	...	..	..	..	3245	...	11959	6358	304	0.27	0.53	0.025	11225	Und	...	Au	S-78
S-79	310	650	14092	1900	...	907	..	..	...	...	14999	6900	280	..	0.46	0.019	13890	Und	CT	Au	S-79
S-80	375	700	17257	1877	...	..	..	..	...	...	17257	4687	483	..	0.27	0.028	10541	TG	...	Au	S-80
S-81	375	700	17257	1877	...	..	..	..	...	...	17257	4687	483	..	0 27	0 028	10541	TG	...	Au	S-81
S-82	350	700	11959	2010	...	..	..	..	3245	...	11959	6358	272	0.27	0.53	0.023	11225	Und	...	Au	S-82
S-83	300	650	10084	..	...	..	..	..	...	...	10084	2886	179	..	0.29	0.018	12048	Und	...	Ml	S-83
S-84	300	700	18772	4238	...	..	..	..	...	...	18772	5318	258	..	0.28	0.014	13968	Und	...	Ml	S-84
S-85	300	700	18772	4238	...	..	..	..	...	...	18772	5318	258	..	0.28	0.014	13968	Und	...	Ml	S-85
S-86	300	650	10084	..	...	..	..	..	...	...	10084	2886	179	..	0 29	0.018	12048	Und	...	Ml	S-86
S-87	300	650	10084	..	...	..	..	..	...	...	10084	2886	179	..	0.29	0.018	12048	Und	...	Ml	S-87
S-88	300	650	10084	..	...	..	..	..	...	...	10084	2886	179	..	0.29	0.018	12048	Und	...	Ml	S-88
S-89	300	700	18772	4238	...	..	..	..	...	...	18772	5100	257	..	0.27	0.014	13968	Und	...	Ml	S-89

Stoker-Fired Units, 200 to 299 Lb per Sq In. Gage Pressure

S-90	275	750	12515	3815	...	..	..	..	9435	...	12515	4250	...	0.75	0.34	...	12376	Und	...	Ml	S-90
S-91	275	750	12515	3815	...	..	..	..	9435	...	12515	4250	...	0.75	0.34	...	12376	Und	...	Ml	S-91
S-92	275	750	12515	3815	...	..	..	..	9435	...	12515	4250	...	0.75	0.34	...	12376	Und	...	Ml	S-92
S-93	275	750	12515	3815	...	..	..	..	9435	...	12515	4250	...	0.75	0.34	...	12376	Und	...	Ml	S-93
S-94	275	650	11120	1999	...	210	..	..	6674	3600	11330	4900	309	0.58	0.43	0.027	13475	Und	...	Au	S-94
S-95	275	650	11120	1999	...	210	..	..	6674	3600	11330	4900	309	0.58	0.43	0.027	13475	Und	...	Au	S-95
S-96	275	750	12515	3815	...	..	..	..	9435	...	12515	4250	...	0.75	0.34	...	12376	Und	...	Ml	S-96
S-97	290	650	12743	5748	...	..	..	..	8100	9600	12743	4738	161	0 64	0.37	0.013	14869	TG	...	Ml	S-97
S-98	275	600	13840	3863	...	..	..	..	8256	...	13840	4150	284	0.60	0.30	0.020	13029	Und	...	Ml	S-98
S-99	290	650	12705	2475	...	..	..	..	9022	8633	12705	4782	168	0.73	0.38	0.013	14869	TG	...	Ml	S-99
S-100	275	750	12100	4000	...	..	..	..	3145	...	12100	4250	...	0.26	0.36	...	12376	Und	...	Ml	S-100
S-101	275	650	11120	1999	...	210	..	..	6674	3600	11330	4900	309	0.58	0.43	0.027	13475	Und	...	Au +	S-101
S-102	275	650	11120	1999	...	210	..	..	6674	3600	11330	4900	309	0.58	0.43	0 027	13475	Und	...	Au	S-102
S-103	235	420	10100	1450	...	222	..	..	5820	37200	10322	5412	221	0.56	0.52	0.020	13880	Und	...	Au	S-103
S-104	270	640	12100	4000	...	460	..	37	10692	...	12560	6500	378	0.85	0.52	0.030	14177	Und	CC	Au	S-104
S-105	235	420	10100	1450	...	222	..	..	5820	37200	10322	5412	221	0.56	0.52	0.020	13880	Und	...	Au	S-105
S-106	225	625	11400	1485	...	485	..	160	...	...	11885	3710	212	..	0.31	0.018	14199	Und	CF	Au	S-106
S-107	265	600	14217	2582	...	488	..	..	...	50276	14705	6650	310	..	0.45	0.021	13727	Und	...	Au	S-107
S-108	210	617	14403	3511	...	..	..	..	9218	...	14403	3024	332	0.64	0.21	0.023	9863	TG	...	Ml	S-108
S-109	225	625	11400	1485	...	485	..	160	...	...	11885	3710	212	..	0.31	0.018	14199	Und	CF	Au	S-109
S-110	225	625	11400	1485	...	485	..	160	...	...	11885	3710	212	..	0.31	0.018	14199	Und	CF	Au	S-110
S-111	265	600	14217	2582	...	488	..	..	...	42000	14705	6650	310	..	0.45	0.020	13727	Und	...	Au	S-111
S-112	270	589	18900	2135	...	..	..	..	...	...	18900	8000	472	..	0.42	0.025	14177	Und	...	Au	S-112
S-113	225	625	11400	1485	...	485	..	160	...	...	11858	3712	212	..	0.31	0.018	14199	Und	CF	Au	S-113
S-114	290	700	28210	6338	...	..	..	..	16350	...	28210	10800	377	0.58	0.39	0.014	12700	Und	...	Ml	S-114
S-115	265	620	14217	2582	...	..	..	131	6853	...	14217	7200	310	0.48	0.51	0.022	13727	Und	...	Au	S-115
S-116	275	600	12625	3863	...	..	..	..	8256	...	12625	4450	264	0.66	0.36	0.021	13029	Und	...	Ml	S-116
S-117	290	700	29130	5900	...	..	..	..	16350	...	29130	15000	474	0.56	0.52	0.016	12700	Und	...	Ml	S-117
S-118	290	700	29130	5900	...	..	..	..	16350	...	29130	15000	474	0.56	0.52	0.016	12700	Und	...	Ml	S-118
S-119	290	700	29130	5900	...	..	..	..	16350	...	29130	15000	474	0.56	0.52	0.016	12700	Und	...	Ml	S-119
S-120	290	700	29130	5900	...	..	..	..	16350	...	29130	15000	474	0.56	0.52	0.016	12700	Und	...	Ml	S-120
S-121	290	700	28210	6338	...	..	..	..	16350	...	28210	10800	377	0.58	0.39	0.014	12700	Und	...	Au	S-121
S-122	290	700	29130	5900	...	..	..	..	16350	...	29130	15000	474	0.56	0.52	0.016	12700	Und	...	Ml	S-122
S-123	265	600	14217	2582	...	..	..	157	5250	22072	14217	6650	310	0.37	0.47	0.022	13727	Und	...	Au	S-123

Pulverized-Fuel-Fired Units, 500 to 700 Lb per Sq In. Gage Pressure

P-1	630	735	15110	3400	...	302	..	2000	7300	9888	15412	12916	...	0.47	0.84	...	14111	St	...	Ml	P-1

Pulverized-Fuel-Fired Units, 400 to 499 Lb per Sq In. Gage Pressure

P-2	400	700	13920	6000	...	..	..	1150	8448	...	13920	14300	...	0.60	1.00	...	9751	St	...	Au	P-2
P-3	400	700	13920	6000	...	..	..	1150	8448	...	13920	14300	...	0.60	1.00	...	9751	St	...	Au	P-3
P-4	400	750	18096	2870	...	830	..	1400	...	...	18922	14267	...	..	0.75	...	10609	St	...	Ml	P-4
P-5	400	700	13920	6000	...	..	..	1150	8448	...	13920	14300	...	0.60	1.00	...	9751	St	...	Au	P-5
P-6	400	700	13920	6000	...	..	..	1150	8448	...	13920	14300	...	0.60	1.00	...	9751	St	...	Au	P-6
P-7	400	750	18092	2901	...	830	..	1400	...	...	18922	14267	...	..	0.75	...	10609	St	...	Ml	P-7
P-8	425	725	60706	13900	...	6331	1014	..	...	82720	68051	38200	...	..	0.56	...	14633	St	EP	Au	P-8
P-9	425	725	60706	13900	...	6331	1014	..	...	82720	68051	38200	...	..	0.56	...	14633	St	EP	Au	P-9
P-10	425	725	60706	13900	...	6331	1014	..	...	82720	68051	38200	...	..	0.56	...	13633	St	EP	Au	P-10
P-11	400	700	21620	3545	...	2578	..	..	..	16400	24198	18000	...	..	0 75	...	12385	St	...	Ce	P-11
P-12	400	750	17864	3608	...	790	337	1400	...	...	18991	14267	...	..	0.75	...	10609	St	...	Ml	P-12
P-13	410	700	14809	3430	...	3474	560	..	...	28900	18943	15888	...	..	0.84	...	14633	St	Cy	Au	P-13
P-14	410	700	14809	3430	...	3474	560	..	...	28900	18943	15888	...	..	0.84	...	14633	St	Cy	Au	P-14
P-15	416	700	29087	6070	...	..	1220	2400	18849	...	30307	25140	...	0 63	0.83	...	13630	St	EP	Ml	P-15

TABLE 1 DESCRIPTIVE DATA AND DESIGN FACTORS (*Continued*)

Unit No.	Steam pressure, lb per sq in. gage	Steam temperature, F	Heating surface in sq ft: Boiler	Superheater	Reheater	Water-cooled walls	Hearth screen	Air-cooled walls	Economizer	Air preheater	Total saturated heating surface	Volume of combustion chamber, cu ft	Projected grate area, sq ft	Ratio economizer to saturated surface	Ratio furnace volume to saturated surface	Ratio grate area to saturated surface	Heat value of fuel, Btu per lb	Method of firing[a]	Type of dust collector[b]	Combustion control[c]	Unit No.
Pulverized-Fuel-Fired Units, 400 to 499 Lb. per Sq In. Gage Pressure (Continued)																					
P-16	400	700	21620	3545	...	2578	..	..	...	16400	24198	18000	...	..	0.75	...	12385	St	...	Ce	P-16
P-17	410	700	14809	3430	...	3474	560	..	...	28900	18943	15888	...	..	0.84	...	14633	St	Cy	Au	P-17
P-18	410	700	14809	3430	...	3474	560	..	...	28900	18943	15888	...	..	0.84	...	14633	St	Cy	Au	P-18
P-19	416	700	29087	6070	...	..	1220	2400	18849	...	30307	25140	...	0.63	0.83	...	13630	St	EP	Ml	P-19
P-20	416	700	29087	6070	...	..	1220	2400	18849	...	30307	25140	...	0.63	0.83	...	13630	St	EP	Ml	P-20
P-21	410	700	14809	3430	...	3474	560	..	...	28900	18943	15888	...	..	0.84	...	14633	St	Cy	Au	P-21
P-22	410	700	14809	3430	...	3474	560	..	...	28900	18943	15888	...	..	0.84	...	14633	St	Cy	Au	P-22
P-23	400	700	10620	3500	...	135	330	1100	9180	...	11085	9102	...	0.83	0.82	...	13700	St	...	Ml	P-23
P-24	416	700	29087	6070	...	..	1220	2400	18849	...	30307	25140	...	0.63	0.83	...	13630	St	EP	Ml	P-24
P-25	400	700	15360	..	...	1450	320	..	...	21600	17130	8250	...	..	0.48	...	12880	Un	SN	Ml	P-25
P-26	400	700	15360	..	...	1450	320	..	...	21600	17130	8250	...	..	0.48	...	12880	Un	SN	Ml	P-26
P-27	400	700	15360	4000	...	1805	715	1150	...	...	21276	18539	...	..	0.88	...	7565	St	...	Ml	P-27
P-28	400	700	18756	4000	...	1805	715	1150	...	...	21276	18539	...	..	0.88	...	7565	St	...	Ml	P-28
P-29	400	700	18756	4000	...	1805	715	1150	...	...	21276	18539	...	..	0.88	...	7565	St	...	Ml	P-29
P-30	400	700	18756	4000	...	1805	715	1150	...	...	21276	18539	...	..	0.88	...	7565	St	...	Ml	P-30
P-31	400	700	15360	..	...	1450	320	..	...	21600	17130	8250	...	..	0.48	...	12880	Un	SN	Ml	P-31
Pulverized-Fuel-Fired Units, 300 to 399 Lb per Sq In. Gage Pressure																					
P-32	375	690	5940	3000	...	2468	..	..	8850	41700	8408	12397	...	1.00	1.50	...	10395	Un	Cy	Ml	P-32
P-33	375	690	6811	2430	...	1772	1402	..	5250	25000	9985	5000	...	0.52	0.50	...	10395	Un	...	Ml	P-33
P-34	385	700	9571	3420	...	842	483	729	5016	10980	10896	12800	...	0.46	1.20	...	13938	St	...	Ml	P-34
P-35	385	700	9571	3420	...	842	483	729	5016	10980	10896	12800	...	0.46	1.20	...	13938	St	...	Ml	P-35
P-36	385	700	9571	3420	...	842	483	729	5016	10980	10896	12800	...	0.46	1.20	...	13938	St	...	Ml	P-36
P-37	385	700	9571	3420	...	842	483	729	5016	10980	10896	12800	...	0.46	1.20	...	13938	St	...	Ml	P-37
P-38	385	700	9571	3420	...	842	483	729	5016	10980	10896	12800	...	0.46	1.20	...	13938	St	...	Ml	P-38
P-39	385	700	9571	3420	...	842	483	729	5016	10980	10896	12800	...	0.46	1.20	...	13938	St	...	Ml	P-39
P-40	385	700	9571	3420	...	842	483	729	5016	10980	10896	12800	...	0.46	1.20	...	13938	St	...	Ml	P-40
P-41	350	?	14770	2130	...	..	450	3275	7776	...	15220	11000	...	0.51	0.72	...	13804	St	...	Ml	P-41
P-42	300	650	17251	2869	...	1693	..	..	8620	11000	18944	13200	...	0.44	0.70	...	12048	Un	...	Ml	P-42
P-43	385	700	9571	3420	...	842	483	729	5016	10980	10896	12800	...	0.46	1.20	...	13938	St	...	Ml	P-43
P-44	350	?	14770	2130	...	..	450	3275	7776	...	15220	11000	...	0.51	0.72	...	13804	St	...	Ml	P-44
P-45	375	640	10850	6288	...	..	275	1652	2470	...	11125	8160	...	0.22	0.73	...	10250	St	...	Ml	P-45
P-46	300	650	17251	2869	...	1693	..	..	8620	11000	18944	13200	...	0.44	0.70	...	12048	Un	...	Ml	P-46
P-47	350	?	14770	2130	...	..	450	3275	7776	...	15220	11000	...	0.51	0.72	...	13804	St	...	Ml	P-47
P-48	350	?	14770	2130	...	..	450	3275	7776	...	15220	11000	...	0.51	0.72	...	13804	St	...	Ml	P-48
P-49	350	?	14770	2130	...	..	450	3275	7776	...	15220	11000	...	0.51	0.72	...	13804	St	...	Ml	P-49
P-50	350	?	14770	2130	...	..	450	3275	7776	...	15220	11000	...	0.51	0.72	...	13804	St	...	Ml	P-50
P-51	380	265	20468	4094	...	2050	..	..	...	11700	22518	13500	...	..	0.60	...	12385	St	...	Ce	P-51
P-52	375	640	10850	6288	...	..	275	1652	2470	...	11125	8160	...	0.22	0.73	...	10250	St	...	Ml	P-52
P-53	380	265	20468	4094	...	2050	..	..	...	11700	22518	13500	...	..	0.60	...	12385	St	...	Ce	P-53
P-54	380	265	20468	4094	...	2050	..	..	...	11700	22518	13500	...	..	0.60	...	12385	St	...	Ce	P-54
P-55	380	265	20468	4094	...	2050	..	..	...	11700	22518	13500	...	..	0.60	...	12385	St	...	Ce	P-55
P-56	305	675	27780	5480	...	1980	520	..	...	56000	30280	14800	...	..	0.49	...	13163	St	...	Ml	P-56
P-57	375	640	10850	6288	...	..	275	1652	2470	...	11125	8160	...	0.22	0.73	...	10250	St	...	Ml	P-57
P-58	375	640	10850	6288	...	..	275	1652	2470	...	11125	8160	...	0.22	0.73	...	10250	St	...	Ml	P-58
P-59	305	675	27780	5480	...	1980	520	..	...	56000	30280	14800	...	..	0.49	...	13163	St	...	Ml	P-59
P-60	385	560	14056	2220	...	340	435	1325	...	...	14831	11500	...	..	0.78	...	11993	St	...	Au	P-60
P-61	385	560	14056	2220	...	340	435	1325	...	...	14831	11500	...	..	0.78	...	11993	St	...	Au	P-61
P-62	305	675	27780	5480	...	1980	520	..	...	56000	30280	14800	...	..	0.49	...	13163	St	...	Ml	P-62
P-63	385	560	14056	2220	...	340	435	1325	...	...	14831	11500	...	..	0.78	...	11993	St	...	Au	P-63
P-64	300	700	13138	?	...	850	..	..	3240	45000	13988	5270	...	0 23	0.38	...	12048	Un	...	Ml	P-64
P-65	300	700	13138	?	...	850	..	..	3240	45000	13988	5270	...	0.23	0.38	...	12048	Un	...	Ml	P-65
P-66	390	725	30600	3275	...	..	792	2490	10560	...	31392	30600	...	0.34	0.98	...	12723	St	EP	Au	P-66
P-67	317	420	17650	..	...	..	630	?	13700	...	18280	12000	...	0.78	0.68	...	13508	St	...	Au	P-67
P-68	390	725	30600	3275	...	..	792	2490	10560	...	31392	30600	...	0.34	0.98	...	12723	St	EP	Au	P-68
P-69	390	725	30600	3275	...	..	792	2490	10560	...	31392	30600	...	0.34	0.98	...	12723	St	EP	Au	P-69
P-70	390	725	30600	3275	...	..	792	2490	10560	...	31392	30600	...	0.34	0.98	...	12723	St	EP	Au	P-70
P-71	330	585	12795	4780	...	1705	..	..	...	53400	14500	9600	...	..	0.66	...	13095	Un	...	Ml	P-71
P-72	390	725	30600	3275	...	..	792	2490	10560	...	31392	30600	...	0.34	0.98	...	12723	St	EP	Au	P-72
P-73	390	725	30600	3275	...	..	792	2490	10560	...	31392	30600	...	0.34	0.98	...	12723	St	EP	Au	P-73
P-74	330	605	13135	4100	...	1214	..	..	...	53400	14349	9600	...	..	0.67	...	13095	Un	...	Ml	P-74
Pulverized-Fuel-Fired Units, 200 to 299 Lb per Sq In. Gage Pressure																					
P-75	275	750	13806	7682	...	1825	..	..	13608	...	15631	11626	...	0.87	0.74	...	12376	Un	...	Au	P-75
P-76	275	750	12515	7100	...	2725	..	..	9504	...	15240	10200	...	0.62	0.67	...	12376	Un	...	Au	P-76
P-77	275	750	12515	7100	...	2044	..	..	9504	...	14559	11000	...	0.65	0.76	...	12376	Un	...	Au	P-77
P-78	290	650	13512	3863	...	450	270	40	4833	...	14182	10600	...	0.34	0.75	...	14869	Un	...	Ml	P-78
P-79	275	675	27680	5657	...	1425	640	1030	...	13000	29745	17050	...	..	0 57	...	13163	St	...	Ml	P-79
P-80	275	675	2768	5657	...	1464	640	..	...	13000	29784	17000	...	..	0.57	...	13163	St	...	Ml	P-80

Pulverized-Fuel-Fired Units, 200 to 299 Lb per Sq In. Gage Pressure (Continued)

P-81	265	725	30600	9040	...	..	835	1800	22080	...	31445	26000	...	0.71	0.84	...	12370	St	...	Ml	P-81
P-82	265	725	30600	9040	...	..	835	1800	22080	...	31445	26000	...	0.71	0.84	...	12370	St	...	Ml	P-82
P-83	265	725	30600	9040	...	..	835	1800	22080	...	31445	26000	...	0.71	0.84	...	12370	St	...	Ml	P-83
P-84	265	725	30600	9040	...	..	835	1800	22080	...	31445	26000	...	0.71	0.84	...	12370	St	...	Ml	P-84
P-85	275	675	26510	5657	...	1425	640	1030	...	13000	28575	17050	...	..	0.61	...	13163	St	...	Ml	P-85
P-86	275	675	27680	5657	...	1744	640	..	...	13000	30064	17000	...	..	0.61	...	13163	St	...	Ml	P-86
P-87	275	675	27680	5657	...	1744	640	..	...	13000	30064	17000	...	..	0.61	...	13163	St	...	Ml	P-87
Oil- and Gas-Fired Units																					
OG-1	375	700	14200	3210	...	609	..	..	...	74600	14809	5335	...	..	0.36	...	...	OG	...	Ml	OG-1
OG-2	375	700	14200	3210	...	609	..	..	...	74600	14809	5335	...	..	0.36	...	...	OG	...	Ml	OG-2
OG-3	450	750	34160	6553	...	3220	..	..	...	51232	37380	19800	...	..	0.54	...	...	OG	...	Au	OG-3
OG-4	425	725	15630	4137	...	2729	338	..	...	84000	18359	16500	...	..	0.90	...	...	Oil	...	Au	OG-4
OG-5	400	725	15340	5780	...	600	..	224	6804	32800	15940	8000	...	0.43	0.50	...	...	OG	...	Au	OG-5
OG-6	425	725	17440	4137	...	2729	338	..	...	84000	20507	16500	...	..	0.83	...	...	Oil	...	Au	OG-6
OG-7	375	700	15300	3712	...	1440	414	394	...	16850	17154	8650	...	..	0.50	...	...	OG	...	Au	OG-7
OG-8	425	725	17440	4137	...	2729	338	..	...	84000	20507	16500	...	..	0.83	...	...	Oil	...	Au	OG-8
OG-9	375	700	15300	3712	...	1440	414	394	...	16850	17154	8650	...	..	0.50	...	...	OG	...	Au	OG-9
OG-10	375	700	15300	3712	...	1440	414	394	...	16850	17154	8650	...	..	0.50	...	...	OG	...	Au	OG-10
OG-11	375	700	15300	3712	...	1440	414	394	...	16850	17154	8650	...	..	0.50	...	...	OG	...	Au	OG-11
OG-12	385	700	25450	3331	...	..	..	1614	...	23025	25450	9700	...	..	0.32	...	...	OG	...	Ml	OG-12
OG-13	350	650	17041	2361	...	..	..	952	...	...	17041	5502	...	..	0.32	...	...	OG	...	Ml	OG-13
OG-14	350	650	17041	2361	...	..	..	952	...	...	17041	5502	...	..	0.32	...	...	OG	...	Ml	OG-14
OG-15	350	650	17041	2361	...	..	..	952	...	...	17041	5502	...	..	0.32	...	...	OG	...	Ml	OG-15
OG-16	385	700	25450	3331	...	..	..	1614	...	23025	25450	9700	...	..	0.39	...	...	OG	...	Ml	OG-16
OG-17	385	700	25450	3331	...	..	..	1614	...	23025	25450	9700	...	..	0.39	...	...	OG	...	Ml	OG-17
OG-18	400	700	25560	4072	...	..	..	1614	...	21574	25560	9700	...	..	0.43	...	...	OG	...	Ml	OG-18
OG-19	400	700	25560	4072	...	..	..	1614	...	21574	25560	9700	...	..	0.43	...	...	OG	...	Ml	OG-19
OG-20	460	725	35000	6553	...	1581	..	..	...	51323	36581	14600	...	..	0.40	...	...	OG	...	Au	OG-20
OG-21	460	725	35000	6553	...	1581	..	..	...	51323	36581	14600	...	..	0 40	...	...	OG	...	Ml	OG-21
OG-22	350	650	17041	2361	...	..	..	952	...	...	17041	5502	...	..	0.32	...	...	OG	...	Ml	OG-22
OG-23	400	700	25560	4072	...	..	..	1614	...	21574	25560	9700	...	..	0.43	...	...	OG	...	Ml	OG-23
OG-24	375	685	12500	2400	...	330	..	..	1665	...	12830	6600	...	0.13	0.52	...	...	OG	...	Mi	OG-24
Miscellaneous Combinations																					
M-1	375	700	16863	5000	...	2570	540	..	...	25452	19973	14300	...	..	0.72	...	...	UP	...	Ml	M-1
M-2	375	700	16863	5000	...	2570	540	..	...	25452	19973	14300	...	..	0.72	...	...	UP	...	Ml	M-2
M-3	375	700	16863	5000	...	2570	540	..	...	25452	19973	14300	...	..	0.72	...	...	UP	...	Ml	M-3
M-4	375	685	12500	2400	...	330	..	..	1665	...	12830	6600	...	0.13	0.52	...	...	TO	...	Ml	M-4
M-5	375	685	12500	2400	...	330	..	..	1665	...	12830	6600	...	0.13	0.52	...	...	TO	...	Ml	M-5
M-6	375	685	12500	2400	...	330	..	..	1665	...	12830	6600	...	0.13	0.52	...	...	TO	...	Ml	M-6
Reheat Stoker-Fired Units, 500 to 700 Lb per Sq In. Gage Pressure																					
SR-1	650	725	5031	1215	10473	..	..	..	8837	...	5031	5760	356	1.750	1.145	0.071	11240	TG	...	Ml	SR-1
SR-2	650	725	5031	1215	10473	..	..	..	8837	...	5031	5760	356	1.750	1.145	0.071	11240	TG	...	Ml	SR-2
SR-3	650	725	5031	1215	10473	..	..	..	8837	...	5031	5760	356	1.750	1.145	0.071	11240	TG	...	Ml	SR-3
SR-4	650	725	5031	1215	10473	..	..	..	8837	...	5031	5760	356	1.750	1.145	0.071	11240	TG	...	Ml	SR-4
SR-5	675	725	5978	1600	13481	?	..	..	3011	40500	5978	7790	528	0.510	1.300	0.088	10834	TG	...	Ml	SR-5
SR-6	675	725	5978	1600	13481	?	..	..	3011	40500	5978	7790	528	0.510	1.300	0.088	10834	TG	CT	Ml	SR-6
SR-7	625	725	5031	1381	10473	..	..	..	8503	10291	5031	4920	365	1.690	0.980	0.073	10460	TG	...	Ml	SR-7
SR-8	625	725	5031	1303	11158	..	..	..	8503	18470	5031	4700	365	1.690	0.935	0.073	10460	TG	...	Ml	SR-8
Reheat Pulverized-Fuel-Fired Units, 1890 Lb per Sq In. Gage Pressure																					
PR-1	1390	580	28532	905	933	1010	534	..	...	66800	30076	30100	...	..	1.000	...	13500	St	...	Au	PR-1
Reheat Pulverized-Fuel-Fired Units, 500 to 700 Lb per Sq In. Gage Pressure																					
PR-2	630	735	6066	1700	17612	302	..	2000	7300	9889	6368	12916	...	1.145	1.920	...	14411	St	...	Ml	PR-2

[a] Abbreviations for method of firing: TG, traveling grate; Und, underfeed; St, storage; Un, unit; OG, oil and gas; UP, unit pulverizers with oil and gas; TO, traveling grate with oil and gas.
[b] Abbreviations for type of dust collector: CT, cinder traps; CC, cinder catcher; CF, cinder vane fan; EP, electric precipitator; Cy, cyclone; SN, spray nozzle.
[c] Abbreviations for combustion control: Ml, manual; SA, semi-automatic; Ce, centralized; Au, automatic.

TABLE 2 ANALYSIS OF TIME UTILIZATION OF INDIVIDUAL STEAM-GENERATING UNITS

Unit No.	Total hours	Percentage of total period: Steaming	Banked	Reserve	Crippled	Outages classified as to cause, expressed as percentage of total outage: Scheduled cleaning and overhaul	Changes in equipment	Boiler and superheater	Reheat boiler	Furnace	Stoker	Pulverizing equipment and burners	Economizer and air preheater	Ash equipment	Draft equipment	Instruments	Not chargeable to unit	Availability factor	Use factor	Unit No.
Stoker-Fired Units, 500 to 700 Lb per Sq In. Gage Pressure																				
S-1	24830	37.5	31.8	8.8	21.9	32.5	0.7	4.6	..	27.6	7.9	..	25.5	0.4	0.5	...	0.3	0.781	0.693	S-1
S-2	17507	71.6	1.7	2.5	24.2	30.5	26.2	11.5	..	21.5	9.3	..	1.0	...	..	...	..	0.758	0.733	S-2
S-3	24992	44.6	23.0	12.9	19.5	41.0	..	16.0	..	16.6	4.7	..	11.9	..	3.5	...	0.3	0.805	0.676	S-3
S-4	24863	43.6	27.4	9.5	19.5	31.1	1.5	9.9	..	15.8	8.4	..	22.6	0.4	1.7	...	8.6	0.805	0.710	S-4
S-5	24964	45.5	24.4	9.1	21.0	34.0	0.7	5.5	..	20.7	4.7	..	20.8	3.6	5.3	...	4.7	0.790	0.699	S-5
S-6	17357	77.4	1.8	3.2	17.6	36.8	12.1	11.6	..	26.1	9.6	..	3.5	..	0.7	...	..	0.824	0.792	S-6
S-7	17359	74.4	2.4	3.4	19.8	38.3	11.9	7.4	..	21.8	17.8	..	2.8	..	..	...	..	0.802	0.768	S-7
S-8	17474	70.4	2.3	3.5	23.8	43.0	12.9	8.8	..	23.2	17.7	..	5.6	..	1.7	...	..	0.762	0.727	S-8
S-9	24699	59.7	16.0	6.8	17.5	28.4	0.6	6.6	..	25.8	6.9	..	26.6	..	3.5	...	1.6	0.825	0.757	S-9
S-10	24857	59.4	16.4	8.7	15.5	38.0	..	4.4	..	15.0	1.5	..	31.6	..	9.5	...	..	0.845	0.758	S-10
S-11	26230	59.1	15.6	2.1	23.2	37.5	6.3	16.4	..	9.0	13.5	..	10.3	..	2.4	...	4.6	0.768	0.747	S-11
S-12	24658	54.1	13.3	3.0	29.7	27.9	..	18.7	..	47.9	0.1	..	5.4	..	..	...	..	0.704	0.674	S-12
S-13	26206	59.0	15.0	2.0	24.0	28.4	2.7	14.5	..	7.3	6.9	..	35.8	..	1.0	...	3.4	0.760	0.740	S-13
S-14	24671	56.4	15.8	3.4	24.4	33.3	..	11.0	..	42.2	2.3	..	10.1	..	1.1	...	..	0.756	0.722	S-14
S-15	17474	72.3	6.1	5.8	15.8	8.5	..	31.2	..	23.1	17.0	..	14.2	0.7	1.6	...	3.7	0.842	0.784	S-15
S-16	17453	73.2	3.1	8.3	15.4	20.0	..	26.4	..	22.7	20.7	..	6.0	0.9	3.3	...	..	0.846	0.763	S-16
S-17	24709	62.3	10.6	3.2	23.9	21.7	1.2	18.4	..	49.2	2.9	..	6.2	..	..	...	0.4	0.761	0.729	S-17
S-18	24630	56.3	6.7	3.6	33.4	42.8	7.5	10.9	..	29.0	5.4	..	2.8	..	1.2	...	0.4	0.666	0.630	S-18
S-19	17679	71.5	4.5	8.4	15.6	15.3	0.4	28.6	..	32.8	3.2	..	16.8	0.4	0.7	...	1.8	0.844	0.760	S-19
S-20	24515	60.3	11.6	2.6	25.5	33.3	1.5	18.6	..	45.1	0.2	..	1.3	..	..	...	..	0.745	0.719	S-20
S-21	24744	60.6	9.7	3.2	26.5	30.2	..	7.0	..	44.5	5.8	..	11.5	..	1.0	...	..	0.735	0.703	S-21
S-22	17464	71.7	3.8	6.8	17.7	14.3	..	28.5	..	23.5	8.3	..	13.3	0.5	1.7	...	9.9	0.823	0.755	S-22
S-23	17474	72.4	5.8	2.0	19.8	18.5	0.5	44.4	..	17.2	12.7	..	2.6	..	..	...	4.1	0.802	0.782	S-23
S-24	17470	71.4	4.9	8.5	15.2	14.0	2.2	26.8	..	24.4	20.4	..	5.1	2.6	1.4	...	3.1	0.848	0.763	S-24
Stoker-Fired Units, 400 to 499 Lb per Sq In. Gage Pressure																				
S-25	19824	51.9	23.2	2.3	22.6	55.6	1.0	1.3	..	5.7	20.3	..	1.3	5.9	0.8	...	8.1	0.774	0.751	S-25
S-26	24791	69.2	6.9	5.7	18.2	32.2	..	10.5	..	16.7	23.9	..	8.3	..	6.8	...	1.6	0.818	0.761	S-26
S-27	19851	46.4	27.1	2.2	24.3	46.7	31.8	7.3	..	4.5	9.2	..	0.5	..	..	...	..	0.757	0.735	S-27
S-28	19797	47.2	23.8	2.2	26.8	45.9	39.0	0.9	..	3.2	6.4	..	0.6	2.5	..	...	1.5	0.732	0.710	S-28
S-29	19814	45.5	26.0	4.5	24.0	45.6	41.4	1.9	..	1.7	6.5	..	0.7	..	0.5	...	1.7	0.760	0.715	S-29
S-30	13033	82.5	5.6	0.6	11.3	40.8	..	17.3	..	6.8	14.8	..	4.6	4.6	8.3	...	2.8	0.887	0.881	S-30
S-31	13033	80.2	6.4	0.6	12.7	35.8	..	16.4	..	11.8	13.6	..	4.4	4.8	11.5	...	1.7	0.872	0.866	S-31
S-32	23442	58.6	4 2	0.6	36.6	29.8	1.9	32.2	..	10.7	18.9	..	2.7	0.3	1.5	...	2.0	0.634	0.628	S-32
S-33	13033	83.8	5.2	0.5	10.5	39.8	..	15.5	..	7.2	14.6	..	2.7	8.4	11.7	...	..	0.895	0.890	S-33
S-34	13033	84.5	5.5	0.6	9.4	43.0	3.4	12.2	..	8.7	13.3	..	4.2	6.0	9.1	...	0.1	0.906	0.900	S-34
S-35	23352	65.5	4.5	0.6	29.4	32.1	1.9	25.5	..	4.4	22.4	..	5.7	..	3.8	...	4.2	0.706	0.700	S-35
S-36	23194	56.0	5.2	0.6	38.2	51.0	4.4	12.9	..	4.9	20.6	..	4.1	0.1	1.7	...	0.3	0.618	0.612	S-36
Stoker-Fired Units, 300 to 399 Lb per Sq In. Gage Pressure																				
S-37	17016	94.4	0.4	1.5	3.7	46.8	6.0	0.6	..	21.9	5.7	..	10.0	..	6.0	...	3.0	0.963	0.948	S-37
S-38	13320	68.4	8.7	2.6	20.3	99.7	..	..	..	0.3	..	..	..	..	..	...	..	0.797	0.771	S-38
S-39	8790	78.6	5.8	14.5	1.1	No analysis of outage given												0.989	0.844	S-39
S-40	8790	78.6	5.8	14.9	0.7	No analysis of outage given												0.993	0.844	S-40
S-41	8790	78.6	5.8	14.4	1.2	No analysis of outage given												0.988	0.844	S-41
S-42	8790	78.6	5.8	13.8	1.8	No analysis of outage given												0.982	0.844	S-42
S-43	8790	78.6	5.8	13.9	1.7	No analysis of outage given												0.983	0.844	S-43
S-44	8790	78.8	5.8	14.2	1.2	No analysis of outage given												0.988	0.846	S-44
S-45	8790	78.3	5.7	16.0	..	No analysis of outage given												1.000	0.840	S-45
S-46	8790	55.9	31.4	6.3	6.4	18.7	..	19.8	..	0.9	48.0	..	12.6	..	..	...	..	0.936	0.873	S-46
S-47	18155	25.7	13.8	37.7	22.8	37.7	9.4	25.1	..	2.6	23.2	..	..	0.5	..	...	1.5	0.772	0.395	S-47
S-48	18088	23.9	12.7	33.8	29.6	37.9	2.5	6.5	..	15.5	28.9	..	3.9	0.9	0.5	...	3.4	0.704	0.366	S-48
S-49	18116	45.8	15.9	17.8	20.5	59.7	10.7	4.4	..	2.0	13.2	..	9.8	..	..	...	0.6	0.795	0.617	S-49
S-50	25172	69.4	2.4	4.7	23.5	19.8	..	22.4	..	8.9	31.6	..	9.1	7.8	0.4	...	..	0.765	0.718	S-50
S-51	25205	69.0	3.0	3.9	24.1	15.7	..	27.2	..	10.4	31.4	..	8.4	5.7	1.2	...	..	0.759	0.720	S-51
S-52	25294	67.8	1.8	4.2	26.2	19.5	..	21.2	..	9.0	32.7	..	7.1	10.0	0.5	...	..	0.738	0.696	S-52
S-53	21322	45.9	20.9	10.4	22.8	45.4	..	3.8	..	8.3	13.6	..	26.0	2.3	..	...	0.6	0.772	0.668	S-53
S-54	18439	39.1	18.4	11.9	30.6	38.9	2.6	5.7	..	3.6	7.7	..	37.1	..	0.8	...	3.6	0.694	0.575	S-54
S-55	26212	80.5	12.9	2.0	4.6	42.1	4.7	4.6	..	22.2	13.0	..	4.9	..	7.5	...	1.0	0.954	0.934	S-55
S-56	18144	44.7	16.0	16.8	22.5	57.3	1.0	9.5	..	0.2	18.1	..	11.4	..	..	...	2.5	0.775	0.607	S-56
S-57	8736	71.5	17.3	5.9	5.3	..	..	1.9	..	34.7	17.8	..	45.6	..	..	...	..	0.947	0.888	S-57
S-58	26365	88.4	4.5	2.6	4.5	48.1	4.9	5.2	..	16.7	11.8	..	4.3	..	8.0	...	1.0	0.955	0.929	S-58
S-59	24480	71.0	10.5	0.7	17.8	16.3	17.3	30.4	..	14.8	18.9	..	..	..	1.2	...	1.1	0.822	0.815	S-59
S-60	25931	72.6	8.6	0.8	18.0	20.0	25.8	20.2	..	13.1	19.1	..	..	..	0.3	...	1.5	0.820	0.812	S-60
S-61	26200	86.5	5.5	1.5	6.5	36.3	9.6	10.9	..	16.9	10.7	..	2.7	0.1	10.3	...	2.5	0.935	0.920	S-61
S-62	26210	91.3	0.7	0.3	7.7	25.5	16.4	19.5	..	13.7	19.9	..	3.2	..	1.8	...	..	0.923	0.920	S-62
S-63	28558	72.7	8.5	8.1	10.7	24.2	6.6	30.4	..	19.4	18.5	..	..	0.6	0.3	...	..	0.893	0.812	S-63
S-64	8736	42.2	32.4	11.1	14.3	63.0	..	6.3	..	16.3	14.4	..	..	..	..	...	..	0.857	0.746	S-64

Stoker-Fired Units, 300 to 399 Lb per Sq In. Gage Pressure (Continued)

S-65	26208	79.0	9.2	0.5	11.3	25.8	..	36.4	..	16.4	20.8	..	..	..	0.3	...	0.3	0.887	0.882	S-65
S-66	26208	74.9	11.8	0.3	13.0	22.8	1.4	29.1	..	23.4	21.2	..	..	..	..	...	2.1	0.870	0.867	S-66
S-67	26289	76.6	9.6	0.6	13.2	23.7	0.2	27.4	..	27.9	18.9	..	0.7	..	0.5	...	0.7	0.868	0.862	S-67
S-68	26280	77.2	10.9	0.2	11.7	26.4	0.4	30.6	..	20.0	20.6	..	..	0.6	0.6	...	0.8	0.883	0.881	S-68
S-69	26209	73.7	9.9	0.8	15.6	35.0	6.6	22.9	..	18.2	16.7	..	..	0.4	0.2	...	..	0.844	0.836	S-69
S-70	26165	80.0	8.8	0.2	11.0	36.0	0.3	28.2	..	14.1	18.9	..	..	0.3	2.2	...	..	0.890	0.888	S-70
S-71	26309	76.1	11.0	0.4	12.5	27.2	0.6	26.2	..	23.8	20.5	..	..	0.3	1.2	...	..	0.875	0.871	S-71
S-72	17304	71.3	1.4	11.3	16.0	26.5	7.6	39.5	..	1.3	17.6	..	..	..	7.1	...	0.4	0.840	0.727	S-72
S-73	3528	66.0	23.9	5.2	4.9	36.4	..	36.4	..	27.2	..	..	..	..	..	...	..	0.951	0.899	S-73
S-74	13272	69.6	5.2	1.2	24.0	90.5	..	5.4	..	1.5	2.6	..	..	..	..	...	..	0.760	0.748	S-74
S-75	24933	27.8	17.6	24.9	29.7	40.0	..	24.4	..	10.9	21.0	..	..	3.7	..	...	..	0.703	0.454	S-75
S-76	26399	77.0	9.0	0.5	13.5	36.3	3.7	24.9	..	10.0	24.1	..	..	0.1	0.9	...	..	0.865	0.860	S-76
S-77	25745	76.4	11.2	0.6	11.8	23.6	0.3	32.7	..	20.9	19.1	..	..	..	1.1	...	2.3	0.882	0.876	S-77
S-78	25189	25.8	17.1	27.8	29.3	9.3	..	30.1	..	20.8	30.3	..	..	7.3	1.3	...	0.9	0.707	0.429	S-78
S-79	13080	75.1	13.7	1.8	9.4	96.4	..	1.8	..	..	1.8	..	..	..	..	...	..	0.906	0.888	S-79
S-80	8736	61.5	0.6	12.2	25.7	8.2	3.7	28.4	..	24.8	30.7	..	..	2.1	2.1	...	..	0.743	0.621	S-80
S-81	8736	69.0	4.5	6.3	20.2	12.1	..	41.2	..	17.9	27.6	..	0.6	..	0.6	...	..	0.798	0.735	S-81
S-82	26581	33.2	18.3	20.4	28.1	22.0	..	24.4	..	23.2	22.2	..	..	8.2	..	...	..	0.719	0.515	S-82
S-83	19817	63.9	1.7	6.5	27.9	11.7	1.2	17.3	..	7.6	5.1	..	0.2	4.1	1.2	...	51.6	0.721	0.656	S-83
S-84	26304	66.0	4.9	12.0	17.1	21.3	26.0	37.6	..	3.6	11.0	..	..	..	..	...	0.5	0.829	0.709	S-84
S-85	17404	72.0	1.4	10.9	15.7	28.9	..	32.2	..	5.3	23.5	..	..	..	9.1	...	1.0	0.843	0.734	S-85
S-86	18738	68.2	4.1	8.7	19.0	18.8	..	10.1	..	2.6	3.6	..	..	7.3	0.4	...	57.2	0.810	0.723	S-86
S-87	18816	71.0	3.7	10.0	15.3	12.4	..	10.8	..	2.6	9.4	..	..	0.5	0.9	...	63.4	0.847	0.747	S-87
S-88	18812	68.2	2.8	13.5	15.5	13.4	1.3	8 8	..	5.3	15.4	..	0.5	..	2.0	...	53.3	0.845	0.710	S-88
S-89	26108	69.5	7.6	6.8	16.1	42.9	6.8	23.3	..	1.3	19.2	..	..	..	5.7	...	0.8	0.839	0.771	S-89

Stoker-Fired Units, 200 to 299 Lb per Sq In. Gage Pressure

S-90	21570	26.8	52.6	9.0	11.6	3.2	42.3	5.6	..	10.7	22.7	..	4.1	4.7	5.8	...	0.9	0.884	0.794	S-90
S-91	21673	21.4	42.4	26.1	10.1	11.1	35.5	12.5	..	15.3	20.9	..	0.4	..	2.8	...	1.5	0.899	0.638	S-91
S-92	21692	26.0	51.0	10.5	12.5	37.1	..	22.7	..	22.8	3.1	..	12.0	1.3	0.4	...	0.6	0.875	0.770	S-92
S-93	5712	45.2	50.5	2.3	2.0	..	..	18.8	..	7.1	58.0	..	..	16.1	..	...	..	0.980	0.957	S-93
S-94	18478	71.5	10.8	1.1	16.6	19.7	..	10.6	..	24.0	23.2	..	3.6	..	14.4	...	4.5	0.834	0.823	S-94
S-95	18475	72.5	10.1	0.1	17.3	9.0	..	12.6	..	33.5	25.8	..	6.0	..	8.7	1.0	3.4	0.827	0.826	S-95
S-96	5712	50.6	32.1	2.5	14.8	91.5	..	5.6	..	0.8	..	..	..	2.1	..	...	..	0.852	0.827	S-96
S-97	7222	62.5	15.2	5.7	16.6	48.0	..	34.6	..	4.7	12.7	..	..	..	..	...	..	0.834	0.777	S-97
S-98	17472	25.8	28.4	36.0	9.8	14.0	17.6	42.1	..	4.5	13.9	..	..	..	2.9	...	5.0	0.902	0.542	S-98
S-99	7224	63.5	24.4	5.1	7.0	66.0	..	14.5	..	..	..	..	19.5	..	..	...	..	0.930	0.879	S-99
S-100	5712	33.1	30.2	1.7	35.0	88.6	..	4.3	..	..	7.1	..	..	..	..	...	..	0.650	0.633	S-100
S-101	18480	54.3	25.1	1.5	19.1	26.0	..	9.5	..	32.5	24.1	..	2.5	2.9	0.6	...	1.9	0.809	0.794	S-101
S-102	18478	70.8	10.7	1.0	17.5	13.1	1.4	13.7	..	26.9	22.3	..	2.5	1.6	5.8	...	12.7	0.825	0.815	S-102
S-103	13744	78.0	0.9	11.7	9.4	31.1	..	2.5	..	26.7	10.8	..	28.9	..	..	...	..	0.906	0.789	S-103
S-104	12264	76.0	5.4	0.4	18.2	6.6	..	10.2	..	7.8	41.9	..	..	21.6	7.9	3.1	0.9	0.818	0.814	S-104
S-105	13776	78.6	0.8	12.9	7.7	36.0	3.9	12.2	..	14.2	12.1	..	21.6	..	..	...	..	0.923	0.794	S-105
S-106	25526	64.5	14.8	1.6	19.1	39.4	7.2	32.2	..	12.3	3.6	..	..	0.6	3.5	...	1.2	0.809	0.793	S-106
S-107	25536	42.8	22.8	4.7	29.7	61.0	15.7	2.0	..	4.2	15.9	..	0.5	..	..	...	0.7	0.703	0.656	S-107
S-108	8904	80.9	8.1	4.6	6.4	16.0	..	18.2	..	27.8	36.6	..	..	1.4	..	...	..	0.936	0.890	S-108
S-109	25359	64.5	14.6	1.0	19.9	41.0	8.1	26.6	..	6.1	13.2	..	..	0.3	1.3	...	3.4	0.801	0.791	S-109
S-110	25578	69.0	14.7	1.2	15.1	54.2	..	22.3	..	10.7	6.3	..	..	0.2	..	4.6	1.7	0.849	0.837	S-110
S-111	26041	47.8	18.8	8.5	24.9	63.8	..	3.7	..	4.6	6.6	..	20.7	..	..	...	0.6	0.751	0.666	S-111
S-112	14784	60.0	5.8	0.2	34.0	1.5	89.4	2.2	..	2.0	3.3	..	..	..	1.6	...	..	0.660	0.658	S-112
S-113	25344	64.0	14.2	0.7	21.1	38.0	0.6	27.1	..	20.9	10.7	..	..	1.4	0.6	...	0.7	0.789	0.782	S-113
S-114	30507	55.2	17.1	11.7	16.0	34.7	..	16.6	..	42.4	6 3	..	..	..	..	...	..	0.840	0.723	S-114
S-115	25229	50.0	4.6	22.7	22.7	73.1	..	5.2	..	10.4	7.2	..	1.3	1.9	..	...	0.9	0.773	0.546	S-115
S-116	17472	67.7	10.0	11.2	11.1	35.1	3.1	8.6	..	16.4	25.8	..	8.1	0.2	1.5	...	1.2	0.889	0.777	S-116
S-117	30774	56.0	14.3	16.0	13.7	57.9	..	16.2	..	15.3	7.6	..	1.6	..	1.4	...	..	0.863	0.703	S-117
S-118	11147	50.0	16.5	17.7	15.8	27.4	..	41.2	..	22.5	8.3	..	0.6	..	..	...	..	0.842	0.665	S-118
S-119	30407	34.7	9.1	49.8	6.4	38.6	..	21.5	..	20.4	17.9	..	..	..	..	...	1.6	0.936	0.438	S-119
S-120	11256	54.8	13.5	21.3	10.4	24.5	..	10.7	..	35.1	17.4	..	4.8	..	3.2	...	4.3	0.896	0.683	S-120
S-121	30373	56.7	18.4	9.0	15.9	32.5	..	28.0	..	29.8	8.3	..	..	..	1.0	...	0.4	0.815	0.722	S-121
S-122	11250	48.6	16.3	13.3	21.8	14.6	..	22.2	..	56.2	6.1	..	0.9	..	..	...	..	0.782	0.649	S-122
S-123	25202	66.5	13.5	2.7	17.3	75.0	..	1.5	..	6.5	11.5	..	5.3	0.2	..	...	..	0.827	0.800	S-123

Pulverized-Fuel-Fired Units, 500 to 700 Lb per Sq In. Gage Pressure

P-1	17675	64.3	2.0	14.1	19.6	11.2	..	7.3	..	56.9	3.0	3.3	13.2	..	1.9	2.1	1.1	0.804	0.663	P-1

Pulverized-Fuel-Fired Units, 400 to 499 Lb per Sq In. Gage Pressure

P-2	8736	68.5	13.9	2.2	15.4	5.0	..	27.8	..	45.7	..	16.2	..	5.3	..	...	..	0.846	0.824	P-2
P-3	8736	58.0	14.9	7.7	19.4	9.9	7.8	62.2	..	9.4	..	6.6	..	4.1	..	...	..	0.806	0.729	P-3
P-4	17548	64.0	..	15.2	20.8	4.0	..	45.3	..	39.1	..	2.4	1.7	3.4	3.5	..	0.6	0.792	0.640	P-4
P-5	8736	61.2	17.2	9.2	12.4	8.9	4.8	21.7	..	42.4	..	18.5	..	..	3.7	...	..	0.876	0.784	P-5
P-6	8736	61.2	18.1	6.1	14.6	31.7	3.1	47.9	..	4.8	..	7.6	1.9	1.9	1.1	...	..	0.854	0.793	P-6
P-7	17448	70.4	..	11.7	17.9	1.5	..	31.0	..	57.3	..	..	3.8	0.8	5.2	...	0.4	0.821	0.704	P-7
P-8	10920	62.0	..	38.0	..	..	..	..	..	..	..	..	..	..	..	...	..	1.000	0.620	P-8
P-9	10920	67.5	..	32.5	..	..	..	..	..	..	..	..	..	..	..	...	..	1.000	0.675	P-9
P-10	10920	57.1	..	42.9	..	..	..	..	..	..	..	..	..	..	..	...	..	1.000	0.571	P-10
P-11	8064	82.8	1.1	1.0	15.1	14.2	..	51.4	..	..	..	4.6	1.5	27.6	0.7	...	..	0.849	0.839	P-11
P-12	17446	53.0	..	18.0	29.0	7.0	34.7	24.0	..	28.0	..	..	1.3	3.2	1.6	...	0.2	0.710	0.530	P-12
P-13	10920	80.2	..	19.8	..	..	..	..	..	..	..	..	..	..	..	...	..	1.000	0.802	P-13
P-14	10920	78.0	..	28.0	..	..	..	..	..	..	..	..	..	..	..	...	..	1.000	0.780	P-14

TABLE 2 ANALYSIS OF TIME UTILIZATION OF INDIVIDUAL STEAM-GENERATING UNITS (*Continued*)

Unit No.	Total hours	Percentage of total period: Steaming	Banked	Reserve	Crippled	Outages classified as to cause, expressed as percentage of total outage: Scheduled cleaning and overhaul	Changes in equipment	Boiler and superheater	Reheat boiler	Furnace	Stoker	Pulverizing equipment and burners	Economizer and air preheater	Ash equipment	Draft equipment	Instruments	Not chargeable to unit	Availability factor	Use factor	Unit No.
Pulverized-Fuel-Fired Units, 400 to 499 Lb per Sq In. Gage Pressure (Continued)																				
P-15	24870	56.1	14.2	17.4	12.3	16.0	..	9.7	..	52.7	..	1.4	10.4	3.8	2.0	...	4.0	0.877	0.703	P-15
P-16	7726	72.0	0.3	1.0	26.7	..	..	82.0	..	17.0	..	..	..	..	1.0	...	..	0.733	0.723	P-16
P-17	10920	79.8	0.4	19.8	..	..	..	..	..	..	..	..	..	..	..	...	..	1.000	0.802	P-17
P-18	10920	78.1	..	21.9	..	..	..	..	..	..	..	..	..	..	..	...	..	1.000	0.781	P-18
P-19	24865	54.8	11.0	20.9	13.3	45.8	..	10.6	..	25.5	..	0.3	3.1	4.4	6.2	...	4.1	0.867	0.658	P-19
P-20	24863	55.8	12.5	11.3	20.4	14.7	9.0	16.2	..	45.7	..	2.9	4.8	2.2	2.3	...	2.2	0.796	0.683	P-20
P-21	10920	76.8	..	23.2	..	..	..	..	..	..	..	..	..	..	..	...	..	1.000	0.768	P-21
P-22	10920	77.6	..	22.4	..	..	..	..	..	..	..	..	..	..	..	...	..	1.000	0.776	P-22
P-23	8067	48.8	33.8	11.8	5.6	17.9	..	19.7	..	5.5	..	28.8	17.0	..	4.9	6.2	..	0.944	0.826	P-23
P-24	24863	58.3	12.8	17.4	11.5	23.4	0.7	8.6	..	56.0	..	1.4	4.9	2.4	2.2	...	0.4	0.885	0.711	P-24
P-25	17824	44.4	15.6	33.0	7.0	14.2	29.6	8.9	..	5.3	..	37.2	..	..	0.4	...	3.9	0.930	0.600	P-25
P-26	18813	44.2	13.9	32.8	9.1	25.8	22.2	2.7	..	0.6	..	45.7	..	..	..	...	3.0	0.909	0.581	P-26
P-27	26208	76.7	13.8	4.4	5.1	41.9	..	25.5	..	11.8	..	7.2	..	2.6	4.2	...	6.8	0.949	0.905	P-27
P-28	25704	77.8	7.3	5.9	9.0	24.6	..	40.8	..	13.9	..	9.1	..	3.7	2.3	0.3	5.3	0.910	0.851	P-28
P-29	26208	76.4	10.4	4.2	9.0	27.9	0.8	19.7	..	14.7	..	5.7	..	0.2	20.4	..	10.6	0.910	0.868	P-29
P-30	26208	71.7	15.1	5.1	8.1	22.8	..	35.2	..	16.2	..	16.6	..	2.0	2.6	...	4.6	0.919	0.868	P-30
P-31	18628	51.2	12.8	33.2	2.8	38.1	47.1	0.5	..	..	..	5.5	..	..	8.8	...	..	0.972	0.640	P-31
Pulverized-Fuel-Fired Units, 300 to 399 Lb per Sq In. Gage Pressure																				
P-32	19320	29.9	..	45.0	25.1	12.3	28.3	17.2	..	6.0	..	30.2	1.5	0.2	0.8	...	3.5	0.749	0.299	P-32
P-33	21850	28.2	..	59.2	12.6	17.5	43.6	4.8	..	3.0	..	17.7	3.7	..	8.5	...	1.2	0.874	0.282	P-33
P-34	9408	62.0	11.0	14.6	12.4	34.2	..	8.3	..	39.5	..	..	6.9	..	9.0	...	2.1	0.676	0.730	P-34
P-35	9408	64.4	9.5	11.8	14.3	17.2	2.1	6.8	..	52.3	..	..	10.1	0.7	10.4	...	0.4	0.857	0.739	P-35
P-36	24764	51.2	10.8	18.9	19.1	29.8	0.5	14.3	..	11.2	..	0.4	29.4	1.0	5.7	...	7.7	0.809	0.620	P-36
P-37	24882	52.4	14.6	10.5	22.5	26.5	2.4	33.9	..	15.5	..	0.1	11.9	..	8.6	...	1.1	0.775	0.670	P-37
P-38	24845	33.0	21.4	27.1	18.5	24.3	4.9	18.9	..	29.1	..	..	14.4	0.1	8.3	...	..	0.815	0.544	P-38
P-39	24864	36.0	18.3	23.2	22.5	25.7	8.5	9.7	..	18.1	..	0.2	34.3	..	3.5	...	..	0.775	0.543	P-39
P-40	24963	30.1	21.2	15.1	33.6	34.2	2.9	22.6	..	11.8	..	..	18.1	0.6	9.8	...	..	0.664	0.513	P-40
P-41	8736	70.0	..	22.1	7.9	22.2	..	17.0	..	17.8	..	..	1.7	1.7	1.7	...	37.9	0.921	0.700	P-41
P-42	12935	68.5	..	17.1	14.4	49.4	11.9	8.9	..	3.1	..	0.9	2.1	..	0.8	...	22.9	0.856	0.685	P-42
P-43	25527	46.0	19.5	17.0	17.5	26.0	7.0	16.0	..	23.7	..	..	23.3	0.8	3.2	...	..	0.825	0.655	P-43
P-44	8736	74.5	10.0	0.3	15.2	86.5	..	7.7	..	..	..	..	5.8	..	..	...	..	0.848	0.845	P-44
P-45	16635	53.4	28.2	3.0	15.4	36.9	0.2	2.7	..	25.1	..	3.8	30.4	0.9	..	...	..	0.846	0.816	P-45
P-46	12936	66.6	..	7.3	26.1	26.6	34.0	7.0	..	11.5	..	0.5	4.9	1.7	0.5	...	13.3	0.739	0.666	P-46
P-47	8736	67.2	7.4	6.7	18.7	0.8	..	37.2	..	41.4	..	7.1	..	5.6	7.9	...	..	0.813	0.746	P-47
P-48	8736	73.3	3.0	6.8	16.9	12.4	..	37.0	..	23.8	..	13.2	..	1.5	12.1	...	..	0.831	0.763	P-48
P-49	8736	77.2	0.5	16.8	5.5	..	..	..	..	..	..	..	..	..	..	...	..	0.945	0.777	P-49
P-50	8736	64.0	4.3	4.6	27.1	1.6	..	9.8	..	52.2	..	33.6	0.6	..	..	...	2.2	0.729	0.683	P-50
P-51	8064	55.7	21.0	8.6	14.7	6.0	49.0	4.0	..	..	..	..	31.0	..	..	...	..	0.853	0.767	P-51
P-52	16616	67.1	11.6	12.4	8.9	28.9	7.7	7.1	..	14.6	..	4.7	20.2	0.4	0.4	8.0	8.0	0.911	0.787	P-52
P-53	8064	75.4	0.3	14.1	10.2	13.4	..	45.8	..	22.2	..	3.9	2.2	3.7	8.8	...	..	0.898	0.757	P-53
P-54	8064	77.2	0.6	16.8	5.4	2.4	..	9.5	..	83.0	..	..	..	..	5.1	...	..	0.946	0.778	P-54
P-55	8064	76.3	0.5	15.9	7.3	8.8	..	16.5	..	72.8	..	..	..	1.9	..	...	..	0.927	0.768	P-55
P-56	26085	84.1	0.3	1.3	14.3	10.1	29.7	3.1	..	1.8	..	..	29.9	..	16.0	...	9.4	0.857	0.844	P-56
P-57	16632	64.2	10.9	3.7	21.2	29.0	..	30.1	..	4.4	..	6.3	29.7	..	..	...	0.5	0.788	0.751	P-57
P-58	16608	61.7	14.8	4.4	19.1	42.5	1.8	11.9	..	8.0	..	1.8	32.3	1.6	0.1	...	..	0.809	0.765	P-58
P-59	17520	70.0	0.7	..	29.3	3.5	62.5	4.8	..	4.7	..	..	2.6	1.4	5.9	...	14.6	0.707	0.707	P-59
P-60	26030	56.3	14.1	24.8	4.8	46.7	..	5.4	..	36.7	..	..	..	4.6	6.6	...	..	0.952	0.704	P-60
P-61	26041	54.3	15.4	24.8	5.5	42.7	1.0	10.8	..	32.3	..	..	..	..	13.2	...	..	0.945	0.697	P-61
P-62	17528	78.4	0.6	..	21.0	..	72.0	4.1	..	2.0	..	..	6.5	..	0.9	...	14.5	0.790	0.790	P-62
P-63	26124	53.0	14.2	25.9	6.9	32.9	..	18.7	..	39.4	..	0.2	..	..	8.8	...	..	0.931	0.672	P-63
P-64	18702	67.7	..	8.4	23.9	33.0	5.5	3.6	..	7.3	..	0.5	12.5	7.0	3.6	...	27.0	0.761	0.677	P-64
P-65	18816	66.8	..	6.0	27.2	26.6	6.4	17.2	..	10.3	..	2.1	7.1	5.9	4.1	...	20.3	0.728	0.688	P-65
P-66	8232	42.3	24.2	28.2	5.3	38.5	..	53.5	..	..	..	..	8.0	..	..	...	..	0.947	0.665	P-66
P-67	8770	69.2	19.7	4.8	6.3	13.0	65.0	..	..	22.0	..	..	..	..	..	...	..	0.937	0.889	P-67
P-68	8064	67.3	8.4	19.5	4.8	45.3	..	54.7	..	..	..	..	..	..	..	...	..	0.952	0.757	P-68
P-69	8736	49.0	21.2	22.6	7.2	50.1	..	34.3	..	15.6	..	..	..	..	..	...	..	0.928	0.702	P-69
P-70	8736	72.2	8.7	12.1	7.0	46.9	..	53.1	..	..	..	..	..	..	..	...	..	0.930	0.809	P-70
P-71	26207	63.4	0.8	16.8	19.0	5.8	24.7	31.5	..	9.9	..	3.1	19.7	1.2	3.2	...	0.9	0.810	0.642	P-71
P-72	8736	59.9	4.2	25.3	10.6	18.5	..	81.5	..	..	..	..	..	..	..	...	..	0.894	0.641	P-72
P-73	8736	52.8	23.2	17.5	6.5	56.4	..	43.6	..	..	..	..	..	..	..	...	..	0.935	0.760	P-73
P-74	26050	64.0	0.9	11.8	23.3	9.7	15.1	20.6	..	19.8	..	22.2	11.4	..	0.9	...	0.3	0.767	0.649	P-74
Pulverized-Fuel-Fired Units, 200 to 299 Lb per Sq In. Gage Pressure																				
P-75	19556	38.8	31.8	11.0	18.4	15.4	22.6	7.1	..	24.8	..	20.7	1.4	0.2	1.9	0.8	5.1	0.816	0.706	P-75
P-76	5712	26.9	14.0	19.0	40.1	7.0	44.0	..	..	48.0	..	..	..	..	..	...	1.0	0.599	0.409	P-76
P-77	21782	43.7	25.2	11.7	19.4	13.5	59.2	7.5	..	11.0	..	6.3	1.0	..	1.3	...	0.2	0.806	0.689	P-77
P-78	7223	80.7	..	8.3	11.0	21.7	..	51.4	..	22.2	..	..	4.7	..	..	...	..	0.890	0.807	P-78
P-79	26257	87.4	3.5	0.4	8.7	60.2	..	34.2	..	..	..	..	1.0	..	0.2	...	4.4	0.913	0.909	P-79

Pulverized-Fuel-Fired Units, 200 to 299 Lb per Sq In. Gage Pressure (Continued)																			
P-80	17520	86.5	1.9	..	11.6	43.4	2.7	46.4	..	6.7	..	...	..	..	..	...	0.8	0.884	0.884	P-80
P-81	8736	73.0	17.5	..	9.5	15.5	1.4	46.4	..	30.3	..	..	6.4	..	..	...	..	0.905	0.905	P-81
P-82	8736	72.5	15.0	..	12.5	1.5	..	65.3	..	23.2	..	1.1	8.9	..	..	...	..	0.875	0.875	P-82
P-83	8736	78.2	11.6	..	10.2	14.0	..	54.5	..	27.5	..	..	..	..	4.0	...	..	0.898	0.898	P-83
P-84	8736	81.0	8.5	..	10.5	19.3	..	36.0	..	2.3	..	1.3	..	..	41.1	...	..	0.895	0.895	P-84
P-85	26257	90.3	2.2	..	7.5	36.0	3.6	40.0	..	17.3	..	..	..	..	1.4	...	1.7	0.925	0.925	P-85
P-86	17520	85.3	8.6	..	6.1	6.7	..	69.6	..	23.7	..	..	..	..	..	...	..	0.939	0.939	P-86
P-87	17520	86.0	6.0	..	8.0	24.1	2.7	69.6	..	2.5	..	..	..	..	..	...	1.1	0.920	0.920	P-87
Oil- and Gas-Fired Units																			
OG-1	10829	70.4	8.8	4.8	16.0	21.3	28.6	13.0	..	23.5	..	..	3.5	1.3	4.8	...	4.0	0.840	0.792	OG-1
OG-2	25473	59.4	18.5	10.6	11.5	38.8	16.6	5.4	..	22.1	..	..	6.9	..	1.0	...	9.2	0.885	0.779	OG-2
OG-3	8232	86.0	4.1	4.8	5.1	29.7	..	62.5	..	2.2	..	..	..	..	5.6	...	..	0.949	0.901	OG-3
OG-4	18896	58.0	19.3	13.5	9.2	47.9	..	42.0	..	6.7	..	..	3.4	..	..	...	..	0.908	0.773	OG-4
OG-5	8232	62.1	12.3	14.0	11.6	62.7	..	23.5	..	11.9	..	..	1.9	..	..	...	..	0.884	0.744	OG-5
OG-6	18965	51.8	19.5	17.1	11.6	37.5	19.1	13.9	..	21.4	..	..	2.3	..	0.4	...	5.4	0.884	0.713	OG-6
OG-7	8736	64.0	0.5	16.1	19.4	39.7	..	36.5	..	21.8	..	..	..	..	..	...	2.0	0.806	0.645	OG-7
OG-8	18884	36.6	19.4	31.5	12.5	24.5	27.9	14.0	..	12.5	..	..	19.4	..	0.7	0.2	0.8	0.875	0.560	OG-8
OG-9	8736	57.5	0.5	20.2	21.8	16.5	..	78.2	..	2.3	..	..	0.5	..	..	...	2.5	0.782	0.580	OG-9
OG-10	8736	87.6	0.5	3.5	8.4	49.3	..	50.7	..	..	..	..	..	..	..	...	..	0.916	0.881	OG-10
OG-11	8568	90.6	0.3	6.7	2.4	98.0	..	..	..	..	..	..	..	..	..	...	2.0	0.976	0.909	OG-11
OG-12	13100	55.9	..	25.5	18.6	..	0.1	61.1	..	29.1	..	1.0	0.1	..	1.4	...	7.2	0.841	0.559	OG-12
OG-13	26209	51.0	31.8	9.2	8.0	57.2	5.2	15.3	..	18.6	..	..	..	..	3.7	...	..	0.920	0.828	OG-13
OG-14	26208	49.8	25.1	16.7	8.4	60.6	11.0	7.1	..	10.3	..	..	..	..	2.4	...	8.6	0.916	0.749	OG-14
OG-15	26210	42.9	32.5	16.1	8.5	56.8	0.4	16.7	..	21.5	..	..	..	..	3.5	...	1.1	0.915	0.754	OG-15
OG-16	17468	31.5	0.9	26.1	41.5	7.4	0.7	47.7	..	29.2	..	1.2	..	..	0.3	...	13.5	0.585	0.324	OG-16
OG-17	13100	49.1	..	31.0	19.9	..	..	64.6	..	30.2	..	0.7	1.5	..	0.4	...	2.6	0.801	0.491	OG-17
OG-18	18100	45.0	3.4	27.2	24.4	..	13.2	42.8	..	18.0	..	0.6	5.4	..	1.2	...	18.8	0.756	0.484	OG-18
OG-19	20828	58.7	0.3	21.2	19.8	22.2	0.1	38.1	..	31.7	..	..	1.4	..	1.1	...	5.4	0.802	0.590	OG-19
OG-20	10152	73.8	15.0	0.2	11.0	72.4	..	13.0	..	14.1	..	..	..	..	..	...	0.5	0.890	0.888	OG-20
OG-21	10251	70.0	17.6	0.6	11.8	48.5	..	6.0	..	6.9	..	..	..	..	..	...	38.6	0.882	0.876	OG-21
OG-22	26009	48.7	31.2	14.0	6.1	67.4	2.9	9.3	..	7.8	..	..	..	..	7.5	...	5.1	0.939	0.799	OG-22
OG-23	18140	35.0	0.8	38.1	26.1	12.9	..	63.0	..	16.6	..	0.3	0.7	..	0.7	0.1	6.4	0.739	0.358	OG-23
OG-24	25537	74.3	20.0	2.4	3.3	43.5	..	33.1	..	23.4	..	..	..	..	..	...	..	0.967	0.943	OG-24
Miscellaneous Combinations																			
M-1	21000	84.6	5.1	0.4	9.9	66.8	0.9	9.8	..	11.3	..	..	1.6	..	5.6	...	4.0	0.901	0.897	M-1
M-2	21638	89.7	3.0	0.1	7.2	70.7	..	15.6	..	2.5	5.6	..	1.6	..	2.1	...	1.9	0.928	0.927	M-2
M-3	20991	83.6	3.1	..	13.3	57.6	..	13.6	..	23.7	0.5	..	2.2	..	1.5	...	0.9	0.867	0.867	M-3
M-4	25534	48.4	35.5	12.4	3.7	32.8	4.1	17.7	..	13.7	..	..	6.1	..	..	...	25.6	0.963	0.839	M-4
M-5	25530	54.7	16.9	21.9	6.5	12.5	40.0	7.1	..	33.3	..	..	2.8	..	0.7	...	3.6	0.935	0.716	M-5
M-6	25467	13.3	57.2	23.0	6.5	6.6	..	6.4	..	29.2	57.8	..	..	..	..	...	..	0.935	0.705	M-6
Reheat Stoker-Fired Units, 500 to 599 Lb per Sq In. Gage Pressure																			
SR-1	24863	63.8	0.1	4.3	31.8	8.8	..	7.3	25.5	20.9	0.4	..	1.0	..	..	...	36.1	0.682	0.639	SR-1
SR-2	24587	61.0	0.1	3.9	35.0	9.3	..	8.0	24.1	18.3	0.3	..	1.5	..	..	...	38.5	0.650	0.611	SR-2
SR-3	17462	66.2	0.2	4.2	29.4	16.0	4.4	13.1	29.7	8.2	4.9	..	1.9	..	..	...	21.8	0.706	0.664	SR-3
SR-4	17509	71.0	0.3	3.8	24.9	19.1	2.0	12.4	24.3	10.0	5.9	..	0.9	..	..	...	25.4	0.751	0.713	SR-4
SR-5	17474	81.0	0.3	6.5	12.2	29.3	0.1	20.9	..	23.3	6.6	..	7.9	0.8	1.0	...	10.1	0.878	0.813	SR-5
SR-6	17471	80.3	0.5	5.2	14.0	11.6	0.7	31.2	..	24.0	11.2	..	5.4	1.2	0.5	...	14.2	0.860	0.808	SR-6
SR-7	24854	65.3	6.9	6.9	20.9	39.7	0.7	14.3	..	24.7	5.2	..	11.8	0.2	0.7	...	2.7	0.791	0.722	SR-7
SR-8	24837	63.3	6.3	6.6	23.8	31.7	0.9	6.7	..	19.6	9.8	..	23.2	0.8	4.8	...	2.5	0.762	0.696	SR-8
Reheat Pulverized-Fuel-Fired Units, 1390 Lb per Sq In. Gage Pressure																			
PR-1	8736	86.0	0.6	1.6	11.8	33.7	18.4	11.9	33.3	2.7	..	..	..	..	..	...	..	0.882	0.866	PR-1
Reheat Pulverized-Fuel-Fired Units, 500 to 700 Lb per Sq In. Gage Pressure																			
PR-2	17492	73.0	0.4	5.3	21.3	12.0	9.5	34.3	2.2	17.1	..	1.5	17.2	..	6.1	...	0.1	0.787	0.734	PR-2

TABLE 3 INDIVIDUAL HEAT-TRANSFER RATES OF STEAM-GENERATING UNITS

Unit No.	Length of period reported, years	Total water evaporated, 1000 lb	Total steaming hours	Average evaporation, 1000 lb per hr	Average evaporation per unit of: Saturated steam surface, lb per sq ft per hr	Furnace volume, lb per cu ft per hr	Projected grate area, lb per sq ft per hr	Average heat output of boiler proper: In saturated steam, kb per sq ft sat. surf. per hr	In superheated steam, kb per sq ft sat. surf. per hr	Average heat output of entire unit represented by: Saturated steam surface, kb per sq ft per hr	Superheated steam: Saturated steam surface, kb per sq ft per hr	Superheated steam: Furnace volume, kb per cu ft per hr	Superheated steam: Projected grate area, kb per sq ft per hr	Unit No.
Stoker-Fired Units, 500 to 700 Lb per Sq In. Gage Pressure														
S-1	3	1,008,546	9,318	108	7.7	24.6	296	7.1	8.3	7.9	9.2	29.1	350	S-1
S-2	2	2,191,331	12,538	175	7.5	21.2	331	6.8	7.9	7.8	8.9	25.1	391	S-2
S-3	3	1,151,147	11,128	103	7.4	20.8	290	6.7	7.9	7.6	8.7	24.6	343	S-3
S-4	3	1,115,398	10,840	103	7.3	20.8	290	6.8	7.9	7.5	8.6	24.6	343	S-4
S-5	3	1,160,793	11,355	102	7.3	20.6	286	6.8	7.9	7.5	8.6	24.4	338	S-5
S-6	2	2,271,157	13,431	169	7.3	20.5	320	6.6	7.7	7.5	8.6	24.2	378	S-6
S-7	2	2,134,393	12,906	165	7.1	20.0	312	6.4	7.5	7.3	8.4	23.6	369	S-7
S-8	2	2,023,716	12,360	164	7.1	19.9	310	6.4	7.5	7.3	8.4	23.5	367	S-8
S-9	3	1,392,139	14,741	94	6.7	21.4	258	6.3	7.3	6.9	7.9	25.3	305	S-9
S-10	3	1,382,752	14,780	94	6.7	21.4	258	6.1	7.3	6.9	7.9	25.3	305	S-10
S-11	3	1,702,474	15,511	110	6.6	18.3	248	5.6	6.7	6.2	7.4	20.4	276	S-11
S-12	3	1,085,014	13,068	83	5.9	14.4	234	5.3	6.4	6.1	7.0	17.1	277	S-12
S-13	3	1,816,804	15,435	117	6.3	20.3	264	5.4	6.4	5.9	7.0	24.0	293	S-13
S-14	3	1,087,078	13,655	80	5.7	13.9	225	5.2	6.2	5.9	6.7	16.4	266	S-14
S-15	2	1,279,189	12,637	101	5.6	13.0	191	5.5	6.4	5.9	6.7	15.5	228	S-15
S-16	2	1,277,890	12,795	100	5.6	11.8	190	5.6	6.4	5.9	6.7	14.1	226	S-16
S-17	3	1,207,696	15,270	79	5.6	13.7	222	5.2	6.1	5.8	6.6	16.2	262	S-17
S-18	3	1,033,193	13,654	76	5.4	13.2	214	5.0	5.3	5.6	6.4	15.6	253	S-18
S-19	2	1,220,397	12,633	97	5.4	12.4	184	5.3	6.1	5.7	6.4	14.8	219	S-19
S-20	3	1,075,057	14,552	74	5.3	12.8	208	4.9	5.3	5.5	6.3	15.1	246	S-20
S-21	3	1,094,097	14,805	74	5.3	12.8	208	4.9	5.7	5.5	6.3	15.1	246	S-21
S-22	2	1,189,541	12,528	95	5.3	12.2	180	5.2	6.0	5.5	6.3	14.5	214	S-22
S-23	2	1,184,372	12,654	94	5.2	12.0	178	5.1	5.9	5.4	6.2	14.3	212	S-23
S-24	2	1,152,372	12,494	92	5.1	11.8	174	5.1	5.8	5.3	6.1	14.1	207	S-24
Stoker-Fired Units, 400 to 499 Lb per Sq In. Gage Pressure														
S-25	2	803,589	10,398	77	7.4	14.3	284	6.8	7.9	7.6	7.6	14.6	291	S-25
S-26	3	1,766,631	17,107	103	6.3	13.2	306	5.9	6.9	6.5	7.5	15.8	366	S-26
S-27	2	687,594	9,306	74	7.1	13.7	274	6.7	7.5	7.3	7.3	14.1	281	S-27
S-28	2	677,622	9,349	72	6.9	13.3	266	6.4	7.3	7.1	7.1	13.6	273	S-28
S-29	2	659,980	9,108	72	6.9	13.3	266	6.4	7.3	7.1	7.1	13.6	273	S-29
S-30	1½	2,103,427	10,747	196	6.9	22.0	374	5.8	6.9	5.8	6.9	22.0	374	S-30
S-31	1½	2,028,428	10,463	194	6.8	21.8	370	5.7	6.8	5.7	6.8	21.8	370	S-31
S-32	3	2,383,661	13,762	174	6.1	20.4	348	5.5	6.5	5.5	6.8	21.9	374	S-32
S-33	1½	2,016,071	10,904	185	6.5	20.8	353	5.5	6.5	5.5	6.5	20.8	353	S-33
S-34	1½	2,050,464	11,013	186	6.5	20.9	355	5.5	6.5	5.5	6.5	20.9	355	S-34
S-35	3	2,557,724	15,290	167	5.9	19.6	334	5.4	6.4	5.4	6.4	21.2	361	S-35
S-36	3	2,054,550	12,954	155	5.5	18.2	310	5.0	5.9	5.0	5.9	19.6	334	S-36
Stoker-Fired Units, 300 to 399 Lb per Sq In. Gage Pressure														
S-37	2	2,101,536	16,063	131	9.5	12.6	240	8.3	9.8	9.8	11.3	14.9	284	S-37
S-38	1½	657,095	9,120	72	4.8	10.4	257	4.8	5.3	4.8	10.4	11.4	282	S-38
S-39	1	967,330	6,909	140	8.6	20.8	545	7.7	9.1	8.6	10.0	24.2	633	S-39
S-40	1	967,330	6,909	140	8.6	20.8	547	7.7	9.1	8.6	10.0	24.2	635	S-40
S-41	1	967,330	6,909	140	8.6	20.8	547	7.7	9.1	8.6	10.0	24.2	635	S-41
S-42	1	967,330	6,909	140	8.6	20.8	547	7.7	9.1	8.6	10.0	24.2	635	S-42
S-43	1	967,330	6,909	140	8.6	20.8	547	7.7	9.1	8.6	10.0	24.2	635	S-43
S-44	1	967,330	6,909	140	8.6	20.8	547	7.7	9.1	8.6	10.0	24.2	635	S-44
S-45	1	977,244	6,882	140	8.4	20.8	545	7.5	8.9	8.4	9.8	24.2	633	S-45
S-46	1	571,807	4,871	120	8.3	29.6	324	7.3	8.3	8.2	9.2	32.8	358	S-46
S-47	2	404,723	4,666	87	7.8	16.7	268	7.2	8.4	8.0	9.2	19.7	316	S-47
S-48	2	359,744	4,312	83	7.4	16.0	256	6.8	7.9	7.6	8.7	18.8	302	S-48
S-49	2	721,363	8,301	87	6.6	15.8	226	6.3	7.3	6.8	7.8	18.7	267	S-49
S-50	3	1,401,722	17,456	80	6.2	11.9	222	5.7	6.8	6.4	7.4	14.3	266	S-50
S-51	3	1,363,346	17,387	79	6.2	11.5	219	5.7	6.8	6.4	7.4	13.8	262	S-51
S-52	3	1,345,038	17,162	78	6.1	11.5	216	5.6	6.7	6.3	7.3	13.8	259	S-52
S-53	2½	925,981	9,787	95	6.3	19.8	232	5.7	6.5	7.1	7.3	22.9	269	S-53
S-54	2	581,195	7,204	81	6.2	14.7	210	5.9	6.9	6.4	7.3	17.4	248	S-54
S-55	3	1,779,184	21,142	84	6.1	15.5	460	5.6	6.5	6.3	7.2	18.4	545	S-55
S-56	2	651,118	8,107	80	6.1	14.5	208	5.8	6.8	6.3	7.2	17.2	246	S-56
S-57	1	725,741	6,233	118	6.0	...	...	5.4	6.4	6.1	7.0	...	...	S-57
S-58	3	1,910,479	23,304	82	5.9	15.2	450	5.4	6.4	6.0	7.0	18.0	534	S-58
S-59	3	2,378,490	17,399	137	5.8	15.3	348	5.3	6.3	8.0	9.2	19.7	316	S-59
S-60	3	2,580,725	18,813	138	5.9	14.5	400	5.4	6.4	5.4	6.4	15.7	433	S-60
S-61	3	1,652,083	22,620	73	5.3	17.5	400	4.8	5.8	5.4	6.3	21.0	470	S-61
S-62	3	2,363,870	23,941	98	5.2	12.0	198	4.7	5.7	5.3	6.2	14.4	238	S-62
S-63	3	2,743,869	20,730	133	5.7	14.0	385	5.2	6.2	5.2	6.2	15.2	417	S-63
S-64	1	244,926	3,666	67	5.4	17.6	279	5.3	5.9	5.3	6.0	19.5	308	S-64
S-65	3	2,676,228	20,731	129	5.5	13.2	375	5.0	6.0	5.0	6.0	14.3	407	S-65
S-66	3	2,500,852	19,663	127	5.4	14.2	332	5.0	5.9	5.0	5.9	15.4	349	S-66
S-67	3	2,573,530	20,169	128	5.4	14.3	325	5.0	5.9	5.0	5.9	15.5	352	S-67
S-68	3	2,587,639	20,324	127	5.4	14.2	322	5.0	5.9	4.9	5.9	15.3	348	S-68
S-69	3	2,411,107	19,359	125	5.3	13.1	362	4.9	5.8	4.8	5.8	14.2	392	S-69
S-70	3	2,710,125	20,891	130	5.5	13.7	378	4.9	5.8	4.9	5.8	14.5	400	S-70
S-71	3	2,493,040	20,041	125	5.3	14.0	318	4.9	5.8	4.9	5.8	15.2	345	S-71
S-72	2	854,218	12,338	92	4.9	18.0	372	4.9	5.7	4.9	5.7	20.8	431	S-72
S-73	½	203,888	2,330	86	5.0	21.4	425	5.0	5.7	5.0	5.7	24.4	484	S-73
S-74	1½	698,070	9,243	76	5.1	11.0	271	5.1	5.6	5.1	5.6	12.1	299	S-74
S-75	3	392,153	6,948	57	4.8	9.0	210	4.8	5.6	4.9	5.6	10.6	249	S-75
S-76	3	2,498,899	20,320	123	5.2	12.9	357	4.8	5.6	4.8	5.6	13.9	386	S-76
S-77	3	2,358,824	19,658	120	5.1	13.4	304	4.7	5.5	4.7	5.5	14.5	330	S-77
S-78	3	354,677	6,498	55	4.6	8.6	181	4.6	5.3	4.7	5.5	10.2	214	S-78
S-79	1½	682,981	9,836	70	4.7	10.1	250	4.7	5.3	4.7	5.3	11.4	282	S-79
S-80	1	394,099	5,371	73	4.2	15.6	151	4.1	4.8	4.1	4.8	17.8	172	S-80
S-81	1	434,812	6,170	71	4.1	15.2	147	4.0	4.7	4.0	4.7	15.8	168	S-81
S-82	3	392,228	8,816	45	3.8	7.1	165	3.8	4.4	3.9	4.5	8.4	195	S-82
S-83	2	465,785	12,660	37	3.7	12.8	207	3.6	4.1	3.6	4.1	14.1	228	S-83
S-84	3	1,149,334	17,358	66	3.5	12.4	256	3.5	4.1	3.5	4.1	14.4	296	S-84
S-85	2	826,806	12,556	66	3.5	12.9	268	3.5	4.1	3.5	4.1	15.0	310	S-85
S-86	2	461,000	12,799	36	3.6	12.5	201	3.5	4.0	3.5	4.0	13.8	221	S-86
S-87	2	471,108	13,345	36	3.6	12.5	201	3.5	3.9	3.5	3.9	13.8	222	S-87
S-88	2	454,116	12,835	34	3.4	11.8	190	3.3	3.8	3.3	3.8	13.0	210	S-88
S-89	3	1,066,060	18,177	59	3.2	11.6	230	3.2	3.7	3.2	3.7	13.5	267	S-89

TABLE 3 INDIVIDUAL HEAT-TRANSFER RATES OF STEAM-GENERATING UNITS (*Continued*)

Unit No.	Length of period reported, years	Total water evaporated, 1000 lb	Total steaming hours	Average evaporation, 1000 lb per hr	Average evaporation per unit of: Saturated steam surface, lb per sq ft per hr	Furnace volume, lb per cu ft per hr	Projected grate area, lb per sq ft per hr	Average heat output of boiler proper: In saturated steam, kb per sq ft sat. surf. per hr	In superheated steam, kb per sq ft sat. surf. per hr	Average heat output of entire unit represented by: Saturated steam surface, kb per sq ft per hr	Superheated steam: Saturated steam surface, kb per sq ft per hr	Superheated steam: Furnace volume, kb per cu ft per hr	Superheated steam: Projected grate area, kb per sq ft per hr	Unit No.
Stoker-Fired Units, 200 to 299 Lb per Sq In. Gage Pressure														
S-90	2½	573,822	5,777	99	7.9	23.2	...	7.4	8.9	8.2	9.7	28.4	...	S-90
S-91	2½	442,686	4,625	96	7.7	22.6	...	7.3	8.8	7.9	9.4	27.6	...	S-91
S-92	2½	486,015	5,646	86	6.9	20.2	...	6.5	7.8	7.1	8.5	24.7	...	S-92
S-93	½	222,214	2,575	86	6.9	20.2	...	6.4	7.7	7.1	8.4	24.7	...	S-93
S-94	2	1,024,108	13,048	79	6.9	16.1	254	6.6	7.5	7.3	8.2	19.1	301	S-94
S-95	2	1,054,930	13,322	79	6.9	16.1	254	6.6	7.5	7.3	8.2	19.1	301	S-95
S-96	½	238,282	2,888	83	6.6	19.5	...	6.1	7.4	6.8	8.1	23.8	...	S-96
S-97	1	454,200	4,512	101	7.9	21.3	316	6.9	8.0	6.9	8.0	21.6	320	S-97
S-98	3	676,625	7,189	94	6.8	22.6	330	6.8	7.6	7.3	8.0	26.6	388	S-98
S-99	1	445,921	4,587	97	7.6	20.3	288	6.6	7.6	6.6	7.6	20.4	289	S-99
S-100	½	138,656	1,891	74	6.1	17.4	...	5.7	6.9	6.3	7.5	21.2	...	S-100
S-101	2	863,657	11,917	72	6.2	14.7	232	5.9	6.8	6.5	7.4	17.5	276	S-101
S-102	2	922,561	13,600	68	5.9	13.9	219	5.6	6.5	6.2	7.0	16.5	260	S-102
S-103	1½	707,878	10,728	73	7.1	13.5	330	6.3	6.4	6.9	7.0	13.3	325	S-103
S-104	1½	1,005,702	13,765	73	5.8	11.2	193	5.3	6.1	5.9	6.6	12.8	221	S-104
S-105	1½	720,179	10,824	67	6.5	12.4	303	5.8	5.9	6.3	6.4	12.2	299	S-105
S-106	3	1,112,302	16,463	68	5.7	18.3	320	5.6	6.3	5.6	6.3	20.2	354	S-106
S-107	3	855,693	10,958	78	5.3	11.7	252	5.6	6.2	5.6	6.2	13.7	295	S-107
S-108	1	542,145	7,214	75	5.2	24.8	226	5.5	6.2	5.5	6.2	29.4	264	S-108
S-109	3	1,056,883	16,369	65	5.5	17.5	307	5.4	6.1	5.4	6.1	19.3	339	S-109
S-110	3	1,102,319	17,369	63	5.3	17.0	297	5.2	5.9	5.2	5.9	18.8	328	S-110
S-111	3	902,599	12,444	73	4.9	11.0	236	5.2	5.8	5.2	5.8	12.9	276	S-111
S-112	1½	862,367	8,882	98	5.2	12.2	207	5.3	5.8	5.3	5.8	13.6	232	S-112
S-113	3	1,009,585	16,205	62	5.2	16.7	292	5.1	5.7	5.1	5.7	18.5	323	S-113
S-114	3½	2,231,191	16,846	133	4.7	12.3	353	4.4	5.2	4.8	5.6	14.6	418	S-114
S-115	3	818,607	12,646	65	4.6	9.0	210	4.4	5.0	4.9	5.5	10.7	249	S-115
S-116	3	1,113,126	18,764	57	4.7	13.3	224	4.7	5.2	5.0	5.5	15.6	264	S-116
S-117	3½	2,275,265	17,252	132	4.5	8.8	278	4.3	5.0	4.6	5.3	10.4	330	S-117
S-118	1½	718,280	5,571	129	4.4	8.6	272	4.0	4.7	4.5	5.2	10.2	322	S-118
S-119	3½	1,324,887	10,554	126	4.3	8.4	266	4.1	4.8	4.4	5.1	9.9	315	S-119
S-120	1½	775,823	6,156	126	4.3	8.4	266	3.9	4.6	4.4	5.1	9.9	315	S-120
S-121	3½	2,043,859	17,304	118	4.2	10.9	312	4.0	4.7	4.3	5.0	12.9	370	S-121
S-122	1½	641,575	5,466	117	4.0	7.8	236	3.6	4.3	4.1	4.7	9.3	280	S-122
S-123	3	1,106,038	16,744	66	4.6	8.9	213	4.4	4.9	4.9	5.4	10.7	249	S-123
Pulverized-Fuel-Fired Units, 500 to 700 Lb per Sq In. Gage Pressure														
P-1	3	2,067,385	15,349	135	8.7	10.4	...	7.6	9.0	8.4	9.9	11.8	...	P-1
Pulverized-Fuel-Fired Units, 400 to 499 Lb per Sq In. Gage Pressure														
P-2	1	721,770	5,983	121	8.7	8.5	...	8.0	9.4	8.9	10.2	10.0	...	P-2
P-3	1	595,710	5,067	117	8.4	8.2	...	7.7	9.0	8.5	9.9	9.6	...	P-3
P-4	2	1,787,810	11,206	160	8.5	11.2	...	7.8	9.3	7.8	9.3	12.3	...	P-4
P-5	1	592,690	5,353	110	7.9	7.7	...	7.3	8.5	7.8	9.3	9.1	...	P-5
P-6	1	571,985	5,345	107	7.7	7.5	...	7.1	8.3	7.9	9.0	8.8	...	P-6
P-7	2	1,884,527	12,299	153	8.1	10.7	...	7.4	8.9	8.1	8.9	11.7	...	P-7
P-8	1	3,596,099	6,781	530	7.8	13.9	...	7.1	8.3	7.1	8.3	14.9	...	P-8
P-9	1	3,926,120	7,385	533	7.8	13.9	...	7.1	8.3	7.1	8.3	14.9	...	P-9
P-10	1	3,306,058	6,248	530	7.8	13.9	...	7.1	8.3	7.1	8.3	14.9	...	P-10
P-11	1	1,190,890	6,680	178	7.4	9.9	...	6.8	7.9	6.8	7.9	10.6	...	P-11
P-12	2	1,342,995	9,226	146	7.7	10.2	...	7.1	7.8	7.1	7.8	11.2	...	P-12
P-13	1	1,191,991	8,761	136	7.2	8.6	...	6.6	7.7	6.6	7.7	9.2	...	P-13
P-14	1	1,138,259	8,502	133	7.0	8.4	...	6.4	7.5	6.4	7.5	9.0	...	P-14
P-15	3	2,532,818	13,745	189	6.3	7.5	...	5.7	6.7	6.4	7.4	8.0	...	P-15
P-16	1	924,873	5,564	166	6.9	9.2	...	6.3	7.4	6.3	7.4	9.9	...	P-16
P-17	1	1,125,499	8,709	129	6.8	8.1	...	6.2	7.3	6.2	7.3	8.6	...	P-17
P-18	1	1,097,860	8,533	129	6.8	8.1	...	6.2	7.3	6.2	7.3	8.7	..	P-18
P-19	3	2,548,391	13,695	186	6.1	7.4	...	5.5	6.5	6.2	7.2	8.7	...	P-19
P-20	3	2,551,691	13,884	184	6.1	7.3	...	5.5	6.5	6.2	7.2	8.6	...	P-20
P-21	1	1,059,872	8,397	126	6.7	7.9	...	6.1	7.2	6.1	7.2	8.5	...	P-21
P-22	1	1,077,539	8,482	127	6.7	8.0	...	6.1	7.2	6.1	7.2	8.4	...	P-22
P-23	1	265,266	3,933	68	6.1	7.5	...	6.0	7.0	6.0	7.0	8.6	...	P-23
P-24	3	2,613,469	14,581	179	5.9	7.1	...	5.4	6.3	6.0	6.9	8.3	...	P-24
P-25	2	806,304	7,925	102	6.0	12.4	...	5.6	6.6	5.6	6.6	13.6	...	P-25
P-26	2	817,310	8,325	98	5.7	11.9	...	5.4	6.3	5.4	6.3	13.0	...	P-26
P-27	3	2,510,699	20,022	125	5.9	6.7	...	5.3	6.3	5.3	6.3	7.1	...	P-27
P-28	3	2,500,171	19,998	125	5.9	6.7	...	5.3	6.3	5.3	6.3	7.1	...	P-28
P-29	3	2,521,619	20,002	126	5.9	6.8	...	5.3	6.3	5.3	6.3	7.2	...	P-29
P-30	3	2,295,699	18,821	122	5.7	6.6	...	5.2	6.0	5.2	6.0	7.0	...	P-30
P-31	2	889,289	9,538	93	5.4	11.3	...	5.1	5.9	5.1	5.9	12.4	...	P-31
Pulverized-Fuel-Fired Units, 300 to 399 Lb per Sq In. Gage Pressure														
P-32	2	797,610	5,780	138	16.4	17.2	...	14.9	17.4	16.9	18.8	12.8	...	P-32
P-33	2	585,524	6,154	95	11.1	13.0	...	10.1	11.8	11.4	12.7	21.8	...	P-33
P-34	1	606,774	5,837	104	9.5	8.1	...	9.1	10.5	9.6	11.1	9.5	...	P-34
P-35	1	624,663	6,059	100	9.2	7.8	...	8.7	10.2	9.3	10.7	9.1	...	P-35
P-36	3	1,081,869	12,696	85	7.8	6.6	...	7.1	8.3	7.9	9.1	7.7	...	P-36
P-37	3	1,082,817	13,047	83	7.6	6.5	...	7.0	8.1	7.7	8.9	7.6	...	P-37
P-38	3	668,051	8,204	82	7.5	6.4	...	6.9	8.0	7.6	8.8	7.5	...	P-38
P-39	3	736,136	8,952	82	7.5	6.4	...	6.9	8.0	7.6	8.8	7.5	...	P-39
P-40	3	600,404	7,507	80	7.4	6.3	...	6.8	7.9	7.5	8.7	7.4	...	P-40
P-41	1	684,362	6,106	112	7.3	10.2	...	6.8	7.8	7.5	8.5	11.8	...	P-41
P-42	2½	1,256,544	8,860	142	7.5	10.8	...	7.3	8.0	7.3	8.5	12.2	...	P-42
P-43	3	924,345	11,729	79	7.3	6.1	...	6.7	7.8	7.4	8.5	7.1	...	P-43
P-44	1	780,807	6,003	130	7.2	11.8	...	7.4	8.4	7.4	8.4	13.7	...	P-44
P-45	2	701,873	8,950	79	7.1	9.3	...	7.0	7.9	7.4	8.3	10.9	...	P-45
P-46	2½	1,166,060	8,461	138	7.3	10.4	...	7.0	7.8	7.0	8.2	11.7	...	P-46
P-47	1	686,044	6,554	105	6.9	9.5	...	6.4	7.3	7.1	8.0	11.0	...	P-47
P-48	1	681,873	6,564	104	6.8	9.5	...	6.3	7.2	7.0	7.9	11.0	...	P-48
P-49	1	698,676	6,746	104	6.8	9.5	...	6.3	7.2	7.0	7.9	11.0	...	P-49
P-50	1	695,934	6,661	104	6.8	9.5	...	6.3	7.2	7.0	7.9	11.0	...	P-50
P-51	1	894,422	5,421	165	7.3	12.2	...	6.7	7.8	6.7	7.8	13.1	...	P-51
P-52	2	854,301	11,395	75	6.7	9.3	...	6.7	7.5	7.0	7.8	10.9	...	P-52

TABLE 3 INDIVIDUAL HEAT-TRANSFER RATES OF STEAM-GENERATING UNITS (*Continued*)

Unit No.	Length of period reported, years	Total water evaporated, 1000 lb	Total steaming hours	Average evaporation, 1000 lb per hr	Average evaporation per unit of: Saturated steam surface, lb per sq ft per hr	Furnace volume, lb per cu ft per hr	Projected grate area, lb per sq ft per hr	Average heat output of boiler proper: In saturated steam, kb per sq ft sat. surf. per hr	In superheated steam, kb per sq ft sat. surf. per hr	Average heat output of entire unit represented by: Saturated steam surface, kb per sq ft per hr	Superheated steam: Saturated steam surface, kb per sq ft per hr	Superheated steam: Furnace volume, kb per cu ft per hr	Superheated steam: Projected grate area, kb per sq ft per hr	Unit No.
Pulverized-Fuel-Fired Units, 300 to 399 Lb per Sq In. Gage Pressure (Continued)														
P-53	1	979,462	6,098	160	7.1	11.9	...	6.5	7.6	6.5	7.6	12.7	...	P-53
P-54	1	822,484	5,152	159	7.1	11.8	...	6.5	7.6	6.5	7.6	12.6	...	P-54
P-55	1	714,880	4,545	157	7.0	11.7	...	6.4	7.5	6.4	7.5	12.5	...	P-55
P-56	3	4,741,800	21,971	216	7.1	14.6	...	6.4	7.5	6.4	7.5	15.4	...	P-56
P-57	2	740,666	10,418	71	6.4	8.8	...	6.4	7.1	6.7	7.5	10.3	...	P-57
P-58	2	727,458	10,295	71	6.4	8.8	...	6.4	7.1	6.7	7.5	10.3	...	P-58
P-59	2	2,645,800	12,270	216	7.1	14.6	...	6.5	7.5	6.5	7.5	15.4	...	P-59
P-60	3	1,463,645	14,683	100	6.8	8.7	...	6.9	7.4	6.9	7.4	9.5	...	P-60
P-61	3	1,390,408	14,177	98	6.6	8.5	...	6.7	7.2	6.7	7.2	9.2	...	P-61
P-62	2	2,863,600	13,739	208	6.8	14.8	...	6.2	7.2	6.2	7.2	15.6	...	P-62
P-63	3	1,335,964	13,863	96	6.5	8.3	...	6.6	7.1	6.6	7.1	9.0	...	P-63
P-64	2½	1,053,169	12,669	83	5.9	15.7	...	5.4	6.3	5.7	6.6	17.7	...	P-64
P-65	2½	987,424	12,585	79	5.7	15.0	...	5.2	6.1	5.5	6.4	16.9	...	P-65
P-66	1	588,174	3,483	169	5.4	5.5	...	5.0	5.9	5.5	6.4	6.5	...	P-66
P-67	1	620,970	6,084	102	5.6	8.5	...	5.1	5.7	5.7	6.3	9.5	...	P-67
P-68	1	847,005	5,094	166	5.3	5.4	...	4.9	5.8	5.4	6.3	6.4	...	P-68
P-69	1	682,086	4,282	159	5.1	5.2	...	4.7	5.5	5.2	6.0	6.2	...	P-69
P-70	1	987,977	6,297	157	5.0	5.1	...	4.6	5.4	5.1	5.9	6.1	...	P-70
P-71	3	1,296,525	16,645	78	5.4	8.1	...	5.4	5.9	5.4	5.9	8.9	...	P-71
P-72	1	799,668	5,243	153	4.9	5.0	...	4.5	5.3	5.0	5.8	5.9	...	P-72
P-73	1	707,221	4,616	152	4.8	5.0	...	4.4	5.2	4.9	5.8	5.9	...	P-73
P-74	3	1,161,632	16,649	70	4.9	7.3	...	4.9	5.4	4.9	5.4	8.1	...	P-74
Pulverized-Fuel-Fired Units, 200 to 299 Lb per Sq In. Gage Pressure														
P-75	2½	1,325,222	7,600	174	11.1	14.9	...	10.4	12.5	11.4	13.5	18.1	...	P-75
P-76	1	213,651	1,536	141	9.2	12.8	...	8.6	10.4	9.5	11.2	15.6	...	P-76
P-77	2½	1,239,244	9,523	130	8.9	12.7	...	8.3	10.0	9.1	10.8	15.4	...	P-77
P-78	1	674,626	5,829	116	8.1	10.9	...	7.6	8.7	7.6	8.7	11.8	...	P-78
P-79	3	4,535,400	22,991	197	6.6	11.6	...	6.0	7.0	6.0	7.0	12.2	...	P-79
P-80	2	2,829,900	15,153	187	6.3	11.0	...	5.7	6.7	5.7	6.7	11.6	...	P-80
P-81	1	1,066,331	6,377	168	5.3	6.5	...	4.8	5.8	5.4	6.4	7.8	...	P-81
P-82	1	1,063,182	6,339	168	5.3	6.5	...	4.8	5.8	5.4	6.4	7.8	...	P-82
P-83	1	1,141,413	6,828	167	5.3	6.4	...	4.8	5.8	5.4	6.4	7.7	...	P-83
P-84	1	1,184,966	7,069	168	5.3	6.5	...	4.8	5.8	5.4	6.4	7.8	...	P-84
P-85	3	4,145,000	23,689	175	6.1	10.3	...	5.5	6.4	5.5	6.4	10.9	...	P-85
P-86	2	2,635,000	14,960	176	5.9	10.3	...	5.4	6.2	5.4	6.2	10.9	...	P-86
P-87	2	2,661,400	15,077	177	5.9	10.4	...	5.4	6.2	5.4	6.2	11.0	...	P-87
Oil- and Gas-Fired Units														
OG-1	1	1,130,036	7,611	149	10.1	28.0	...	9.0	10.3	9.0	10.3	28.4	...	OG-1
OG-2	3	1,994,658	15,187	132	8.9	24.8	...	8.2	9.6	8.2	9.6	26.7	...	OG-2
OG-3	1	2,239,528	7,086	316	8.5	16.0	...	7.5	9.0	7.5	9.0	16.9	...	OG-3
OG-4	2	1,548,209	10,956	141	7.5	8.5	...	6.9	8.1	6.9	8.1	9.2	...	OG-4
OG-5	1	556,517	5,119	109	6.9	13.6	...	6.2	7.4	6.4	7.5	14.9	...	OG-5
OG-6	2	1,357,150	9,844	138	6.7	8.4	...	6.2	7.3	6.2	7.3	9.1	...	OG-6
OG-7	1	586,112	5,258	111	6.6	13.5	...	5.7	7.1	5.7	7.1	13.7	...	OG-7
OG-8	2	914,115	6,910	132	6.4	8.0	...	5.9	6.9	5.9	7.0	8.7	...	OG-8
OG-9	1	561,306	5,014	112	6.5	13.7	...	5.8	6.6	5.8	6.6	13.9	...	OG-9
OG-10	1	838,839	7,660	109	6.4	13.3	...	5.7	6.5	5.7	6.5	13.5	...	OG-10
OG-11	1	819,188	7,765	105	6.1	12.8	...	5.4	6.2	5.4	6.2	13.0	...	OG-11
OG-12	1½	1,100,328	7,314	150	5.9	15.4	...	5.2	6.1	5.2	6.1	15.9	...	OG-12
OG-13	3	1,173,343	13,380	88	5.2	16.0	...	4.6	5.3	4.6	5.3	16.3	...	OG-13
OG-14	3	1,153,099	13,069	88	5.2	16.0	...	4.6	5.3	4.6	5.3	16.3	...	OG-14
OG-15	3	965,612	11,279	86	5.1	15.6	...	4.5	5.2	4.5	5.2	15.9	...	OG-15
OG-16	1	701,749	5,490	128	5.0	13.2	...	4.4	5.2	4.4	5.2	13.6	...	OG-16
OG-17	1	772,755	6,439	120	4.7	12.4	...	4.2	4.9	4.2	4.9	13.0	...	OG-17
OG-18	1½	980,405	8,103	121	4.8	12.6	...	4.2	4.9	4.2	4.9	13.0	...	OG-18
OG-19	2	1,396,592	12,049	116	4.5	12.0	...	4.0	4.7	4.0	4.7	12.4	...	OG-19
OG-20	1	1,122,847	7,479	150	4.1	10.3	...	3.8	4.4	3.8	4.4	11.2	...	OG-20
OG-21	1	1,073,605	7,184	150	4.1	10.3	...	3.8	4.4	3.8	4.4	11.2	...	OG-21
OG-22	3	841,684	12,672	67	3.9	12.2	...	3.5	4.0	3.5	4.0	12.4	...	OG-22
OG-23	1½	628,420	6,660	94	3.7	9.7	...	3.2	3.8	3.2	3.8	10.0	...	OG-23
OG-24	3	254,101	6,373	40	3.1	6.0	...	2.9	3.3	3.1	3.5	6.9	...	OG-24
Miscellaneous Combinations														
M-1	3	2,656,432	17,783	149	7.7	10.4	...	6.8	8.0	6.8	8.0	10.8	...	M-1
M-2	3	2,889,626	19,408	149	7.7	10.4	...	6.8	8.0	6.8	8.0	10.8	...	M-2
M-3	3	2,581,052	17,535	147	7.6	10.3	...	6.7	7.9	6.7	7.9	10.7	...	M-3
M-4	3	883,690	12,364	71	5.5	10.8	...	5.1	6.0	5.5	6.3	12.4	...	M-4
M-5	3	1,037,318	13,983	74	5.8	11.2	...	5.3	6.2	5.3	6.1	11.7	...	M-5
M-6	3	182,659	3,382	54	4.2	8.2	...	3.9	4.5	4.2	4.8	9.4	...	M-6

analyzed by a single group of men in the office of the authors. Particular attention was given to crippled periods or outages and their extent, cause, and frequency, with enough other information to give the authors some idea of the operating and load conditions to which the individual units were subjected. Design and descriptive data were also obtained for each unit studied.

Tables 1, 2, and 3 give these data for each unit separately. For convenience they are grouped according to method of firing and further subdivided into steam-pressure classes. They are ordered according to average rate of driving, expressed as kb (1000Btu) output of the entire unit per square foot of saturated heating surface. Each unit has a designation consisting of a letter or letters and a number. The letter classifies the unit as to method of firing or the type of fuel burned, as: S, stoker-fired; P, pulverized-coal-fired; OG, oil- and gas-fired. A fourth group consisting of units using coal and either gas or oil is designated by the letter M. The numbers indicate the position in the group in which the unit falls. In the case of reheat boilers, the letter R follows the firing-method designation and precedes the number. Each individual unit carries the same designation in every table in which it may appear.

At this point it might be well to insert a note of caution to the reader so that he may not be misled by mere figures. As in the first year of the study, some difficulty was encountered in appor-

tioning the reported crippled hours among the various causes. The operators were requested to give the primary cause of outage in each case, but many occasions arose in which more than one part of the unit was worked on at one time. For instance, the boiler might be shut down and crippled for repairs to the furnace, but the operators might very naturally take advantage of the opportunity to make extensive repairs to the boiler or stoker. These latter might not be in any way urgent, but would certainly tend to defer or even avoid future shutdown for such work. Also it is the practice of many operators to schedule repair work at definite intervals. This permits the maintenance crew to do the most work in the least elapsed time by working on several parts of the unit simultaneously. In such instances the operators were requested to give more detailed information, especially when the original log sheets showed an appreciable divergence from the average. By this means the authors have been able to correct much of the misleading data.

Table 4 is a summary of the operating records, with the units grouped according to steam pressures and firing methods and further subdivided according to the length of time represented by the operating records received. The figures under the heading "Group performances" are averages weighted according to the number of boiler-hours involved in each case. The method of weighting was described in detail in the progress report, under the same title as this paper, presented before the Power Division at the 1931 Annual Meeting. The figures for class performances were computed in a similar manner.

discussed point. This is shown for the units studied in Table 8. The decrease in availability due to preheat is still marked, being from 0.018 to 0.063, but not as great as that shown by the 1929 results, when the decrease was in all classes greater than 0.05. This would seem to indicate that the use of preheat is now better understood or that the degree of preheat has been reduced. At any rate the results are so consistent that it is reasonable to assume that they represent fact.

In order to show the degree of use of the various classes of steam-generating units, load-duration curves were constructed for each class. They are shown in Fig. 1. The percentage of maximum continuous steaming capacity, as given by the operators, is plotted against the percentage of steaming time. With

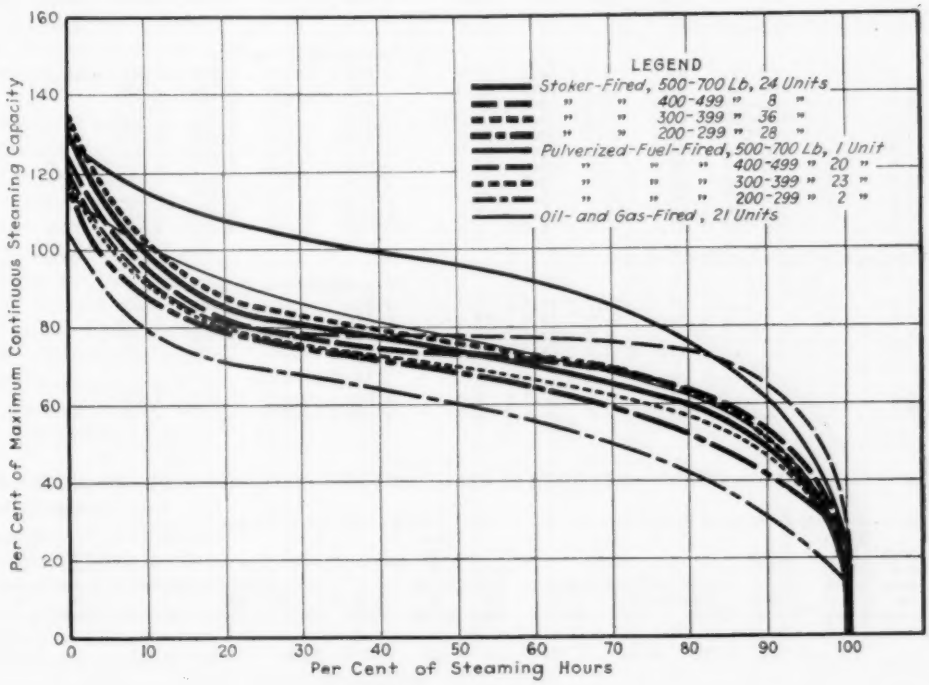


FIG. 1 LOAD-DURATION CURVES

Table 5 is reprinted from the 1931 paper (where it appeared as Fig. 2). It compares the results of the three-year study with those obtained in the first year. The 1931 paper contained a detailed discussion of the comparison, which need not be repeated here.

Further subdivision of the pulverized-coal-fired and stoker-fired groups has been made. Table 6 compares the unit and storage types of pulverized-coal firing. The availability factors first given are those taken directly from Table 2 and weighted as in Table 4. The second pair were obtained by eliminating those outages which are not affected by the method of firing and retaining only those which bear directly. At first glance it would seem that there is very little difference between the two types, but on closer inspection of the second comparison the storage type has the advantage. There is too much difference, however, in the number of boiler-hours involved to justify placing too great weight on this result. Also, the difference between high and low limits in each case is far greater than the difference between the averages.

Table 7 shows underfeed and traveling-grate stokers treated in like manner. Here again the difference may be within the accuracy of the data.

The effect of air preheat on unit availability is another much-

the exception of a group of two boilers below the average and a single unit above, the main body lies in a narrow band across the face of the chart, showing that the boilers in general were operated at approximately equal rates with respect to the standards set up for them by the designers. This in a sense bears out the conclusion arrived at after the first year's study. It was noted at that time that the apparent severity of use based on heat output per unit of heating surface had no appreciable effect on availability. Reference is made to Figs. 9 and 10 of the 1929 report. Fig. 1 of this paper indicates that boilers showing the higher unit rates of output were purposely designed for such conditions and apparently have fulfilled the intents of their designers.

The discussion up to this point has dealt with boiler outages grouped under a few necessarily large heads. Realizing that further study of these individual heads would be of considerable interest, the authors choose the three largest items charged to each unit—for instance, the stoker, furnace, and boiler or the pulverizing equipment, boiler, and economizer, as the case might be—and requested that the operators make a brief statement as to outstanding troubles and primary causes.

Table 9 is an attempt to summarize the replies and to put them in such form that comparisons may be made between pressure and

TABLE 4 SUMMARY OF OPERATING RECORDS

Period covered in years	Number of units in group	Pressure class, lb per sq in. gage	Percentage of total period: Steaming	Banked	Reserve	Crippled	Use factor	Availability factors: Individual High	Individual Low	Group	Group performances: Percentage of total period: Steaming	Banked	Reserve	Crippled	Use factor	Availability factor
Stoker-Fired Units																
3.0	14	500–700	57.3	18.1	5.9	18.7	0.754	0.846	0.774	0.813						
2.0	10	500–700	74.5	3.8	5.5	16.2	0.783	0.852	0.802	0.838	62.8	13.5	5.8	17.9	0.763	0.821
3.0	1	400–499	69.3	6.9	2.6	21.2	0.762	0.788	0.788	0.788						
2.5	3	400–499	60.4	5.0	0.6	34.0	0.654	0.713	0.622	0.660						
2.0	4	400–499	48.4	25.3	2.2	24.1	0.737	0.788	0.730	0.759						
1.5	4	400–499	83.0	7.0	0.6	9.4(?)	0.900	0.913	0.898	0.906	63.1	12.4	1.3	23.2	0.755	0.768
3.0	24	300–399	70.6	9.0	4.9	15.5	0.796	0.936	0.708	0.845						
2.0	13	300–399	58.7	9.3	15.1	16.9	0.680	0.964	0.707	0.831						
1.5	4	300–399	72.5	8.0	2.1	17.4	0.805	0.904	0.760	0.826						
1.0	13	300–399	70.9	9.6	11.9	7.6(?)	0.805	1.000	0.933	0.924	68.0	9.1	7.8	15.1	0.771	0.849
3.0	12	200–299	55.8	14.8	11.5	17.9	0.706	0.936	0.705	0.821						
2.5	3	200–299	24.8	48.8	15.1	11.3	0.736	0.899	0.875	0.887						
2.0	6	200–299	62.6	12.8	8.4	16.2	0.754	0.905	0.761	0.838						
1.5	4	200–299	72.8	3.2	6.3	17.7	0.760	0.922	0.660	0.823						
1.0	7	200–299	56.7	23.5	3.2	16.6	0.820	0.980	0.650	0.834	55.2	18.0	10.0	16.8	0.732	0.832
Performance of stoker-fired class											63.2	12.6	7.3	16.9	0.758	0.831
Pulverized-Fuel-Fired Units																
3.0	1	500–700	64.3	2.1	14.3	19.3	0.664	0.807	0.807	0.807	64.3	2.1	14.3	19.3	0.664	0.807
3.0	8	400–499	66.4	12.2	10.3	11.1	0.786	0.949	0.788	0.889						
2.0	6	400–499	54.7	7.3	23.8	14.2	0.620	0.972	0.710	0.858						
1.0	15	400–499	71.0	3.9	19.3	5.8(?)	0.749	1.000	0.732	0.942	65.8	8.0	16.5	9.7	0.738	0.903
3.0	12	300–399	52.1	12.7	18.2	17.0	0.648	0.954	0.664	0.830						
2.0	8	300–399	68.0	8.6	3.4	20.0	0.766	0.843	0.738	0.800						
1.0	19	300–399	64.5	8.5	16.3	10.7	0.730	0.954	0.711	0.893	59.0	10.6	14.4	16.0	0.696	0.840
3.0	2	200–299	88.9	2.9	0.1	8.1	0.918	0.925	0.914	0.919						
2.5	2	200–299	41.6	28.4	11.4	18.6	0.700	0.825	0.807	0.814						
2.0	3	200–299	86.0	5.5	..	8.5	0.915	0.939	0.884	0.915						
1.0	6	200–299	71.0	3.5	11.3	14.2	0.745	0.905	0.602	0.858	73.2	9.3	5.4	12.0	0.825	0.879
Performance of pulverized-fuel-fired class											63.8	9.3	13.8	13.1	0.731	0.869
Oil- and Gas-Fired Units																
3.0	6	.....	54.5	26.7	11.6	7.2	0.812	0.969	0.897	0.928						
2.0	4	.....	52.8	14.7	18.9	13.6	0.675	0.908	0.779	0.864						
1.5	3	.....	47.0	2.3	12.5	38.2	0.493	0.706	0.489	0.618						
1.0	11	.....	71.7	5.6	8.9	13.8	0.773	0.977	0.758	0.862	57.9	16.1	12.4	13.6	0.740	0.864
Combined Fuels																
3.0	3	.....	39.0	36.6	19.2	5.2	0.756	0.972	0.936	0.948						
2.5	3	.....	86.2	3.7	0.2	9.9	0.899	0.929	0.868	0.901	60.5	21.7	10.5	7.3	0.822	0.927

TABLE 5 EFFECT OF LENGTH OF STUDY ON RESULTS OBTAINED—PERFORMANCE FIGURES BY CLASSES

1929 findings, all boilers: Period covered in years	Number of units in group	Use factor	Availability factors: Individual High	Individual Low	Group	1931 findings, all boilers: Period covered in years	Number of units in group	Use factor	Availability factors: Individual High	Individual Low	Group	1931 findings, after elimination of questionable units[a]: Period covered in years	Number of units in group	Use factor	Availability factors: Individual High	Individual Low	Group
Stoker-Fired Units																	
1	101	0.769	1.000	0.430	0.839	3	122	0.758	1.000	0.622	0.831	3	111	0.752	0.980	0.622	0.825
Pulverized-Fuel-Fired Units																	
1	62	0.723	0.952	0.494	0.877	3	82	0.731	1.000	0.602	0.869	3	73	0.730	0.972	0.602	0.855
Oil- and Gas-Fired Units																	
1	17	0.753	0.949	0.581	0.897	3	24	0.740	0.977	0.489	0.864	3	24	0.740	0.977	0.489	0.864

[a] Certain numbers given in Table 4 are followed by a question mark. Inspection will show that they represent extraordinarily low values for crippled hours. It is believed that the low values either result from an unintended interpretation of definitions or that the age of the units is not sufficient to have brought about a representative average performance.

TABLE 6 COMPARISON OF AVAILABILITY FACTORS OF PULVERIZED-FUEL-FIRED UNITS OF THE UNIT AND STORAGE TYPES

Type	Total hours reported	Crippled hours	Crippled hours not directly chargeable	Net crippled hours	Availability factor
Unit	234,113	36,198	25,833	10,365	0.955
Storage	828,153	112,887	78,112	34,775	0.958

TABLE 7 COMPARISON OF AVAILABILITY FACTORS OF STOKER-FIRED UNITS OF THE UNDERFEED AND TRAVELING GRATE TYPES

Type	Total hours reported	Crippled hours	Crippled hours not directly chargeable	Net crippled hours	Availability factor
Underfeed	1,305,217	213,404	86,152	127,252	0.903
Traveling grate	701,940	127,701	63,603	64,098	0.909

TABLE 8 COMPARISON OF AVAILABILITY FACTORS OF UNITS HAVING AIR PREHEATERS WITH THOSE NOT HAVING AIR PREHEATERS

	Total hours reported	Crippled hours	Crippled hours not directly chargeable	Net crippled hours	Availability factor
Stoker-Fired Units					
With preheaters	676,662	134,107	20,178	113,929	0.832
Without preheaters	984,016	154,578	51,163	103,424	0.895
Pulverized-Fuel-Fired Units					
With preheaters	671,443	117,567	62,688	54,879	0.918
Without preheaters	351,180	29,063	12,659	16,404	0.953
Oil- and Gas-Fired Units					
With preheaters	168,816	18,749	7,245	11,504	0.932
Without preheaters	141,071	10,802	3,780	7,022	0.950

fuel classes. Some of the finer detail has of necessity been omitted, as it would serve only to confuse, without bringing to light anything of general applicability.

In every instance there were several units for which the outages were classed by the operators as normal or expectable. This was particularly true of the older, well-tried boilers. These are not included in the table for obvious reasons. The causes listed, therefore, may be considered as the result of experimentation with new ideas, contingencies, inadequate or faulty design, peculiar local conditions of fuel and operating methods, or a combination of several or all of these factors.

The three or four largest items of outage are presented for each class of unit and are arranged in order of magnitude. They appear in the table in the column headed "Part of unit crippled." Under the next column, headed "Reason for outage," are listed the causes of shutdown as furnished by the operators.

No attempt was made to list them in order of importance or magnitude as the individual lengths of time are not available. They are considerably abbreviated, but every effort was made to preserve the intended meaning of the operators.

In examining Table 9 it will be noticed that, with one exception, the furnace, stoker, boiler, and superheater were the greatest offenders in the stoker-fired classes. The exception was the large outage due to cleaning dust-clogged air preheaters and excess slag accumulations from the boiler tubes. This latter probably would have resulted in crippling the boiler proper if left long enough, so it might with reason be termed an incipient boiler outage, if not an actual one.

The stokers seem to have had their greatest difficulties with preheated air and the wrong type of coal. In each pressure class one of the largest stoker outages was due to burned parts, caused by preheated air in the two higher pressure classes, and low-ash, low-volatile coal in the 200–299-lb class. No details were given for the 300–399-lb class. As the 500–700-lb class units are fired with traveling grates and the 400–499-lb class use underfeed stokers, no evidence is given as to the relative ability of the two types to resist preheat.

Plain refractory walls were the cause of considerable crippled time in all classes but that of 400–499-lb. In this class 8 of the 12 boilers have furnaces which are almost entirely water-cooled. In many instances slag accumulation and erosion hastened the failure of the plain walls, which was to be expected in view of the large stoker outages due to coal conditions.

Tube replacements and valve repair constituted the greater part of the boiler and superheater outage, with three instances in the 300–399-lb class of warped elements in radiant superheaters. Coal conditions again appear to have considerable effect on performance, as many operators reported excessive slag deposits on tubes and overworked or ineffective soot blowers in cases where the amount of ash entrained with the furnace gases was abnormally high.

A somewhat different picture is presented in the pulverized-fuel column. In only one pressure class does the fuel-burning equipment, i.e., the burners and pulverizing and conveying equipment, take a position as one of the three largest sources of outage. In the 300–399-lb class considerable time was spent in replacing mill hammers, liners, and grinding balls, but in all other instances the operators considered such outages as normal.

TABLE 9 MAJOR OUTAGES

Pressure class	Stoker-fired units			Pulverized-fuel-fired units			Oil- and gas-fired units		
	Order of magnitude	Part of unit crippled	Reason for outage	Order of magnitude	Part of unit crippled	Reason for outage	Order of magnitude	Part of unit crippled	Reason for outage
500–700 lb...	1	Furnace	(a) Plain refractory walls	1	Furnace	(a) Arch failures			
			(b) Low fusion coal			(b) Burned secondary-air-inlet parts			
	2	Stoker	(a) Burned parts due to preheated air	2	Economiser	(a) Some tubes rerolled twice each month			
			(b) Mechanical drive						
	3	Boiler and superheater	(a) Overloaded soot blowers due to coal used						
			(b) Replacement of tubes						
400–499 lb...	1	Stoker	(a) Burned parts due to preheated air	1	Boiler and superheater	(a) Rearrangement of superheater supports			
			(b) High ratings			(b) Failure of superheater tubes			
			(c) Removal of secondary rams and links on link grates						
	2	Boiler and superheater	(a) Renewal of valve seats and disks	2	Furnace	(a) Rearrangement of front walls to secure more suitable air distribution			
	3	Cleaning gas side of heating surface	(a) Plugged air preheaters						
			(b) Excess slag accumulation						
300–399 lb...	1	Stoker	(a) Burned stoker parts	1	Furnace	(a) Failure of water-cooled steel blocks	1	Boiler and superheater	(a) Warping of support members by leaking gases
			(b) Mechanical drive			(b) Rebuilding refractory walls			(b) Leaking stop valves
	2	Boiler and superheater	(a) Tube replacement	2	Boiler and superheater	(a) Replacement of radiant-superheater elements	2	Furnace	(a) Chipping of burner wall brick
			(b) Valve repair			(b) Superheater tubes cut by steam from misaligned soot blower			
			(c) Failure of radiant superheater elements						
			(d) Slag deposits						
	3	Furnace	(a) Slag deposits	3	Pulverising equipment	(a) Renewal of mill hammers, liners, grinding balls			
			(b) Rebuilding refractory walls	4	Draft equipment	(a) Vibration of induced-draft fans			
						(b) Erosion of fan blades by fly ash			
						(c) Failure of fan-shaft assemblies at preheaters			
200–299 lb...	1	Boiler and superheater	(a) Replacement of tubes	1	Boiler and superheater	(a) Tubes cut by steam from misaligned soot blowers			
						(b) Overheated superheater tubes			
	2	Furnace	(a) Rebuilding refractory walls						
	3	Stoker	(a) Burned parts due to low-ash, low-volatile coal						
			(b) Mechanical drive						

Furnace troubles were much the same as with the stoker-fired units where plain refractory walls were concerned. In the cooled or protected walls considerable outage was due to the burning of the wall surface. In the case of one station the bare steel blocks were removed after unsuccessful attempts to keep them in service, and cast-iron blocks were substituted for them. The results had been satisfactory at the time of the last report. In another instance the bare secondary air-port blocks burned badly, necessitating frequent replacement.

Two stations experienced difficulty in maintaining correct alignment of soot-blower elements due to warping. The result was the cutting of boiler and superheater tubes to such an extent that tube replacements were abnormal.

Only one station reported outages other than normal in the oil-and-gas-fired group. Even these were of a comparatively minor character, and all repairs were made during scheduled overhaul periods.

Conclusions

In general, the results of the three-year study bear out the relationships indicated by the 1929 report, the differences being only those of degree. All groups and classes have apparently dropped slightly in availability, probably because of added outages, which generally are not annual occurrences. The question of age might arise, as the greater number of units included in the study were two years older at the end of the three-year period than they were in 1929, but it is not believed to have had much influence in this case because of the fact that boilers of radically different ages were considered in both the short-time and long-time studies.

There still seems to be no appreciable difference in reliability between pulverized-fuel firing and stoker firing. The small margin in favor of the former method is very small when compared with the range between high and low individual values in both groups. While it is true that outages due to the stoker proper are somewhat greater than those chargeable to the pulverizing and burning equipment alone, it must be kept in mind that there are many other factors of intangible character which might easily have a more than balancing effect. Such factors are the amount of stoker and pulverizer repair work done as scheduled overhaul, and local fuel and load-schedule conditions. Also, as pointed out in both previous papers, any hard and fast conclusions must be avoided in a study of this kind because of the many variables involved and the differences of opinion and maintenance standards which are inevitable in a widely scattered group of some sixty-odd operators.

Probably one of the most illuminating parts of this final study is the material contained in Table 9. The influence of local conditions is brought out very sharply. Many units are afflicted with a very poor grade of coal, while others suffer from the use of fuel of too high a grade. Experiments in design are being conducted in many stations. Preheated air is becoming more generally understood.

The modern steam-generating unit seems to have been in a state of rapid and constant development at the time when these data were collected. The authors have reason to believe that this is still the case. Such development inevitably results in abnormal operating and maintenance conditions, which are reflected in the data collected. Suffice it to say that even in this transitory stage, modern steam-generating equipment may be relied on for use from 82 to 86 per cent of the time. There is every indication that this state of affairs will improve somewhat in the future, but until changes in design have become less radical and operating methods more standardized, many abnormal conditions will arise which will tend to confuse and distort the impressions of the investigator and append the question mark to many of his conclusions. For this reason the authors are terminating their study with this paper, believing that a continuation at this time would not be productive of any appreciable increase in the knowledge of the art.

FSP-55-9

Influence of Bends in Inlet Ducts on the Performance of Induced-Draft Fans

By L. S. MARKS,[1] J. LOMAX,[2] AND R. ASHTON,[3] CAMBRIDGE, MASS.

This paper shows the great loss in capacity and efficiency of a fan that results from the location of elbows or bends close to the inlet boxes. The loss is shown to be a function of the type of bend, of its "aspect ratio" or ratio of depth to width, and of the direction in which the air is turned in the bend. By the use of guide vanes in the bend the loss can be almost entirely prevented. When scroll inlet boxes provided with guide vanes are used, this loss does not occur, but the capacity with such boxes is low, although the efficiency is maintained. The cause of the loss is not the friction in the bend, but is the poor distribution of the air leaving the bend and entering the inlet box. The use of a short length of straight duct between the bend and the inlet box will reduce the loss appreciably. Measurements by pitot tube of the distribution of velocity and static pressures at the entrance to the inlet boxes will indicate the existence of this loss and the desirability of inserting guide vanes in the bends.



L. S. MARKS



JOHN LOMAX



RANDOLPH ASHTON

IN A paper by one of the present authors (and another), before the Society in 1931,[4] the results were presented of tests on the influence of the inlet box on the performance of induced-draft fans. It was stated in that paper that the direction and velocity of approach of the gases at the fan inlet were determined by the form and dimensions (1) of the inlet box and (2) of the ducts leading the gases to the inlet box. In the present paper, the second of these factors is considered.

In the great majority of installations, the induced-draft fan is located high up in the power plant. The ducts conveying the gases come from below and make a sharp bend close to the inlet boxes of the fan. The investigations here presented had for their purpose (1) the determination of the influence of such bends on the fan performance and (2) the investigation of means to counteract the harmful effects of such bends. It is found that certain bends diminish greatly the capacity of a fan, but that this capacity can be in large part restored by simple and inexpensive means. It also is demonstrated that the pressure drop in the bend has only a minor influence on fan performance and that the important factor is the lack of uniform distribution of the gases leaving the bend.

The tests were carried out at the Gordon McKay Laboratory of the Harvard Engineering School. The fan used was a double-inlet, Turbovane, induced-draft fan supplied by the B. F. Sturtevant Company. This is the same fan that was used in the tests reported last year. The wheel, Fig. 1, is double-inlet, 38 in. diameter, $23^5/_8$ in. wide, and has radial tips. It is provided with curved aluminum inlet rings. In the majority of tests the modified rectangular box shown in Fig. 2 was used. This is the box that was found to give a performance practically identical with the performance with free inlet. In a few tests a scroll inlet box with guide vanes, Fig. 3, was substituted. This is the box that was found to reduce the capacity of the fan about 30 per cent below the capacity when using the box of Fig. 2. It was used in these tests with the expectation that the vanes would redistribute the arriving gases in such a way as to offset the disturbing effect of the bends; this expectation was in large part realized.

Nearly all the tests were made with the bends bolted directly to the inlet boxes and with a short length of straight duct, with flared inlet, preceding the bend so as to provide uniform air flow to the bend. In some supplementary tests short lengths of straight duct were inserted between the bends and the inlet boxes, to ascertain the extent to which a uniform flow could be reestablished by this means before the air reached the inlet boxes.

Three types of bend were used, long-sweep 45-deg and 90-deg elbows and square elbows. The details of these bends for the modified rectangular box are shown in Fig. 4 and for the scroll inlet box in Fig. 5. It will be seen that the depth of the duct in

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[2] John Lomax was born in Bolton, England, on September 11, 1907. He received the degree of B.Sc.Tech. in 1930 and M.Sc.Tech. in 1931 from the University of Manchester. During the past year he has held a Henry Fellowship and has been studying at the Harvard Engineering School. He received the degree of S.M. in 1932 from Harvard University.

[3] Randolph Ashton was born on December 8, 1897, in Bristol, Pa. He received the degree of A.B. from Harvard College in 1920 and S.B. in Mechanical Engineering from the Harvard Engineering School in 1922. Since that time he has been planner at the Western Electric Company, service engineer with the American Brown Boveri Company, and a member of the technical staff at the Bell Telephone Laboratories. During the past year, he has been doing research work at the Harvard Engineering School and received the degree of S.M. in 1932.

[4] "Influence of Inlet Boxes on the Performance of Induced-Draft Fans," by Lionel S. Marks and Ernst A. Winzenburger, Annual Meeting, New York, N. Y., Nov. 30 to Dec. 4, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

Contributed by the Power Test Codes Committee and presented at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

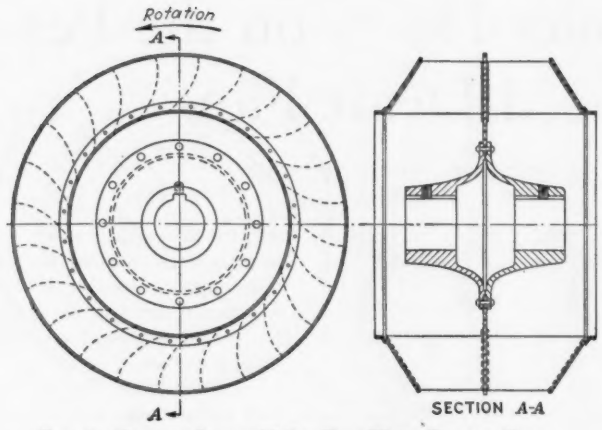


FIG. 1 DOUBLE-INLET 38-IN. FAN WITH RADIAL TIPS

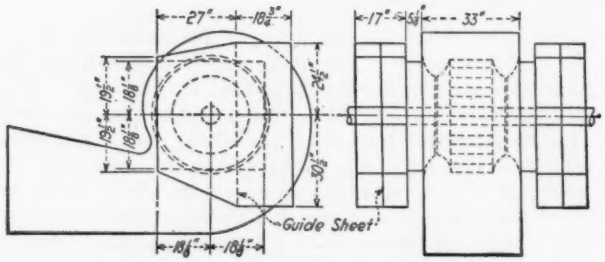


FIG. 2 MODIFIED RECTANGULAR INLET BOXES

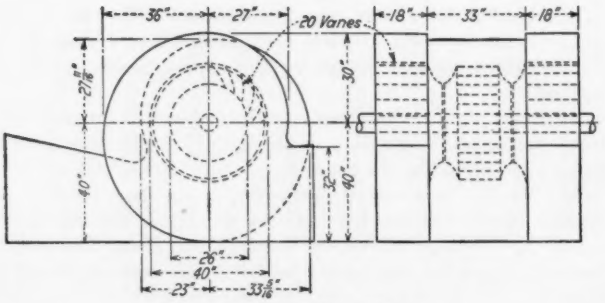


FIG. 3 SCROLL INLET BOXES

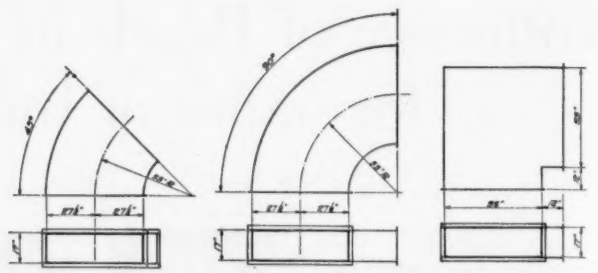


FIG. 4 BENDS USED WITH THE MODIFIED RECTANGULAR INLET BOXES

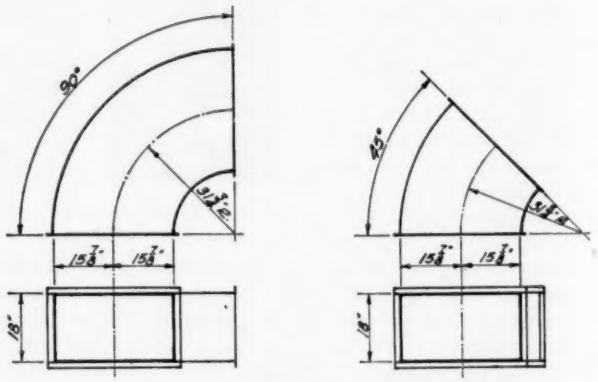


FIG. 5 BENDS USED WITH THE SCROLL INLET BOXES

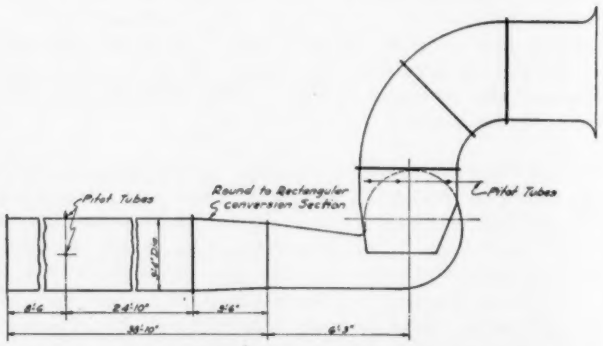


FIG. 6 TEST ARRANGEMENT WITH MODIFIED RECTANGULAR INLET BOXES AND 90-DEG. BENDS

(In direction to give air rotation in bends opposed to the direction of rotation of the fan wheel.)

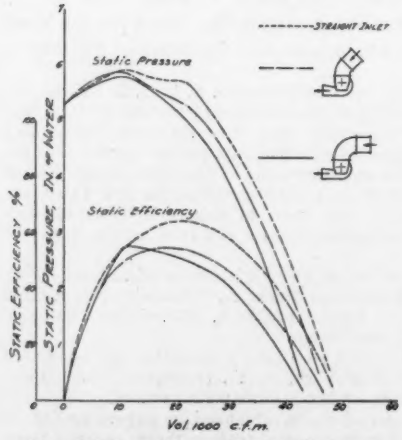


FIG. 7 LONG-SWEEP BENDS, OPPOSED ROTATION

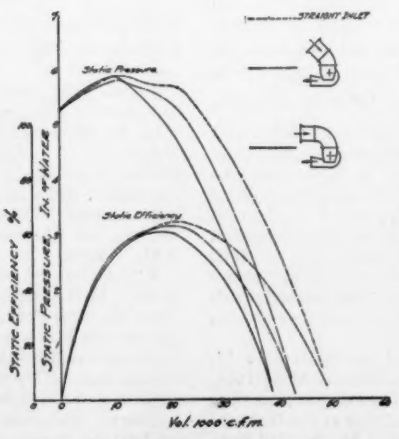


FIG. 8 LONG-SWEEP BENDS, SAME ROTATION

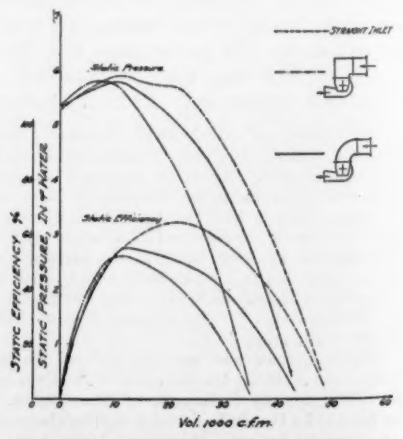


FIG. 9 SQUARE AND LONG-SWEEP BENDS, OPPOSED ROTATION

the plane of the bend is much greater than its dimension at right angles to that plane. For the modified rectangular inlet box this "aspect ratio" is 55 to 17 and for the scroll inlet box it is 31¾ to 18. These high ratios are presumably unfavorable to a good velocity distribution past the bend, and the tests indicate that inlet boxes with smaller aspect ratios would not show so great a loss of capacity resulting from the use of bends close to the inlet boxes.

Each bend was tested in two positions—(1) so oriented as to cause general flow of the air through the bend with the same direction of rotation as the rotation of the fan wheel and (2) with the bend turned through 180 deg so as to reverse this general direction of flow.

The arrangement for the tests is indicated in Fig. 6, which shows a modified rectangular inlet box, a 90-deg long-sweep bend in the position apparently giving air rotation opposed to the direction of rotation of the fan wheel, and a short straight duct with flared inlet. The tests were made in accordance with the Standard Test Code with the exception, necessitated by the objects of the test, that bends were substituted for the prescribed straight inlet ducts. The inlet total pressure was taken as atmospheric pressure except for the "corrected" results. These "corrected" results assume that the total inlet pressure at the fan is equal to the atmospheric pressure minus the friction loss

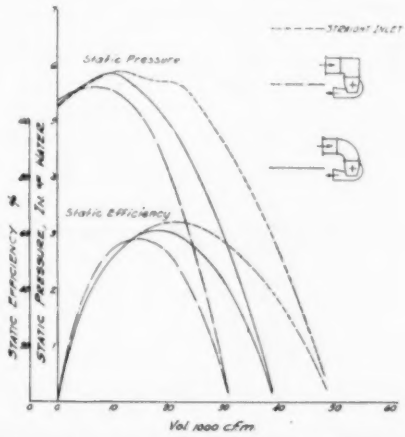


FIG. 10 SQUARE AND LONG-SWEEP BENDS, SAME ROTATION

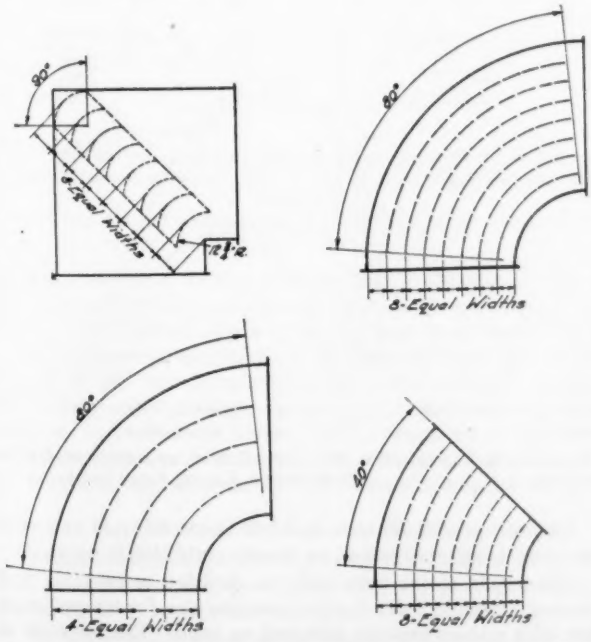


FIG. 11 GUIDE VANES IN THE BENDS FOR THE MODIFIED RECTANGULAR INLET BOXES

TABLE 1 RESULTS OF TESTS OF FAN WITH MODIFIED RECTANGULAR INLET BOXES AND WITH BENDS IN THE INLET DUCTS

(Speed, 900 rpm. Air density, 0.07488)

Volume, cfm	Pressure: Static, in. water	Pressure: Total, in. water	Efficiency: Static, per cent	Efficiency: Total, per cent	Horsepower
		Straight Inlet Duct			
0	5.24	5.24	0	0	8.32
10,360	5.92	6.0077	53.75	54.5	17.98
16,620	5.74	5.956	62.17	64.5	24.3
22,380	5.66	6.052	64.3	68.8	31.0
27,650	4.96	5.566	60.0	67.0	36.16
33,420	4.07	4.945	52.2	63.4	41.1
41,850	2.195	3.574	30.69	49.9	47.25
47,900	0.509	2.312	7.5	34.08	51.15
		45-Deg Bend, Air Rotation Opposed to Wheel			
0	5.285	5.285	0	0	8.05
10,420	5.86	5.944	50.1	50.7	19.26
16,200	5.54	5.745	54.15	56.1	26.1
21,830	5.27	5.634	54.5	58.2	33.35
26,920	4.72	5.286	50.0	56.0	40.15
32,000	3.89	4.691	42.5	51.2	46.1
40,400	2.086	3.359	23.92	38.55	55.5
45,900	0.528	2.179	6.12	25.2	62.4
45,500	0.524	2.152	6.12	25.08	61.4
		45-Deg Bend, Air Rotation Same as Wheel			
0	5.285	5.285	0	0	8.24
10,830	5.815	5.906	56.8	57.7	17.4
16,210	5.63	5.835	61.25	63.5	23.43
21,800	5.295	5.617	63.2	67.6	28.52
26,520	4.482	5.032	57.8	64.9	32.38
31,060	3.556	4.309	48.3	58.4	36.08
37,720	1.35	2.95	27.4	44.05	39.75
41,800	0.424	1.795	6.75	28.6	41.2
42,000	0.428	1.812	7.03	29.8	40.18
		90-Deg Bend, Air Rotation Opposed to Wheel			
0	5.28	5.28	0	0	8.36
11,150	5.79	5.888	55.7	56.5	18.37
16,440	5.45	5.663	53.2	55.2	26.6
21,260	5.03	5.384	49.7	53.15	33.96
26,420	4.476	5.022	48.25	51.9	40.7
31,120	3.627	4.333	38.08	46.1	49.35
38,070	1.858	2.996	20.2	32.6	55.1
42,800	0.465	1.901	5.2	21.22	60.3
42,400	0.456	1.867	5.01	20.5	60.9
		90-Deg Bend, Air Rotation Same as Wheel			
0	5.27	5.27	0	0	8.64
10,200	5.885	5.966	53.35	54.1	17.67
16,110	5.395	5.598	60.9	63.1	22.92
20,810	4.885	5.223	60.2	64.4	26.6
25,030	3.955	4.447	54.3	60.8	29.56
29,120	3.163	3.826	46.1	54.9	31.95
35,180	1.508	2.448	20.72	39.4	34.45
38,200	0.364	1.506	6.2	25.6	35.35
38,620	0.3658	1.4915	6.25	25.5	35.6
		90-Deg Square Bend, Air Rotation Opposed to Wheel			
0	5.35	5.35	0	0	7.16
10,490	5.72	5.82	52.5	53.4	18.01
15,530	5.08	5.28	49.5	51.4	25.1
19,780	4.46	4.76	44.3	47.3	31.46
23,800	3.73	4.15	37.3	41.7	37.2
27,210	2.85	3.43	28.80	34.66	42.5
32,060	1.339	2.140	13.74	22.0	49.10
34,640	0.297	1.239	3.08	12.86	52.6
		90-Deg Square Bend, Air Rotation Same as Wheel			
0	5.375	5.375	0	0	7.35
9,870	5.55	5.626	54.7	55.5	15.79
15,300	5.03	5.213	57.8	59.9	21.0
19,400	4.215	4.502	54.1	57.7	23.8
22,380	3.308	3.698	45.5	50.9	25.58
25,220	2.41	2.907	35.45	42.75	26.96
28,350	1.05	1.678	16.8	26.82	27.84
30,270	0.220	0.938	3.71	15.82	28.22
30,490	0.228	0.954	3.76	15.80	28.75
30,680	0.226	0.948	3.85	16.2	28.38

TABLE 2 COMPARISON OF CAPACITIES OF FAN WITH MODIFIED RECTANGULAR INLET BOXES AND WITH BENDS IN THE INLET DUCTS AT 900 RPM AND 5 IN. STATIC PRESSURE

Type of bend	Direction of air rotation in bend compared with that of wheel	Capacity, cfm	Decrease in capacity, per cent, compared with straight duct
Straight duct	....	27,500	....
45-deg bend	Same	23,800	13.5
45-deg bend	Opposed	24,900	9.5
90-deg bend	Same	19,500	29.0
90-deg bend	Opposed	22,000	20.0
Square bend	Same	15,900	42.0
Square bend	Opposed	16,700	39.0

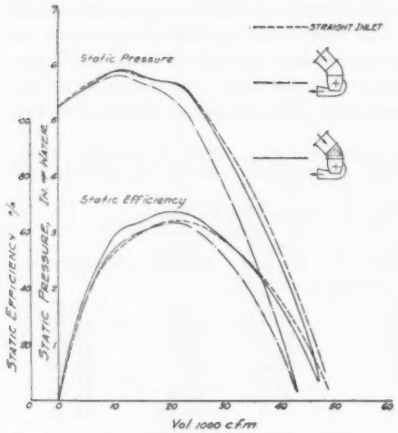


FIG. 12 GUIDE VANES IN 45-DEG BENDS, SAME ROTATION

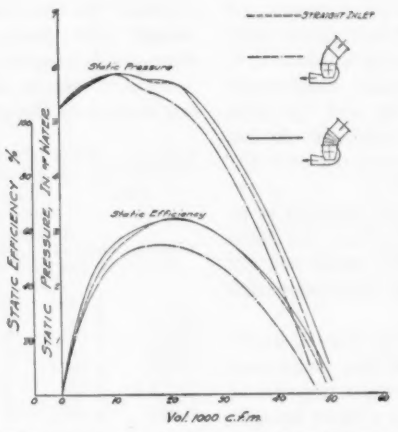


FIG. 13 GUIDE VANES IN 45-DEG BENDS, OPPOSED ROTATION

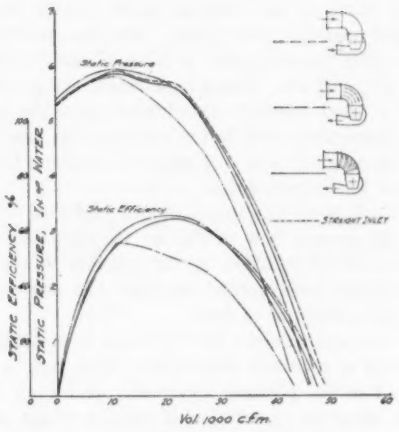


FIG. 14 GUIDE VANES IN 90-DEG BENDS, SAME ROTATION

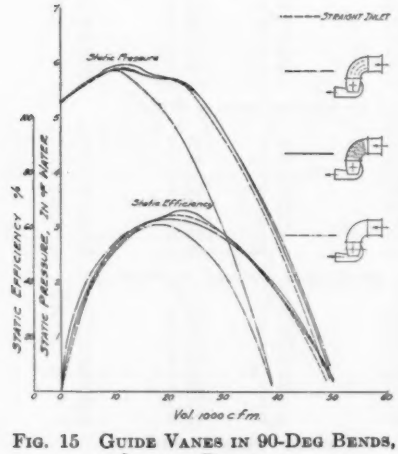


FIG. 15 GUIDE VANES IN 90-DEG BENDS, OPPOSED ROTATION

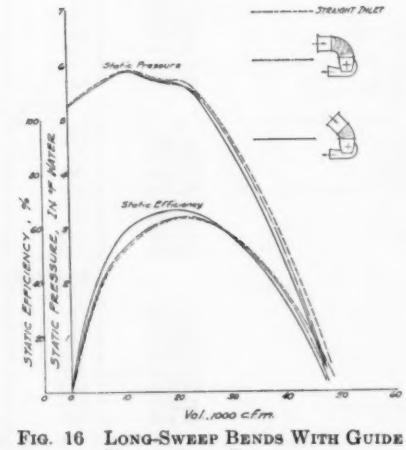


FIG. 16 LONG-SWEEP BENDS WITH GUIDE VANES, SAME ROTATION

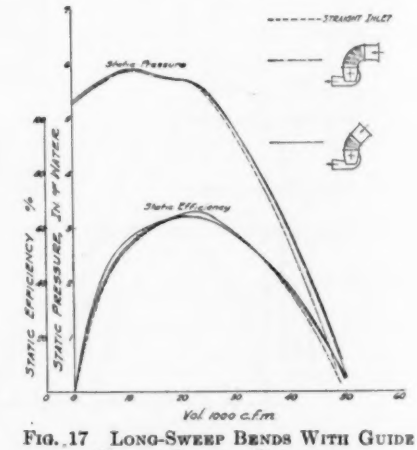


FIG. 17 LONG-SWEEP BENDS WITH GUIDE VANES, OPPOSED ROTATION

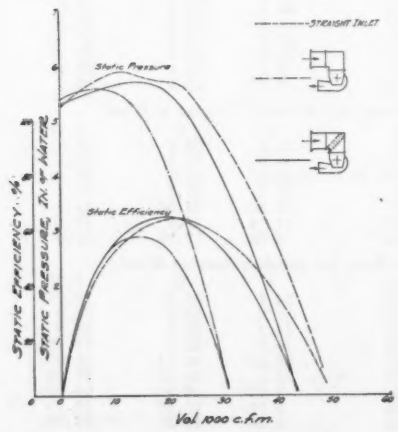


FIG. 18 SQUARE BENDS WITH GUIDE VANES, SAME ROTATION

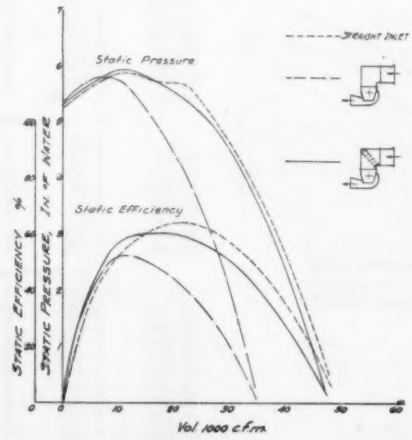


FIG. 19 SQUARE BENDS WITH GUIDE VANES, OPPOSED ROTATION

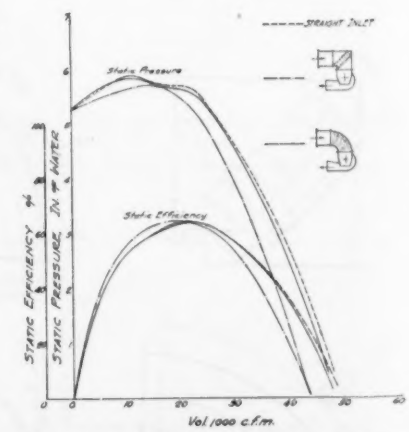


FIG. 20 LONG-SWEEP AND SQUARE BENDS WITH GUIDE VANES, SAME ROTATION

in the bends. The friction losses in the bends were calculated by using the results of the investigations of Loring Wirt[5] and, as already indicated, were found to be comparatively unimportant.

Power input was determined by both electrical and mechanical measurements. The agreement was very close; the mechanical measurements were used in calculating the fan performance.

[5] *Genl. Elec. Rev.*, June, 1927, pp. 286–296.

All tests were made at a speed of about 800 rpm and were corrected to 900 rpm and an air density of 0.07488 lb per cu ft.

The results of the tests with the modified rectangular inlet boxes are given in Table 1, which includes also, for comparison, a test with a short straight duct and no bend. The influence of the bends on the performance of the fan is shown in Figs. 7 to 10. The numerical comparison of Table 2 gives fan capacities at

5 in. static pressure, which is about the desirable normal operating condition for this fan at 900 rpm. It is seen that the loss in capacity for 90-deg bends is approximately twice the loss with 45-deg bends and that the loss in square bends is very great, being in one case almost twice that of a 90-deg long-sweep bend. It also is noteworthy that the capacity is greater when the direction of air rotation in the bend is opposed to the direction of rotation of the wheel. If the air were actually rotating as it entered the fan wheel, this result would seem anomalous, but there is no reason to expect that any definite spin would result

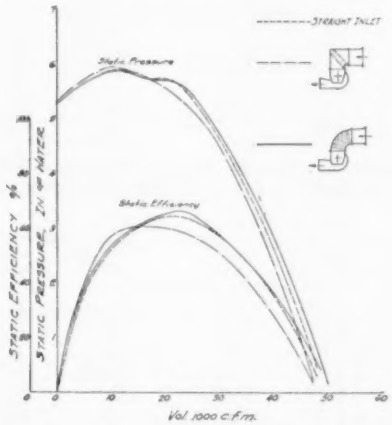


FIG. 21 LONG-SWEEP AND SQUARE BENDS WITH GUIDE VANES, OPPOSED ROTATION

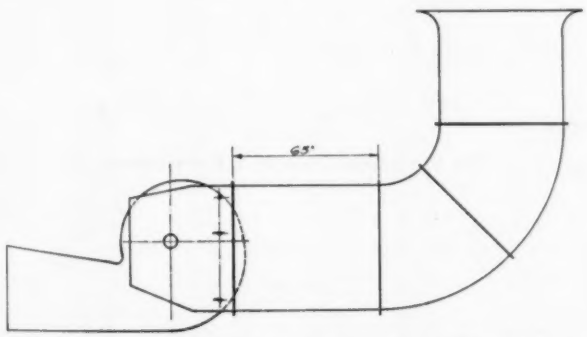


FIG. 22 TEST ARRANGEMENT WITH SHORT STRAIGHT DUCT INTERPOSED BETWEEN A 90-DEG BEND AND THE MODIFIED RECTANGULAR INLET BOX

(Air rotation through the bend in the same direction as the rotation of the fan wheel.)

from the passage of the air around the bend. The actual effect of such passage is rather a crowding of the air on the outer side of the bend. This is shown by pitot-tube readings, as indicated later.

The influence of the bends on fan efficiency also is shown in Figs. 7 to 10. It is interesting to note that with air rotation opposed to wheel rotation (which gives higher capacity) the efficiency is considerably lower. The maximum static efficiency falls from 64 per cent with a straight duct to 55 per cent with 45-deg and 90-deg bends and to 52 per cent with a square bend.

USE OF GUIDE VANES IN BENDS

The tests described indicate clearly that the ill effects of bends are due to the resulting poor distribution of the air arriving at the fan inlet. It seemed that this distribution could be improved by the use of guide vanes in the bends, which would have the effect of converting a single bend into a number of concentric bends of low aspect ratio. The details of the guide vanes used

TABLE 3 RESULTS OF TESTS SHOWING THE EFFECT OF GUIDE VANES IN THE BENDS IN THE INLET DUCTS

(Speed, 900 rpm. Air density, 0.07488)

Volume, cfm	Pressure: Static, in. water	Pressure: Total, in. water	Efficiency: Static, per cent	Efficiency: Total, per cent	Horsepower
45-Deg Bend With Seven Guide Vanes, Air Rotation Opposed to Wheel					
0	5.3	5.3	0	0	7.23
10,510	5.91	5.996	57.25	58	17.09
16,600	5.76	5.976	62.65	64.9	24.12
22,620	5.685	6.085	64.6	69.3	31.32
27,940	5.05	5.658	60.25	67.5	36.85
33,430	4.19	5.065	51.6	62.3	42.75
42,800	2.306	3.741	30.2	49.0	51.4
49,000	0.592	2.482	8.3	34.8	56.7
49,070	0.594	2.481	8.1	33.76	56.65
45-Deg Bend With Seven Guide Vanes, Air Rotation Same as Wheel					
0	5.25	5.25	0	0	7.22
11,160	5.9	5.996	61.3	62.3	16.88
17,110	5.75	5.981	66.0	68.7	23.41
22,700	5.58	5.984	66.95	71.75	29.8
27,430	4.865	5.453	61.15	68.5	34.4
32,650	3.95	4.782	52.4	63.5	38.76
41,050	2.152	3.468	30.75	49.5	45.2
46,150	0.521	2.2	7.96	33.55	47.62
46,500	0.526	2.190	8.2	31.8	46.95
90-Deg Bend With Three Guide Vanes, Air Rotation Opposed to Wheel					
0	5.24	5.24	0	0	7.19
11,580	5.99	6.08	57.3	58.2	17.42
16,620	5.79	6.02	62.3	64.6	24.39
22,300	5.62	5.98	62.4	66.6	31.65
28,030	5.065	5.68	58.5	65.7	38.23
33,400	4.18	5.05	50.9	61.5	43.3
42,990	2.307	3.75	29.87	48.3	52.35
48,900	0.580	2.45	7.7	32.6	57.6
90-Deg Bend With Three Guide Vanes, Air Rotation Same as Wheel					
0	5.38	5.38	0	0	7.79
10,800	5.950	6.041	58.9	59.7	17.13
16,730	5.83	6.05	64.5	66.8	23.8
22,230	5.58	5.99	65.4	70.1	30.13
27,210	4.94	5.52	61.8	69.2	34.5
32,500	4.000	4.831	52.6	63.5	38.9
40,200	2.072	3.33	30.1	48.4	43.5
45,200	0.486	2.082	7.45	31.9	46.3
90-Deg Bend With Seven Guide Vanes, Air Rotation Opposed to Wheel					
0	5.25	5.25	0	0	7.46
10,500	5.88	5.965	55.95	56.8	17.4
16,580	5.75	5.966	61.9	64.1	24.3
23,170	5.65	6.0595	66.9	71.8	31.6
28,050	5.01	5.626	60.53	67.0	37.15
33,550	4.15	5.027	51.8	62.8	42.3
42,480	2.296	3.711	30.5	49.3	50.25
48,650	0.576	2.431	8.04	34.0	55.02
49,060	0.591	2.471	8.23	34.4	55.6
90-Deg Bend With Seven Guide Vanes, Air Rotation Same as Wheel					
0	5.29	8.29	0	0	8.15
10,600	5.84	5.927	54.3	55.1	17.9
16,500	5.705	5.918	61.1	63.5	24.23
22,500	5.58	5.975	64.5	69.0	30.8
27,750	4.93	5.53	60.2	67.6	35.98
32,860	4.025	4.869	52.0	62.9	40.1
41,500	2.146	3.438	30.3	48.5	46.3
46,650	0.533	2.248	8.02	33.8	49.8
46,900	0.543	2.269	8.04	33.6	49.95
90-Deg Square Bend With Seven Guide Vanes, Air Rotation Opposed to Whee					
0	5.33	5.33	0	0	7.26
10,790	5.975	6.065	58.32	59.2	17.4
16,560	5.73	5.944	60.18	62.5	24.85
22,020	5.44	5.82	59.1	63.2	31.92
27,560	4.885	5.479	54.5	61.0	38.92
32,750	4.03	4.864	46.4	56.0	44.75
41,000	2.13	3.445	25.96	41.8	53.48
46,500	0.549	2.241	6.8	27.75	59
90-Deg Square Bend With Seven Guide Vanes, Air Rotation Same as Wheel					
0	55.31	5.31	0	0	7.35
10,300	5.66	5.74	56.8	57.5	16.2
16,250	5.71	5.918	64.0	66.2	22.9
21,610	5.35	5.716	64.5	68.9	28.27
26,780	4.55	5.106	59.5	66.7	32.16
31,400	3.641	4.412	50.4	61.0	35.83
37,500	1.853	2.954	27.8	44.2	39.4
42,580	0.443	1.861	7.88	33.1	41.3
42,700	0.459	1.89	7.5	30.8	41.35

are shown in Fig. 11. In the case of the square bends, the diagonally located guides are of circular form, all of the same radius.

The results of fan tests using these guide vanes in the bends are given in Table 3 and are shown in Figs. 12 to 21. It is apparent on inspection of Figs. 12 to 17 that the full capacity of the fan can be in large part restored by the use of such guide vanes in long-sweep bends. Moreover, it appears from Figs. 14 and 15 that three guide vanes, or an aspect ratio of about 0.8, give almost as good a fan performance as seven guides, or an

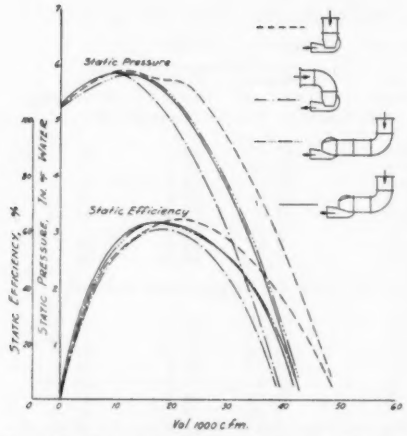


FIG. 23 STRAIGHT DUCT BETWEEN BENDS AND FAN, SAME ROTATION

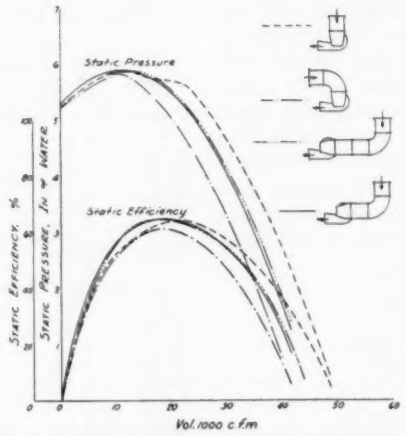


FIG. 24 STRAIGHT DUCT BETWEEN BENDS AND FAN, CORRECTED FOR INLET FRICTION LOSSES

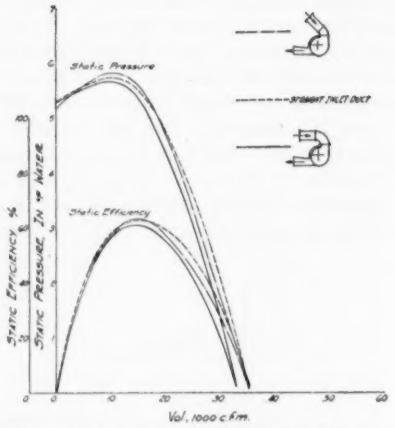


FIG. 26 SCROLL INLET BOX WITH BENDS SAME ROTATION

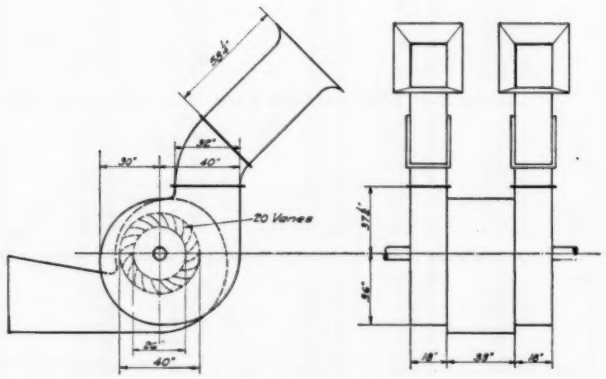


FIG. 25 TEST ARRANGEMENT ON INLET SIDE WITH SCROLL INLET BOXES AND 45-DEG BENDS

(In direction to give air rotation in bends opposed to the direction of rotation of the fan wheel.)

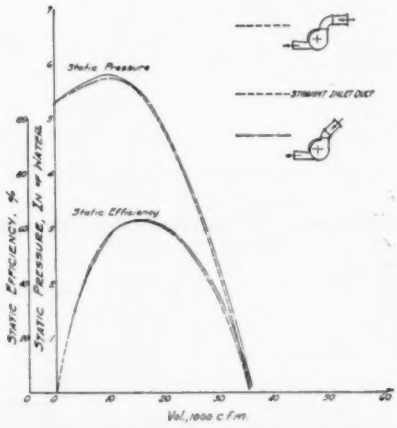


FIG. 27 SCROLL INLET BOX WITH BENDS, OPPOSED ROTATION

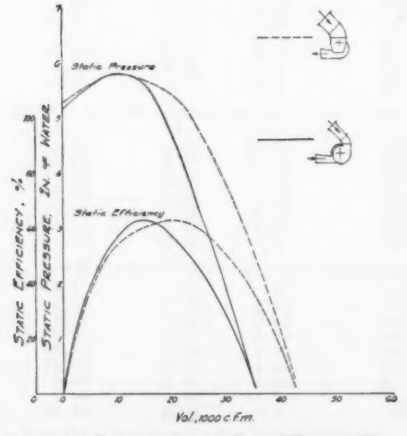


FIG. 28 COMPARISON OF INLET BOXES WITH 45-DEG BENDS, SAME ROTATION

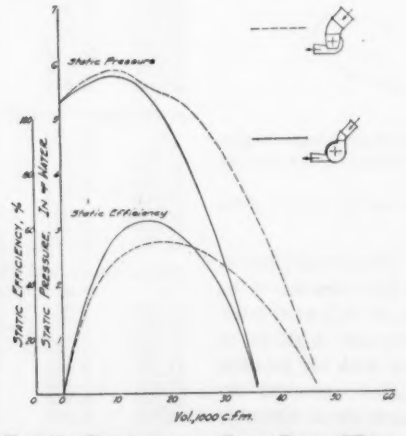


FIG. 29 COMPARISON OF INLET BOXES WITH 45-DEG BENDS, OPPOSED ROTATION

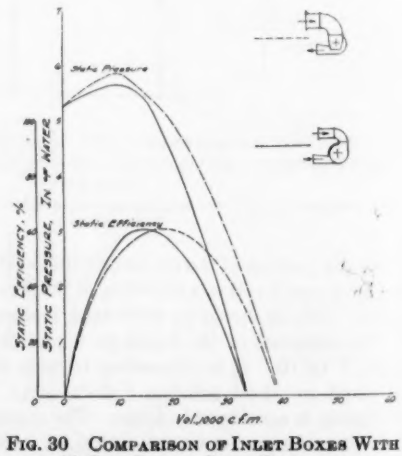


FIG. 30 COMPARISON OF INLET BOXES WITH 90-DEG BENDS, SAME ROTATION

aspect ratio of about 0.4. A smaller number was not tried, but it would appear probable that the harmful effects of long-sweep bends next to the inlet boxes would not be serious so long as the aspect ratio of the duct is not greater than unity.

The guide vanes used with the square bends restore nearly the full capacity with opposed direction of air flow but with some loss in efficiency (Fig. 19), and with the same direction of air flow give it full efficiency but with some loss in capacity (Fig. 18). The gain is greater than with the long-sweep bends, but the fan performance is definitely inferior to that with long-sweep bends and guide vanes. This is clearly shown in Figs. 20 and 21.

STRAIGHTENING THE AIR FLOW PAST THE BENDS

In many installations it is possible to insert short lengths

TABLE 4 RESULTS OF TESTS WITH MODIFIED RECTANGULAR BOXES, 90-DEG BENDS, AIR FLOW THROUGH BENDS WITH ROTATION SAME AS WHEEL, AND WITH SHORT STRAIGHT INLET DUCTS BETWEEN THE BENDS AND THE INLET BOXES

(Speed, 900 rpm. Air density, 0.07488)

Volume, cfm	Pressure: Static, in. water	Pressure: Total, in. water	Efficiency: Static, per cent	Efficiency: Total, per cent	Horsepower
		Length of Straight Duct, 5 Ft 5 In.			
0	5.28	5.28	0	0	7.75
10,490	5.86	5.946	57.3	58.1	16.82
16,500	5.67	5.883	63.5	66.0	23.2
21,750	5.21	5.581	62.6	67.0	28.5
26,350	4.485	5.027	57.6	64.5	32.3
31,200	3.592	4.35	49.5	59.9	35.66
38,200	1.775	2.924	26.6	43.8	40.1
42,100	0.36	1.75	5.77	28.0	41.3
		Length of Straight Duct, 10 Ft 5 In.			
0	5.24	5.24	0	0	7.64
10,380	5.82	5.903	56.25	57.0	16.91
16,580	5.715	5.93	63.8	66.1	23.35
21,380	5.25	5.607	61.9	66.1	28.48
26,150	4.503	5.038	57.2	64.0	32.42
31,180	3.638	4.398	49.8	60.1	35.91
38,300	1.793	2.943	26.75	43.8	40.55
42,500	0.353	1.773	5.6	28.02	44.5

TABLE 5 RESULTS OF TESTS WITH SCROLL INLET BOXES AND WITH BENDS IN THE INLET DUCTS

(Speed, 900 rpm. Air density, 0.07488)

Volume, cfm	Pressure: Static, in. water	Pressure: Total, in. water	Efficiency: Static, per cent	Efficiency: Total, per cent	Horsepower
		Straight Inlet Ducts			
0	5.29	5.29	0	0	7.49
10,440	5.75	5.835	58.5	59.2	16.21
16,100	5.4	5.602	62.9	65.0	21.77
20,420	4.71	5.036	59.3	63.5	25.5
24,350	3.86	4.324	52.85	59.2	28.06
27,750	2.915	3.518	42.6	51.4	29.9
32,500	1.307	2.131	21.2	34.6	31.44
35,160	0.283	1.246	4.9	21.6	31.9
		45-Deg Bends, Air Rotation Opposed to Wheel			
0	5.28	5.28	0	0	7.52
10,550	5.80	5.88	58.9	59.8	16.39
16,170	5.44	5.64	63.1	65.5	21.9
20,600	4.735	5.07	59.9	64.3	25.55
24,490	3.88	4.34	52.6	58.9	28.0
28,080	2.925	3.539	43.4	52.2	29.95
32,880	1.347	2.19	22.25	36.25	31.26
35,430	0.299	1.283	5.27	22.6	31.65
		45-Deg Bends, Air Rotation Same as Wheel			
0	5.16	5.16	0	0	7.75
10,540	5.83	5.92	58.2	59.8	16.52
16,370	5.46	5.67	63.4	65.8	22.20
20,070	4.75	5.06	57.8	61.5	25.93
23,910	3.707	4.15	49.8	55.9	27.9
27,230	2.765	3.33	39.8	48.1	29.7
32,000	1.244	2.007	20.14	33.1	31.07
35,070	0.285	1.205	5.85	20.8	31.9
		90-Deg Bends, Air Rotation Opposed to Whee			
0	5.21	5.21	0	0	7.58
10,500	5.725	5.810	58.1	59.0	16.33
16,045	5.45	5.651	63.7	66.0	21.62
20,380	4.72	5.045	59.5	63.5	25.43
24,400	3.84	4.304	52.7	59.1	27.98
28,220	2.906	3.524	42.8	51.9	30.2
32,970	1.352	2.2	22.2	36.15	31.62
35,800	0.305	1.305	5.39	23.1	31.86
		90-Deg Bends, Air Rotation Same as Whee			
0	5.25	5.25	0	0	7.52
10,380	5.665	5.755	56.9	57.7	16.24
16,020	5.235	5.435	61.0	63.4	21.55
19,870	4.465	4.771	55.6	59.6	25.01
23,330	3.55	3.98	48.1	53.8	27.11
26,400	2.603	3.144	37.7	45.5	28.66
30,420	1.196	1.920	18.8	30.1	30.55
32,930	0.264	1.093	4.41	18.24	31.1

of straight duct between the bends and the inlet boxes. It is a commonly accepted belief that a straight duct whose length is from three to four times the mean of the width and depth of a rectangular duct will result in the restoration of fairly uniform air distribution in the duct. In order to find the effect on fan performance of the introduction of a short length of straight duct, the 90-deg long-sweep bend was used in a position to give air rotation in the bend in the same direction as the rotation of the wheel. This arrangement is shown in Fig. 22. The lengths of duct used were 5 ft 5 in. and 10 ft 5 in., corresponding respectively to 1.8 and 3.5 times the mean duct dimension.

The results of these tests are given in Table 4 and in Fig. 23. It is seen that the shorter length of duct is practically as effective as the greater length and that in both cases nearly half of the loss in capacity is restored and the loss of efficiency is in large part regained. As the ducts introduce friction loss which in Table 4 and Fig. 23 is charged against the fan, further calculations have been made correcting the static pressure for the friction loss in the bends and in the ducts. The results of these are shown in Fig. 24, which indicates that the ducts restore more than half the loss in capacity and that the full fan efficiency is regained.

It may be noticed in Figs. 23 and 24 that the tests with short straight ducts inserted were made with a horizontal position of the inlet boxes. They were compared with tests (1) without a bend and (2) with a bend next to the inlet box, and in both cases with a vertical position of the inlet boxes. That this change, necessitated by the vertical height of the laboratory, has no influence is indicated by the good agreement between the test without bend here reported and the similar test reported last December,[6] which was made with horizontal inlet boxes.

Scroll Inlet Boxes

A series of tests was made with scroll inlet boxes and vanes, as shown in Figs. 3 and 25, to ascertain whether a fan with this type of inlet box was sensitive to non-uniformity of the air stream approaching the inlet or whether the vanes at the inlet made the fan comparatively "foolproof" in the matter of inlet-duct arrangement. The fan was tested with a short straight inlet duct and with 45-deg and 90-deg long-sweep bends of the dimensions shown in Fig. 5. The tests included both orientations of each bend, so as to ascertain the influence of air-rotation direction. The results of these tests are given in Table 5 and Figs. 26 and 27.

An inspection of Figs. 26 and 27 shows that the bends have a negligible influence on the fan performance. Under the conditions of Fig. 27, no effect can be detected when a 90-deg bend is added; and consequently, since the results given are uncorrected for elbow friction, the actual performance of the fan itself was improved. With the 45-deg bend this is still more marked, both as to capacity and efficiency. Under the conditions of Fig. 26, an inferior performance is seen for the 90-deg bend, while with the 45-deg bend the performance is inferior at high capacities and superior at medium capacities. The fan is seen to be practically unaffected by the inlet-duct conditions.

This advantage of the scroll inlet box is obtained at the cost of a considerable reduction in capacity but with a marked increase in maximum efficiency when the air rotation in the bend is opposed to the direction of rotation of the wheel. This is seen in Figs. 29 and 31, which compare the two types of inlet box with 45-deg and 90-deg bends, respectively. With the reversed position of the bends, Figs. 28 and 30, the maximum efficiencies are unchanged, but the capacity does not fall off so much.

Direction of Air-Stream Rotation in the Bends

It is not suggested in this paper that the air in passing through a bend is set in rotation. The term rotation is used merely to indicate the direction in which the stream of air entering a bend is deviated from a rectilinear path. If this direction, as viewed from the side of the fan, is the same as the direction of rotation of the fan wheel, the air flow is spoken of as being in the same direction of rotation as the wheel.

The air-flow direction in the bends has a considerable influence on fan performance in those cases where the influence of the bend is considerable. For bends provided with guide vanes or with the scroll inlet box, this influence is small, but is recognizably of the same kind. With opposed air flow, the capacity (cfm at a given static pressure) is greater than with the same direction of flow, with the occasional exception of a reversed condition at low capacities. This is shown clearly in Figs. 32, 33, and 34. At

[6] *Loc. cit.*

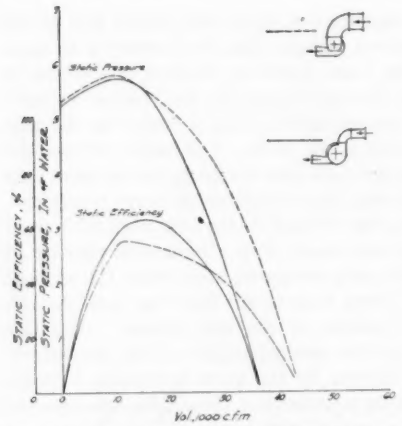


FIG. 31 COMPARISON OF INLET BOXES WITH 90-DEG BENDS, OPPOSED ROTATION

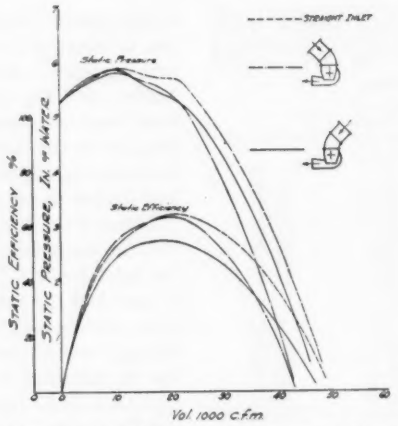


FIG. 32 AIR ROTATION EFFECTS, 45-DEG BENDS

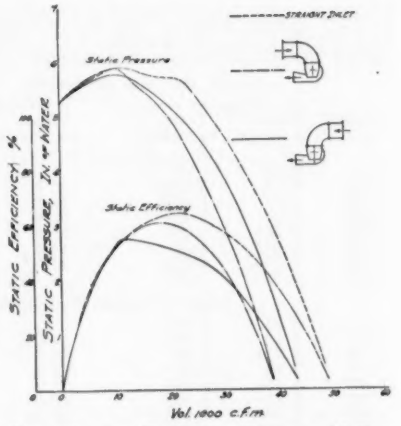


FIG. 33 AIR ROTATION EFFECTS, 90-DEG BENDS

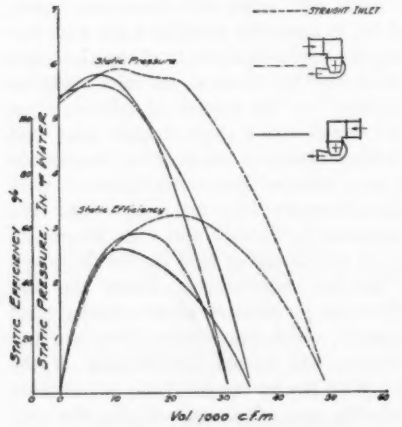


FIG. 34 AIR-ROTATION EFFECTS, SQUARE BENDS

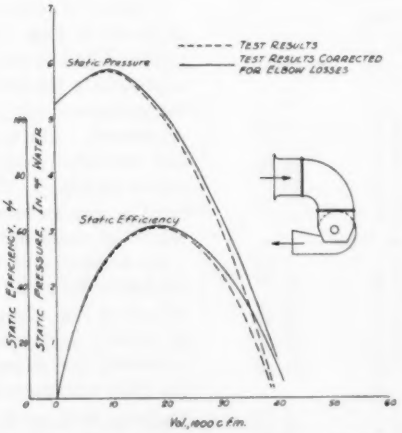


FIG. 35 ELBOW-LOSS CORRECTION, 90-DEG BENDS

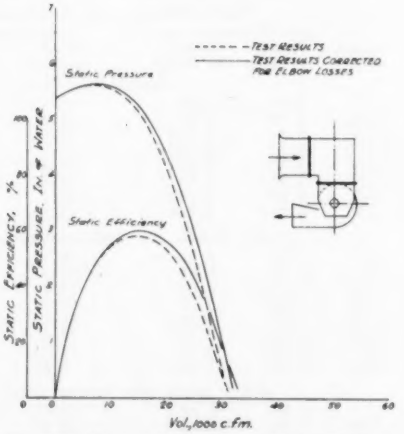


FIG. 36 ELBOW-LOSS CORRECTION, SQUARE BENDS

the same time the efficiencies with opposed air flow are less than with the same direction of flow, except at the highest capacities (cfm). At the normal operating range the difference in capacities is not great, but the difference in efficiencies is considerable. Consequently, the preferred condition for a normal installation should be with air rotation in the same direction as wheel rotation.

CORRECTION FOR FRICTION LOSS IN THE BENDS

The test results and the comparisons already given, with the exceptions specifically noted, consider the bends as part of the fan and neglect the friction losses up to the inlet boxes. The total pressure at the inlet boxes is assumed to be atmospheric pressure. It is obvious that this assumption reduces the static pressure and the static efficiency of the fan below the true values and makes the comparison between a fan with bends and a fan without bends less favorable than it should be. The influence of the bends consequently appears to be greater than it actually is.

Corrections for friction losses can be made if these losses are known with sufficient accuracy. The investigations of Wirt[7] appear to be the only ones available that can be directly applied to this case. Although those investigations seem to be of a high order of accuracy, the authors have thought it best not to rely on them in the presentation of their experimental results.

Calculations have been made on the basis of Wirt's work, and they show that the friction loss is a minor factor when considered in relation to these tests. The corrections attain a maximum value for the tests shown in Figs. 35 and 36, and these are for conditions for which the influence of the bends on air distribution is a maximum, so that the friction effect is comparatively unimportant. In general, since these tests are made on a particular fan provided with particular bends, the exact magnitude of the results obtained are not important. Ignoring the small friction loss does not affect the general results.

DETECTION OF POOR AIR DISTRIBUTION AT THE INLET BOXES

Pitot-tube readings were taken at three locations (see Fig. 6) at the entrance to the inlet boxes during most of these tests. As the air flow is highly irregular at this location when bends are used and as the direction of flow is presumably not parallel to the axis of the duct, it is obvious that pitot tubes oriented parallel to that axis cannot give correct impact or static pressures. It was found, however, that they gave important indications as to uniformity of flow. In the case of the bends without guide vanes, the velocity head was high on the outer side of the bend, while the static pressure was high on the inner side of the bend. It is thought that the conditions of air distribution in any installation can be simply determined by a pitot-tube search of this kind. The evidence available is not considerable, but indicates that with good air distribution a pitot tube located as suggested will show practically constant velocity head in all positions and will show static pressure a little higher in the

[7] *Loc. cit.*

middle of the duct, but without much change in going to the outer positions along the depth of the duct.

Acknowledgments are made to the B. F. Sturtevant Company for the loan of the fan and for supplying the ducts used in this investigation.

Discussion

H. F. Hagen.[8] The reductions in output reported by the authors explain satisfactorily the unsuccessful fan performances encountered so frequently in induced-draft work. Many power plants have curving inlet flues that give the kinds of flows investigated by the authors. The bad effect of such conditions has been understood, at least partially, for some time. It was to avoid these trouble jobs that the writer undertook to develop a fan with a stable performance that would be unaffected by poorly designed flue work.

Experimentation discouraged the use of vanes placed directly in a conventional inlet box. The uneven distribution of air flow apparently occurred before the air entered the box, and no effective arrangement of vanes was discovered that would correct what was already wrong.

The spiral box with vanes was then worked out, and it accomplished the main purpose of the research. The design had the disadvantage of requiring a somewhat larger and more expensive fan when test-block performances alone were considered. Actually, as may be seen by a comparison of the stable curves of the scroll-inlet fan, Figs. 26 and 27, with the reduced, or what might be called the "installed," performances of the simple box, Figs. 7, 8, 9, and 10, the scroll-inlet capacity is fully as great as the capacity of the simple box under actual installation conditions. Its efficiency is much higher, and there is no unexpected, and consequently disastrous, drop in performance. In fact, if tests of the simple box had been made with flues installed, as the authors have done in their tests, there would have been no question about the advantages of the scroll-inlet fan from all standpoints.

Professor Marks' ingenious use of directing vanes in the elbows before the boxes has shown that it is possible to get stable performance with simple boxes with suitable flow conditions, and he has secured the highly desirable results of stable performance and maintained efficiency with no sacrifice of capacity. His work is important to the power-plant operator in showing how large improvements can be made inexpensively in existing installations. The work is also of vital interest to the fan manufacturer as a suggestion for future development.

However, it is almost a necessity that the vanes as used in Professor Marks' experiments be made part of the flue work, and there may be some hesitation on the part of purchasers to assume the consequent responsibility of correcting the flow of gases before the fan even though the fan engineers may be willing to share this responsibility.

The scroll-inlet fan has come through the tests showing that it could as a self-contained unit compensate fairly completely for almost any condition of flow. The installation of a fan designed after this manner would permit the fan manufacturer to assume a clear-cut responsibility, and possibly at a somewhat decreased hazard.

As a result of the authors' work we can feel reasonably certain of successful performance with either the simple box and suitable vanes in the connecting flues or with the self-contained scroll design. The simple box at present is less expensive, while the scroll design may be somewhat more generally reliable. The writer has a leaning toward the fan that is stable in itself and does not depend upon necessary corrective design on the part of others who may not be entirely familiar with the principles involved. Both types of fans are being used at the present time and it will be interesting to observe future tendencies.

C. E. Peck.[9] The data in this paper show the importance of obtaining uniform air distribution at the entrance to a fan. Non-uniform distribution is the principal reason why fan performance in the installation differs considerably from that obtained under test conditions where the inlet is free or special precautions are taken to insure uniform air distribution. Other duct constructions, in addition to bends, which may be located close to the fan inlet and which affect distribution considerably are:

1 Ducts with large total angles of divergence near blower inlet
2 Dampers or regulating valves
3 Sudden change in duct cross-section from a smaller to a larger area.

It has been the writer's experience that a round diverging duct with a total conical angle of divergence as small as 20 deg will not only tend to reduce the fan capacity, but also to produce

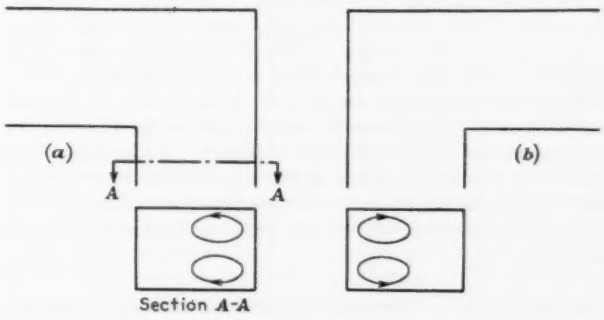


Fig. 37

cyclic variations in the capacity and power input of considerable magnitude. A rectangular duct which diverged in one direction only with an angle of approximately 20 deg caused very unstable fan operation when placed near the inlet of a propeller-type blower. Irregular cyclic variations in capacity of 50 to 75 per cent took place, accompanied by increased noise. This unstable condition existed over the entire pressure-volume range. It was observed in this particular case that the air stream did not follow both sides of the diverging wall simultaneously, but only one side or the other, an irregular shifting of the stream suddenly taking place. Simultaneous with the shifting of the air stream in the diverging duct, the large variations in fan capacity would take place. Various combinations of baffling and screening did not reduce the intensity of this unstable condition to any marked degree. The obvious remedy of reducing the angle of divergence to a small amount (4 to 6 deg total angle), by lengthening out the duct, eliminated this unstable condition.

The foregoing emphasizes the need of carefully laying out the duct installation in which a fan must operate if efficiencies near to those obtained under standard test code conditions are to be maintained. Also, any type of duct construction which may produce non-uniform air distribution at the blower inlet tends to increase the windage noises. In some cases, depending on the type of inlet to the fan, non-uniform air distribution causes marked increases in amount of noise.

[8] Vice-President, in charge of research, B. F. Sturtevant Company, Boston, Mass.

[9] Power Engineering Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Jun. A.S.M.E.

With regard to the flow in the bend, the results indicated that the blower performance was changed depending on the position of the bend relative to the rotation of the blower. As shown in Fig. 37, the flow after the turn has been made gives rise to the well-known "secondary flow," which consists of localized rotations in a plane approximately perpendicular to the side walls of the duct. In (*a*), the direction of the secondary whirls are opposite to those in (*b*), looking at a cross-section of the exit side of the turn. This difference in direction of the flow in each case for the same direction of the rotation of the fan is probably linked up with the difference in blower performance.

The magnitude of these secondary whirls determines the "piling up effect" on the outer wall of the bend, which in turn influences the air distribution in the duct beyond the bend. This depends on the shape of duct, radius ratio at the turn, aspect ratio, etc. The influence of these factors on the losses in elbows has been quite thoroughly investigated by Loring Wirt. (See footnote 5 of the paper.) The authors have shown comparisons of effects on performance with a fixed aspect ratio. For the modified rectangular inlet box, this aspect ratio is defined as 55 to 17. If used in the same sense as in Wirt's article, the aspect ratio is defined in the inverse manner—namely, 17 to 55. Used in this way, Wirt gives data which show that the loss in the elbow increases with decreasing aspect ratio. Similarly, the authors' tests indicate that inlet boxes with larger aspect ratios would not show so great a loss of fan capacity when bends are close to the inlet boxes. This, together with Wirt's tests, indicate that the greater the loss in the bend, the greater the non-uniformity of distribution and the greater the effect on the fan performance when the bend is close to the blower. This conclusion is useful, because if the loss in the elbow is kept small, it may be placed in the duct system close to the fan with little sacrifice of capacity, and the use of guiding vanes may be eliminated.

W. R. Heath.[10] This paper should be of interest to all users of induced-draft fans and to all designers of mechanical draft systems, as it gives the results of carefully made tests showing the influence of various types of bends on the performance of one type and one design of induced-draft fan. Whether the same bends would have the same influence on another type or design of fan is open to question. The writer has observed tests made under unfavorable inlet conditions which show very plainly that a fan with open inlet is much more sensitive to variations of flow approaching the inlets than is a fan fitted with properly designed fixed inlet vanes. The latter fan was very definitely a more "foolproof" type, and in addition the use of vanes at this point improves to a marked degree the fan efficiency.

The influence of the various bends on capacity is interesting, but the efficiency is not of much moment, since the particular type of fan under test does not have the desirable non-overloading horsepower characteristic, and it would be necessary in most applications to provide reserve motor capacity anyway on this account.

The comparison of capacities in Table 2 are based on a constant pressure, which is misleading from a practical application viewpoint. For instance, if a fan were selected to give 27,500 cfm and 5 in. static pressure at 900 rpm, which is the operating condition selected by the authors, the user would desire to know the expected decrease in capacity when operating on the same system and at the same speed and with the type of bend used. Of course, if the capacity were decreased, the resistance of the system would no longer be 5 in., but would become a less value determined by varying this pressure as the square of the capacity. In other words, the "system characteristic" should be plotted and the capacity interception of this curve with the pressure curve of the fan when throttled by the individual bend should be compared with the original 27,500 cfm desired. Incidentally, this is the method of comparison used in the A.S.M.E. paper of 1931 by one of the co-authors of the current paper.[11] On this basis, Table 2 would show:

		—System-characteristic—method		—Constant-pressure—method	
Type of bend	Direction of rotation	Capacity	Actual decrease in capacity, per cent, compared with straight duct	Capacity	Decrease in capacity, per cent, shown in paper
Straight duct	.....	27,500	...	27,500	...
45-deg bend	Same	24,800	10.0	23,800	13.5
45-deg bend	Opposed	26,100	5.0	24,900	9.5
90-deg bend	Same	24,750	9.5	19,500	29.0
90-deg bend	Opposed	26,100	4.75	22,000	20.0
90-deg square	Same	22,300	18.5	15,900	42.0
90-deg square	Opposed	23,700	13.5	16,900	39.0

The new table shows that, when compared on a practical basis, the decrease in capacity of the particular fan under test with the various type of bends averages one-third to one-half the values indicated by the paper—still too much to ignore, but considerably less than the authors intimate.

Under the heading, "Straightening the Air Flow Past the Bends," and in Fig. 24, the authors show that by introducing a straight length of duct 5 ft. 5 in. long (1.8 diam) the loss in capacity is restored by more than half. However, the addition of another 5 ft 0 in. (1.7 diam) restores a very small portion of the balance of the loss. It is therefore reasonable to suppose that each additional 5 ft 0 in. of duct will have less and less effect, or, in other words, that it would be impossibie to restore the full fan capacity by any practical length of duct. It would then follow that a bend anywhere in the system would have a harmful effect on the performance of this particular fan. This would indicate one of two things: (1) that the base curve of the fan with straight boxes is higher than could be duplicated on future tests and therefore in error, or (2) that the particular fan under test was unduly sensitive to slight changes in air flow. The latter is more likely, since the fan inlet is open (i.e., not equipped with guide vanes to stabilize its performance). Furthermore, the "wave" in the pressure curve so apparent in nearly all the curves strongly indicates an unstable condition inherent in the fan at at least one point; also, this is the same fan which was used in preparing the paper a year ago, previously referred to, in which the authors found that with rectangular boxes without guide sheets, "the performance was unstable" and that "the fan would develop periodically a roaring noise which coincided with a power increase of 10 to 15 per cent, but without noteworthy change in the discharge volume."

This paper claims certain advantages for the scroll-type inlet boxes when there are bends at the entrance to the boxes. An investigation of the data, however, shows the following:

Assume the same desired condition of 27,500 cfm and 5 in. static pressure at 900 rpm with a 90-deg bend at entrance to the boxes and with the preferred condition of air rotation the same as wheel rotation. Selecting the proper size fan both with rectangular-inlet boxes and with scroll-inlet boxes from the data corresponding to Fig. 10 and Fig. 26, respectively, thus taking full account of the effect of the bend on the fan performance, gives the comparison shown on the next page.

Hence, it is apparent that, as concluded in the 1931 paper, the scroll-inlet boxes fall below the rectangular-inlet boxes in

[10] Assistant Chief Engineer, Buffalo Forge Company, Buffalo, N. Y.

[11] "Influence of Inlet Boxes on the Performance of Induced-Draft Fans," by L. S. Marks and E. A. Winzenburger, Trans. A.S.M.E., vol. 54, no. 21 (1932), paper FSP-54-16.

Type box	Size fan	Cfm	Static pressure, in.	Rpm	Hp
Scroll.....	41½	27,500	5	900	40.3
Rectangular...	40¼	27,500	5	900	37.4

NOTE: Points on curves for fan performance from which these ratings were selected are from actual readings as given in the paper.

the order of merit. In the foregoing example, the fan with rectangular boxes would be slightly smaller, and therefore slightly cheaper, would have a less expensive construction of inlet boxes, and would consume less power. In addition, when the induced-draft fan handles gas with a considerable concentration of fly ash, as is so frequently the case, the scroll-inlet boxes would be subject to a scrubbing action by the ash and consequent erosion in a manner similar to the fan housing scroll itself without relieving the wear on the rest of the fan. This condition is not present with rectangular boxes.

The following is a summation of the foregoing discussion: (1) The paper shows completely the influence of bends in inlet ducts on the performance of one type or design of induced-draft fan only—a type of fan that does not have the desirable nonoverloading horsepower characteristic and a type of fan that shows some inclination toward inherent instability. The findings are not applicable to a fan fitted with properly designed fixed-inlet vanes. (2) The comparisons given in Table 2 are not made on a practical basis. (3) Rectangular boxes are, when intelligently used, a better selection than scroll-type boxes.

W. A. ROWE.[12] Under the heading, "Direction of Air-Stream Rotation," there is indicated the basis for the designation. If one will refer to Fig. 25, showing the arrangement with scroll inlet, it would seem as though a more appropriate designation with the scroll-inlet box would have reference to the side of the center line on which the downcomer is connected to the duct. As long as the inlet duct is connected to the right-hand side of that elevation, the direction the air takes with reference to the wheel might be more properly designated as of the same rotation. This would then be true were the 45-deg bend tilted in the opposite direction as well.

The consistency in the performance when viewed in this manner is indicated by the curves in Fig. 27 and the heavy-line curves on Figs. 28, 29, and 30. This interpretation would lay emphasis rather on the direction that the air particle takes in the inlet box just prior to entering the inlet of the fan and with reference to the direction of rotation of the fan wheel, rather than on any changes in direction at more remote locations in the inlet ducts.

It might be pointed out that while these determinations were based on one particular type of fan, there are other types of fans of equal or even possibly wider use in induced-draft work. In the case of all these types not only is care in the design of the inlet boxes and connections important, but it is also necessary to avoid unfavorable outlet connections. Failure to observe these precautions constitutes the great majority of fan-trouble jobs or complaints.

[12] American Blower Corporation, Detroit, Mich.

The important part of this contribution is to my mind disclosed in the last paragraph on the fifth page, where it is pointed out that the full capacity of the fan can be in large part restored by the use of proper precautionary measures in the design of the bends. Many years' experience in the adaptation of apparatus to commercial applications convinces that there are an almost infinite number of combinations of poor arrangements that can be encountered. The wide distribution which information of this kind will receive it is hoped may correct in large measure mistakes in the future by those who design or install those parts of the systems. While the fan manufacturer should for his own interest be on the alert to anticipate possible problems of this kind in order to prevent them at the inception, he should not be held to a guaranty of performance so all-embracing as to include every possible external disturbance of this nature.

This paper confirms the propriety of the existing Fan Testing Code in its refusal to approve any approximate method of field testing which might be used for the purpose of determining whether a fan complies with or fails to comply with its guaranteed performance. The code does not say where the test shall be made, but how it is to be done, and such check tests may be made either in the laboratory or the field, providing the method and means used comply with the provisions of the test code.

AUTHORS' CLOSURE

The authors are glad to have the confirming statements by Mr. Peck and to have his list of additional causes of lack of uniform air flow. The quantity called the "aspect ratio" in this paper is the reciprocal of the quantity so called in Wirt's paper.

With reference to the discussion by Mr. Heath, the authors believe that since the loss of capacity is due to poor distribution of the air approaching the fan inlet, the loss is likely to be of the same order of magnitude for all fans which are not provided with devices for overcoming this poor distribution. The authors are glad to have Mr. Heath's additional table showing the influence of bends on the capacity of a fan connected with a given system. This table is perhaps the most generally useful comparison. Table 2 was prepared as giving the basis for determining the increase in size of fan necessary for maintaining the fan capacity with the stated resistance and tip speed.

No evidence is presented by Mr. Heath to support his conclusion that the findings of this paper are not applicable to a fan with fixed inlet vanes. The second paragraph in Mr. Hagen's discussion of this paper presents a contrary view to that suggested by Mr. Heath. The authors have no first-hand information on this subject. In any case, it would appear to be preferable to obtain a uniform air stream entering the inlet boxes rather than attempt to correct lack of uniformity by special devices at the fan.

In reply to Mr. Rowe, the "direction of air-stream rotation" is defined as the direction in the bend, and this was the variable in the tests. A reference to Table 5 and to Figs. 26 and 27 shows that the performance of the fan with scroll-inlet boxes is affected by this direction of air-stream rotation, though not to as great an extent as with simple inlet boxes.

FSP-55-10

A System for the Measurement of Steam With Flow Nozzles for Turbine Performance Tests

By SANFORD A. MOSS[1] and WISTAR W. JOHNSON,[2] LYNN, MASS.



S. A. Moss

In this paper the authors give particulars regarding a system of steam-flow measurement by means of a nozzle for performance tests of jet-condensing, non-condensing, or extraction turbines where condensate measurements are out of the question. The system as outlined is on a laboratory basis in order to insure the accuracy necessary when steam-turbine guarantees are involved. One of the two appendixes presents an account of tests on which the system is based; the other deals with formulas for steam flow.



W. W. Johnson

THIS paper presents a system of steam-flow measurement by means of a nozzle for performance tests of jet-condensing, non-condensing, or extraction turbines where condensate measurements are out of the question. The flow is calculated from the pressure and temperature ahead of the nozzle, the direct reading of the differential pressure across the nozzle, and the dimensions of the nozzle and pipe. The system as outlined is on a laboratory basis in order to insure the accuracy necessary when steam-turbine guarantees are involved. Appendix No. 1 includes an account of tests on which the system is based.

This system was initiated by E. D. Dickinson, designing engineer of the Steam Turbine Department at the Lynn Works of the General Electric Company. It is based on recognized principles for the measurement of air at ordinary pressures,[3] of high-pressure air,[4] of low-pressure air from blowers,[5] and as an actual standard for calibration of orifices in thin plates.[6] Reference is made also to work at the Engineering Experiment Station at Ohio State University by Profs. S. R. Beitler and Paul Bucher.[7]

The Flow Nozzle

The Nozzle Shape, shown in Fig. 1, shall include a well-rounded approach formed by two circles which practically give a quarter-ellipse with a ratio of axes of 3:2. This is very nearly the shape used by the Bureau of Standards in Research Paper 49 already cited.

Nozzle Throat. However, even though there is a very gradual approach, it cannot be certain that the jet will completely fill the curved portion of the nozzle. Therefore, at the end of the curved portion, there shall be a parallel portion forming the nozzle throat, with a length of at least half a nozzle diameter. This is to insure that the jet has the exact diameter of the nozzle parallel portion at the point where its static pressure is taken and leaves no uncertainty as to jet size to be taken account of by the nozzle discharge coefficient.

The Throat Diameter D shall be selected to give differential pressures of at least 2 in. of mercury for the smallest flow and between 10 in. of mercury and 10 per cent of the high-side pressure for the largest flow with usual steam pressures. It is not proper to use a differential pressure too small for accuracy, to avoid pressure loss. It is obvious that the greater the differen-

[1] In charge of Mechanical Engineering Research, Thomson Research Laboratory, General Electric Co. Mem. A.S.M.E. Dr. Moss began his engineering career by serving a four years' apprenticeship as a machinist in San Francisco, Calif., afterward spending some years as a draftsman. He entered the University of California and was graduated from the College of Engineering in 1896, afterward receiving a Master's Degree from the University of California and a Doctor's Degree from Cornell University. This college work was interspersed with professional work as a draftsman and engineer. He became associated with the General Electric Company in 1903 and has since been engaged on mechanical engineering research work connected with gas turbines, steam turbines, centrifugal compressors, and superchargers for internal-combustion engines. He has been granted a number of patents for apparatus produced by the General Electric Company.

[2] Turbine Department, General Electric Co. Mem. A.S.M.E. Mr. Johnson received the degree of A.B. from the University of Oregon in 1907, B.A. from Oxford University in 1911, and B.S. from Massachusetts Institute of Technology in 1913. He has been associated with the General Electric Co. since 1913, with the exception of 1920, when he was connected with the Federal Shipbuilding Co. His work has been in steam research and mathematical calculations on steam turbines and turbine design and testing. He is a member of the A.S.M.E. Power Test Codes Individual Committee No. 11 on Complete Steam Power Plants.

[3] "The Impact Tube," by Sanford A. Moss, Trans. A.S.M.E., 1916, p. 761.

"Measurement of Flow of Air and Gas With Nozzles," by Sanford A. Moss, Trans. A.S.M.E., 1928, paper APM-50-3.

"Engineering Computations for Air and Gases," by Sanford A. Moss and C. W. Smith. Trans. A.S.M.E., 1929, paper APM-52-8.

[4] "Trade Standards" (Fourth Edition), 1930, The Compressed Air Society, New York.

[5] "Standard Test Code," National Association of Fan Manufacturers and American Society of Heating and Ventilating Engineers, 1923.

[6] Research Paper No. 49, Bureau of Standards, March, 1929; "Discharge Coefficients of Square-Edged Orifices for Measuring the Flow of Air."

[7] "Study of Primary Metering Elements in 3-In. Pipe," read at A.S.M.E. Meeting, Dec., 1931.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., Dec. 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

tial pressure, the better will be the accuracy of the test. The throat diameter D shall be between 40 and 60 per cent of the pipe diameter. In determining the nozzle diameter and U-tube capacity when laying out a test, use formula [4], Appendix No. 2.

Low-Side Taps. There shall be two to four low-side static-pressure holes. For nozzles of 2 in. throat diameter or larger, the throat pressures holes, if any, shall be between $^1/_4$ in. and $D/8$, but not larger than $^3/_4$ in. For nozzles smaller than 2 in. throat diameter, the throat pressure holes shall be about $D/8$, but not smaller than $^1/_8$ in. All static-pressure holes must have a straight length of at least twice the hole diameter, as established by the piezometer tests of Prof. C. M. Allen and L. J. Hooper.[8]

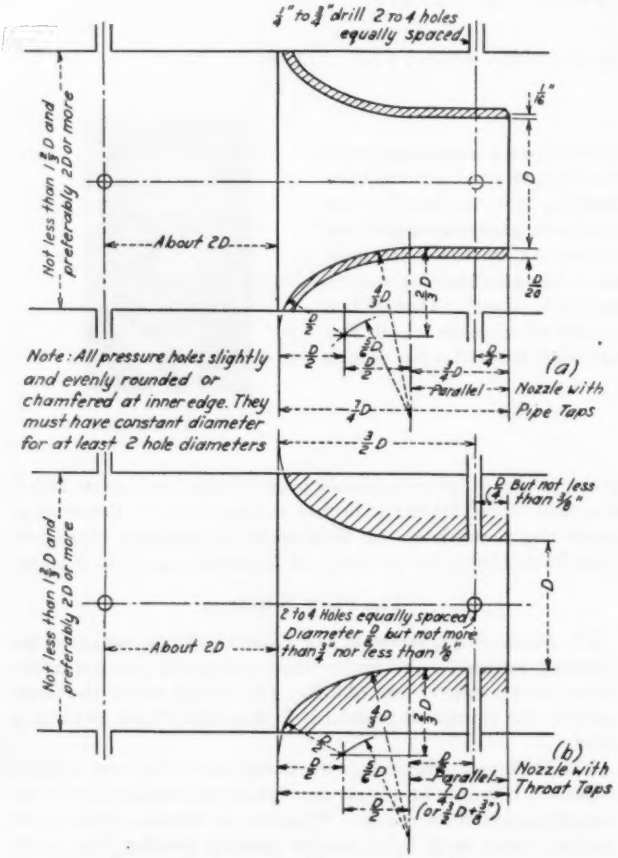


FIG. 1 DIAGRAMS OF RECOMMENDED NOZZLES

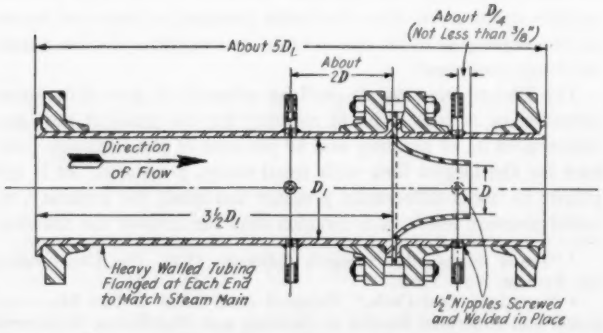


FIG. 2 PROPOSED DETAILS OF SPECIAL PIPE SECTION CARRYING NOZZLE WITH PIPE TAPS

These pressure taps shall be located about 0.25 D, but not less than $^3/_8$ in., upstream from the plane of the nozzle end. They may be arranged in either of two methods: throat taps directly in the nozzle throat, as in Figs. 1*b*, 5, 6, and 8, so as to give the actual throat pressure; or pipe taps in the pipe wall, as in Figs. 1*a*, 2, 3, 4, and 9, so as to give the pressure of the region into which the jet discharges.

When throat taps are used, special care shall be taken because of the high throat velocity to insure that the inner edges are perfectly formed. This may be accomplished by first rough-finishing the nozzle throat, drilling, reaming, and then plugging the throat holes with tight plugs, and finally finishing the throat; or the nozzle throat may be finished by grinding after the holes are drilled. In either case the inner edges of the pressure holes shall be smoothed with crocus cloth so as to remove all sharp edges and burrs. These edges should be inspected with a magnifying glass.

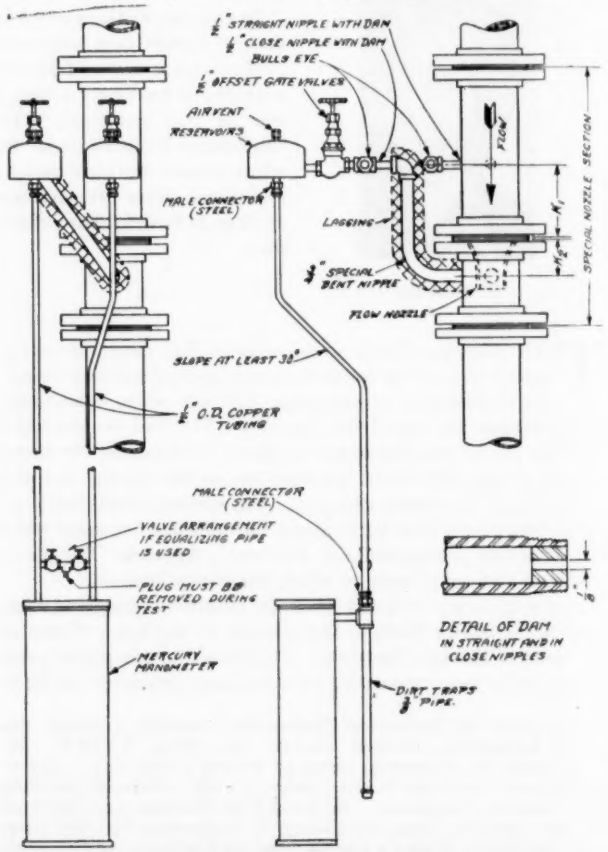


FIG. 3 PROPOSED SET-UP WITH PIPE TAPS FOR VERTICAL STEAM MAIN

When low-side pipe taps are used, the same rule in regard to length of hole shall be followed. Ordinary care to insure that the inner edges of the holes are slightly and evenly rounded or chamfered is sufficient.

The Nozzle Inner Surface shall be made of non-corrodible material, and shall have a polish which must be maintained by frequent cleaning. An important consideration is that no ripples shall be perceptible to the touch of the finger tips along the curve, its junction with the parallel portion, or the parallel portion itself. The nozzle diameter D shall be carefully measured in at least 4 axial and 4 radial places by means of an inside micrometer or carefully fitted inside gages measured by an outside micrometer. The various measurements must agree within about $D/1000$, but preferably within about 0.001 in.

[8] "Piezometer Investigation," by Charles M. Allen and Leslie J. Hooper, Trans. A.S.M.E., 1931, paper HYD-54-1.

Straight Pipe Preceding Nozzle. An essential item for smooth flow through a nozzle is a proper length of preceding straight pipe. One of the important items of the present work was an investigation of the actual distances at which a disturbance in the preceding pipe would affect the nozzle flow, given in Appendix No. 1. A rule previously given is that a nozzle should be preceded by 10 nozzle diameters of straight pipe in case of a preceding symmetrical irregularity such as a change in diameter, and by 20 nozzle diameters in case of an unsymmetrical irregularity, such as an elbow or tee. The conclusion is that this rule is conservative and should be followed if possible. However, if this is impossible, the lengths of straight pipe shall never be less than 8 and 16 nozzle diameters, respectively. The nozzle should be well centered with the preceding pipe, and care should be taken that gaskets do not extend into the nozzle opening.

A gate valve partly closed for controlling pressure gives serious irregularity and is not permissible unless it is about 200 D (not necessarily straight pipe) ahead of the nozzle or followed by a device for equalizing the steam flow such as a steam separator or other chamber. A globe valve wide open or partly closed was found to be an ordinary unsymmetrical irregularity, whether in a straight run or followed by an elbow or tee.

Valve and bend combinations occasionally occur which cause vortical motion which may give large errors. These may be eliminated by use of an egg box or straightener of some kind. It is probable that the error due to a vortex will always be so large as to indicate when a straightener is needed. However, some engineers may prefer to use a straightener in the pipe preceding the nozzle in every case.

Two to four high-side static-pressure holes with straight length of at least two hole diameters shall be provided in this pipe at a distance of about 2 D from the nozzle. The inner edges of these holes shall be slightly and evenly rounded or chamfered. The pipe diameter shall be measured in the plane of the holes to within about 0.01 in.

The straight pipe preceding the nozzle and for a short distance beyond the nozzle, as well as the nozzle itself, shall be well lagged.

The piping should be arranged so that the nozzle can be removed after a test and a filler substituted, which should always be done.

Special Nozzle Pipe Section. A section of bored pipe as in Fig. 2 may be arranged for insertion in the pipe line especially for the flow-nozzle test. This shall have a straight length of three to four pipe diameters on the inlet side of the nozzle, and one or two pipe diameters on the discharge side. The two to four sets of static-pressure holes shall be arranged in the special section as shown in Fig. 3 for vertical and in Fig. 4 for horizontal installations of the steam pipe.

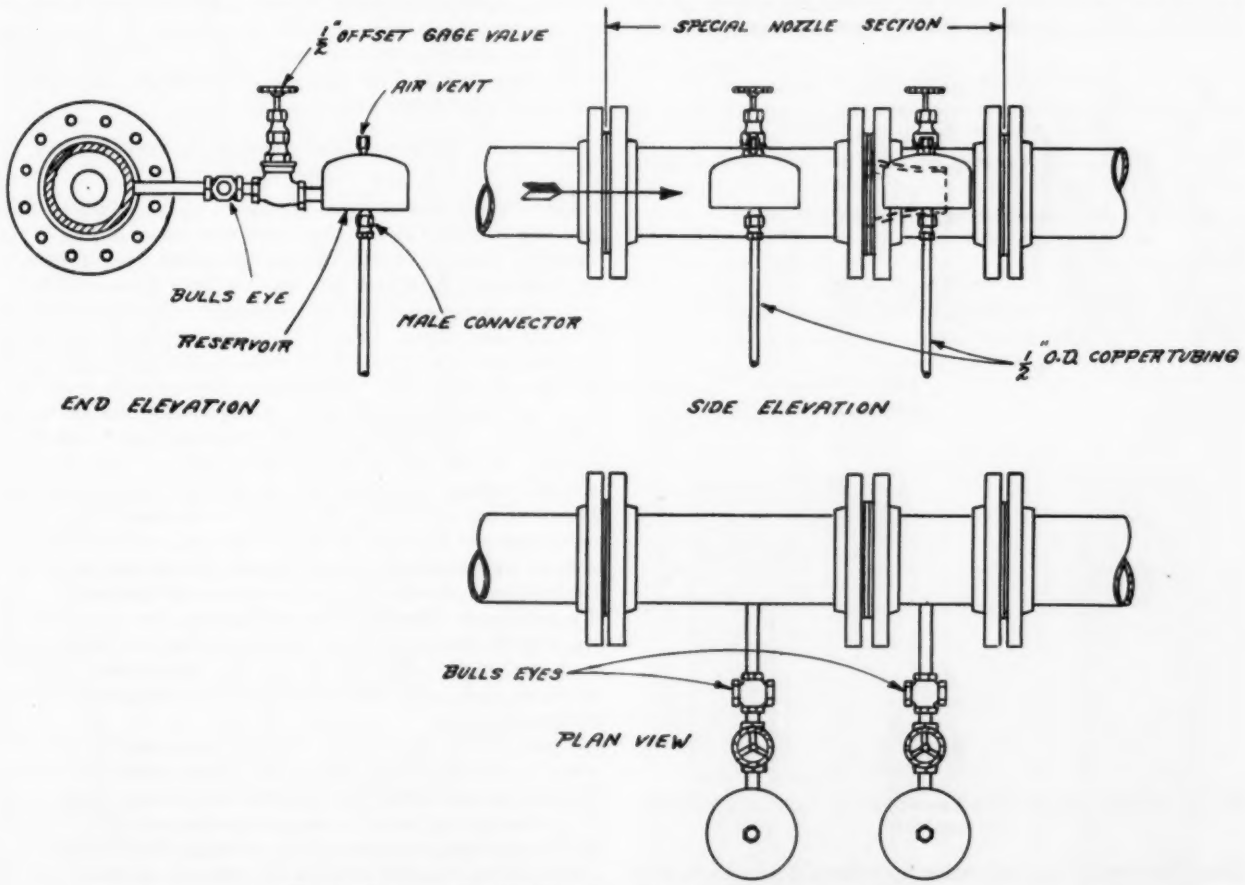


FIG. 4 PROPOSED SET-UP WITH PIPE TAPS IN HORIZONTAL STEAM MAIN

The pipes shall be bored accurately to size after the pressure nipples have been welded in so as to remove any irregularities in the pipe surface as well as to insure that the two or more sets of static-pressure connections will have identical finish one with the other as to hole edge, sharpness, etc. After the pipe has been bored, these holes shall be carefully inspected and wire edges or other irregularities removed as stated earlier under "Low-Side Taps."

The nozzles shall be doweled or otherwise secured in place in such a way that the center line of the nozzle throat coincides with

the center line of the pipe. A small amount of eccentricity of the nozzle in the nozzle pipe may produce errors of considerable magnitude, so that there is necessity for very accurate centering of the nozzle in the measuring section.

When such a special section is used, the nozzle discharge coefficient may be determined by a calibration of the complete special section, in a laboratory, with weighed flow, and with the static holes and all details exactly as in the test.

Nozzle Discharge Coefficient. This is the ratio of the actual flow to the theoretical flow, computed by a rational formula such as discussed in Appendix No. 2.

Values are given on curve 3, Fig. 10, and on Fig. 13 for a considerable range of conditions, and as the system becomes used, a complete body of coefficients will become established. Until such time as the nozzle discharge coefficients are completely established, there must be a calibration by weighed flows of each new nozzle.

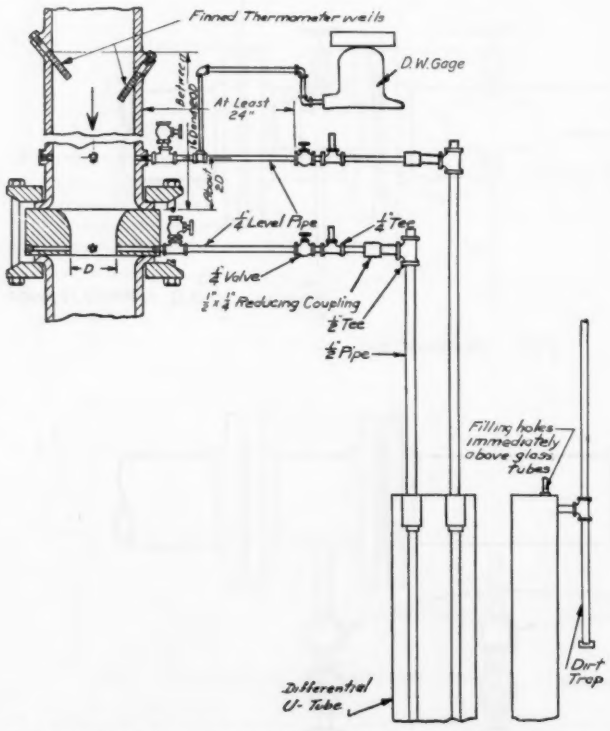


FIG. 5 SET-UP USED IN LYNN TESTS (THROAT TAPS) FOR VERTICAL STEAM MAIN

When throat taps are used, the nozzle discharge coefficient is the same as that of a nozzle freely discharging into the atmosphere or other large region. With a rational flow formula such as prescribed in Appendix No. 2 the coefficient is very nearly a true friction coefficient, and its value is nearly 0.99 over a wide range of velocities. When pipe taps are used the coefficients are slightly lower, due to greater magnitude of the differential pressure. This is because of partial evacuation of the region back of the nozzle end owing to entrainment by the nozzle jet of the surrounding fluid. In either case the coefficients derived from a calibration are reproducible with accuracy.

INITIAL TEMPERATURE AND PRESSURE

Initial Temperature. The system as thus far developed applies only to cases where the steam preceding the nozzle is superheated.

The temperature shall be measured by two mercury-in-glass thermometers inserted in finned wells, arranged and calibrated as specified in the A.S.M.E. Test Code, or by an equivalent system such as thermocouples or resistance thermometers competently handled. These shall be located in the pipe ahead of the nozzle at distances from it of not less than 16 D and preferably 20 D. Wells in a vertical run shall be inserted at an angle of about 45 deg. The general arrangement is shown in Fig. 5.

High-Side Gage Pressure. There shall be at least two accurate gages for measurement of the high-side static pressure connected to separate pipes for pressure measurement leading from the high-side static-pressure holes, or to independent static-pressure holes in the same plane. In the Lynn tests reported herewith deadweight gages were used as described in Appendix No. 1. However, carefully calibrated Bourdon gages or mercury manometers are permissible with especial precautions to insure that steam does not enter them and with correction for water column in the pressure pipe. All high-side gage-pressure pipes shall be filled with water before connecting the gages.

PIPES AND GAGES FOR MEASUREMENT OF DIFFERENTIAL PRESSURE

Lynn Piping Method. One scheme of piping which may be used was employed in the Lynn tests and shown in Fig. 5. It comprises $^1/_4$-in. pipes, carefully cleaned before erecting, extending horizontally from each pressure hole for a distance of 36 in. or more to insure a constant level of water. These pipes include blow-offs and unions. In each $^1/_4$-in. pipe, at least 24 in. from the main line, is a valve. Each valve shall be wide open all of the time during a test. Immediately beyond each valve is a $^1/_4$-in. tee with the branch hole pointing upward, for use in the initial filling. After filling and before starting a test, each tee is plugged. At the end of each $^1/_4$-in. pipe is a $^1/_4$-in. $\times$ $^1/_2$-in. or $^3/_4$-in. bushing or reducer, and then a $^1/_2$-in. or $^3/_4$-in. tee with the branch hole horizontal, one run hole being plugged and pointing upward and the other pointing downward, and connected by $^1/_2$-in. or $^3/_4$-in. vertical or slanting pipe to the differential U-tube. The horizontal pipe shall always be made truly horizontal by use of a good level. Slanting pipe shall have a slope of at least 10 deg. with the horizontal, and if between 10 deg. and 30 deg. shall be at least $^3/_4$-in. Between 30 deg. and vertical the pipe shall be at least $^1/_2$-in. To avoid leaks, the use of compression fittings is often desirable.

A great deal of care shall be taken to insure that the pipes for pressure measurement are filled with air-free water before a test. When the arrangement of Fig. 5 is used, filling is done from a small tank, using distilled water, which provides some head for flushing out air and which prevents air from entering with the water.

After filling, the deflections of the mercury columns shall be measured and compared with the vertical distance, if any, between the levels of the high-side and low-side static-pressure holes, divided by 12.6, the difference between the specific gravity of mercury and water. A maximum variation of 0.03 in. is allowed between the observed zero and the calculated zero due to difference of level. However, the calculated zero due to the measured difference in level, if any, shall be used in working up the tests.

When steam is on the main line, each horizontal pipe will be hot for a few inches next to the main, but each vertical or slanting pipe should always be cold, showing that it is filled with water.

If for any reason the valves in the pressure lines are shut off following a test, the plugs in the filling tees next to the valves should be removed if steam pressure is left on the line, because a leak through a valve seat on one side and a leak to atmosphere on the other side of the differential U-tube will result in a continuous displacement of the mercury column, with a possible loss of mercury and breaking of the glass if steam enters the U-tube.

Reservoir Piping Method. Fig. 3 for vertical and Fig. 4 for

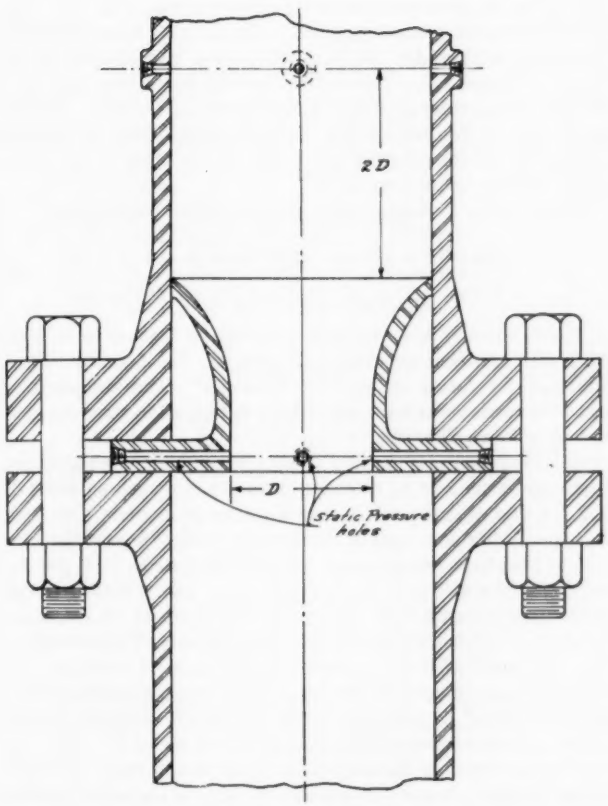


FIG. 6 PROPOSED OPTIONAL ARRANGEMENT OF NOZZLE WITH THROAT TAPS

horizontal installations show a method which may be used for connecting the nozzle to the manometer with reservoirs for condensing the steam. This arrangement is so made as to maintain equal water legs over the manometers at all times. As will be noted, the special nozzle pipe or the straight-line piping is equipped with one straight and one bent nipple for differential-pressure measurement in the case of a vertical installation. The purpose of the $^1/_4$-in. bent nipple is to bring the lower reservoir up to a level with the one connected on to the straight nipple as shown in Fig. 3. The bent nipple shall be well lagged to prevent condensation collecting therein. At the inlet end of the straight nipple shall be placed a dam having a $^1/_8$-in. hole, this hole being flush with the inside wall of the special pipe section. At the inlet end of the close nipple immediately following the bent nipple shall be placed a dam of similar construction. Following these nipples shall be placed a bull's-eye composed of crosses fitted with heavy Pyrex glass. The purpose of the bull's-eye is to visualize the exact water head on the differential manometer, and thus to render certain the fact that the water heads on the manometer are of equal height. With this method, filling is done by condensation, after blowing out the piping with steam.

Offset gate valves and reservoirs of a volume at least equal to that of the volume of the manometer tubes, complete the assembly.

One-half-inch O.D. copper tubing together with steel compression fittings may be used as the connecting tubing between the nozzle pipe and the manometer. The pitch or slope in the connecting tubing between the nozzle pipe and the manometer shall be at least one inch per foot, and preferably as nearly vertical as possible. It shall contain no sags or pockets at any point.

Piping Details. When proper precautions are taken there will be no leaks in the pressure pipe after the first assembly. Leaks in the pressure pipe shown by drops of water or a steam blow must be stopped before any tests are made.

An important point is that each pipe shall be extended downward at least 15 in. beyond the connection to the U-tube with a removable cap at the bottom. This provides that the U-tube connection is taken off the side of the pipe, and gives a dirt trap which has been found necessary in keeping the mercury columns clean. With some kinds of U-tubes, this is also very useful when cleaning the tubes without dismantling, because water can be forced through the U-tubes from the pressure pipe above, displacing the mercury and the dirt which collects in the dirt trap and is drained out by removing the cap at the bottom of the trap.

The pressure pipe may or may not contain an equalizing connection between the high and low sides. The use of such a connection may render the system more flexible.

If this connection is used during turbine performance tests, it shall be fitted with two valves on either side of a bleeder tee. This tee will be plugged when the piping is being filled, but the plug shall be removed after the valves are closed and during a test. Any leak through either valve will be disclosed by seepage of water from the open tee. Such leaks must of course be stopped.

If an equalizing connection is not used, easily arranged precautions must be taken to have the two valves in each pair of pressure pipes always opened before steam is turned on the main line, to avoid any chance of blowing mercury out of the differen-

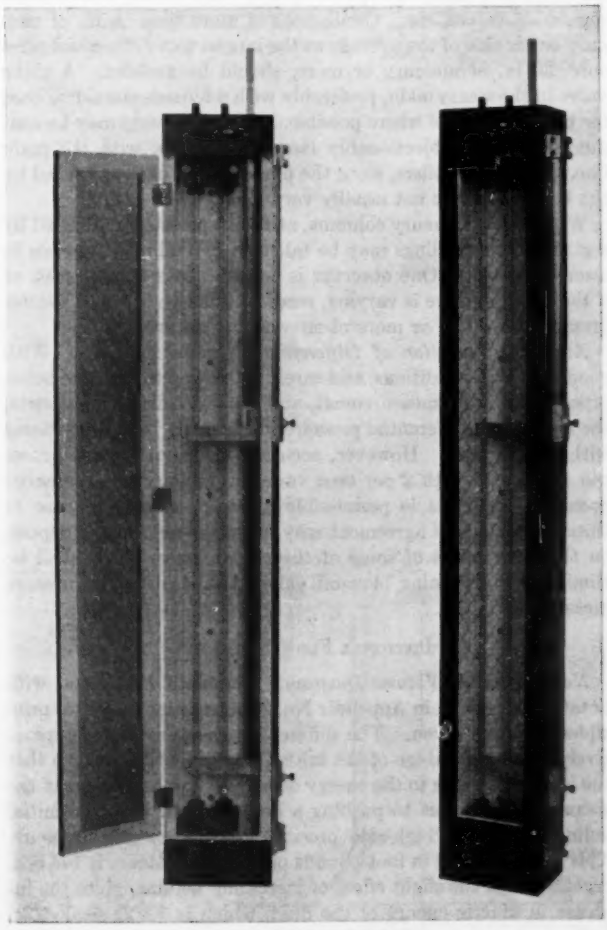


FIG. 7 DIFFERENTIAL U-TUBES FOR HIGH PRESSURE USED IN LYNN TESTS

tial U-tube. If, however, the valves have been closed for repairs of any kind, they should be opened cautiously while watching the mercury U-tube, by first cracking one, then the second, then opening the first valve a little, and then the second, etc.

Differential Mercury U-Tubes. U-tubes for this work shall be accurate to about 0.01 in. of mercury, which necessitates the use of sliders of some type. The type used in the Lynn tests is shown in Fig. 7. Definite sliders are provided with plates in front of and behind the mercury columns which can be sighted across the top of meniscus. Further details are given in Appendix No. 1. An optional equipment is a single-leg mercury column with a reservoir at the base, if the method for measuring the height of the column above the variable level in the reservoir is completely verified. Often the only way to do this will be by comparison with a two-legged U-tube such as described in Appendix No. 1, for the full-scale range.

The zero readings of the mercury U-tubes must be verified before connecting up; and again in place after filling with water.

Conduct of Test

Tests shall be conducted according to the general procedure outlined in the A.S.M.E. Turbine Test Code, particularly Pars. 67–72. Pressure fluctuations in the steam main are more objectionable in a flow-nozzle test than in a weighed-flow test because they set up oscillations of the differential mercury columns. Tests should therefore be made with the turbine governing valves blocked. Special study also should be made to reduce pressure fluctuations from other sources, such as other turbines or engines, regulating valves, etc. Oscillations of more than $^1/_2$ in. of mercury either side of the average at the largest test differential pressure, 30 in. of mercury or more, should be avoided. A globe valve in the steam main, preferably with a bypass, should be used for pressure control where possible. However, tests may be conducted without objectionably large oscillations, with the main line open to the boilers, since the pressure fluctuations caused by the boiler itself are not usually very great.

With steady mercury columns, and with pressure controlled by a globe valve, readings may be taken at five-minute intervals in a one-hour test. One observer is ample. In a shorter test, or if the main pressure is varying, readings shall be taken at shorter intervals, and two or more observers may be needed.

Allowable Variation of Differential-Pressure Readings. With good approach conditions, and careful attention to pressure holes, pipes for pressure measurement, and other details of the system, the two to four differential pressures can usually be made to agree within 1 per cent. However, accuracy sufficient for most cases can be secured with 2 per cent variation among the differential pressures, and this is permissible in many cases. In case of throat taps, lack of agreement may mean the presence of deposit on the inner edges of some of the throat holes. This shall be eliminated by opening blow-off valves in the pipes for pressure measurement.

Rational Flow Formula

Nozzle Pressure-Volume Diagram. Complete deductions, with notation, are given in Appendix No. 2, but the fundamental principles are here given. The differential pressure is a comparatively small percentage of the initial pressure. This means that the jet velocity due to the energy developed by the passage of the steam is mostly due to pushing a body of fluid with the initial volume from the high-side pressure to the low-side pressure. This energy, which in foot-pounds per pound of steam is $144\, v_1 \Delta$, together with the slight effect of increasing volume, gives the increase in kinetic energy of the fluid, which is $V^2/2g - V_1^2/2g$.

One way to present this matter is to consider the work done by the passage from high-side to low-side pressure as given by the area of a diagram on a pressure-volume plane, Fig. 16. The large rectangular area corresponds to the energy $v_1 \Delta$, and the small triangular area under the isentropic curve corresponds to the energy of expansion. This means that the major item needed in the theoretical computation of flow is the initial volume. Recent work by J. H. Keenan and the A.S.M.E. Committee on Thermal Properties of Steam[9] establishes this initial volume for steam under modern conditions of use.

Terms of the Formula. The final formula comes out as

$$w = \frac{1890 c D^2 [1 + 2\,\alpha(t_1 - t_s)]\sqrt{p_1\,\Delta\,(1 - 1.17\,\Delta/p_1)}}{\sqrt{p_1 v_1}\,\sqrt{1 - (D/D_1)^4 + 1.562\,(D/D_1)^4\,\Delta/p_1}} \quad . . [3]$$

This is a nearly exact formula and may be used up to $\Delta/p_1 = 0.10$ with an accuracy given in Appendix No. 2, under "Final Formula for Small Differential Pressure." The computation must be carried out with logarithms, a long slide rule, etc. and not by a 10-in. slide rule.

In formula [3], w is the flow in pounds per hour, c is the nozzle discharge coefficient to be taken from a curve such as curve 3, Fig. 10, and D is the cold nozzle diameter in inches. The term $2\,\alpha\,(t_1 - t_s)$ takes account of the slight increase of nozzle diameter due to the steam temperature. Under the radical, p_1 is the absolute initial static pressure in lb per sq in., and Δ is the average differential pressure in lb per sq in. The term $(1 - 1.17\,\Delta/p_1)$ is a factor which very accurately takes account of expansibility. The constant 1.17 is $3/2n$,where n is an empirical constant. In previous work n has been the exponent of the approximate isentropic curve pv^n = constant, but in our work it is merely an empirical constant whose value is determined so as to make our formula give results agreeing with values taken from Keenan's Steam Tables. In the denominator of [3], v_1 is the initial specific volume in cu ft per lb taken from the Steam Tables. The remaining radical in the denominator is a correction factor to take account of the velocity of approach in the pipe where p_1 is measured.

Appendix No. 1

EXPERIMENTAL BASIS OF THE FLOW-NOZZLE SYSTEM

The system outlined in the body of this paper is the result of recent experience gained during tests with steam nozzles which are described in this Appendix, in addition to many years' work with air as previously stated. These steam tests were made with seven nozzles. Four of them, a 4-in. × 2.20-in. (58 per cent diameter ratio), a 6-in. × 3.12-in. (51 per cent), an 8-in. × 3.67-in. (46 per cent), and an 8-in. × 4.17-in. (54 per cent), shown on Fig. 8, were tested at Lynn using throat taps. Three, a 3-in. × 1.38-in. (46 per cent), a 3-in. × 1.69-in. (56 per cent, and a 6-in. × 3.55-in. (58 per cent), shown on Fig. 9, were tested at Ohio State University with both water and steam, using low-side pipe taps.

Lynn Tests—With Throat Taps

The 4-in. × 2.20-in. nozzle was located in a downward run of pipe, with 82 in. of straight pipe ahead and 80 in. following. The pipe and nozzle were well lagged. Pressure was held at 215 lb abs by hand manipulation of a globe valve about 20 ft ahead of the nozzle. Temperatures were read on calibrated mercury-in-glass thermometers in finned wells, one 84 in. ahead of and one 42 in. after the nozzle. Superheat was taken as the steam was delivered by the boilers, varying from about 50 F to 100 F at different times. The variation during each test, however, was small. Initial pressures were first read at 4 in., and 16 in. ahead of the nozzle, and as no difference was discernible, 4 in., or $2D$,

[9] "Steam Tables and Mollier Diagram," by Joseph H. Keenan, 1930, published by the A.S.M.E.

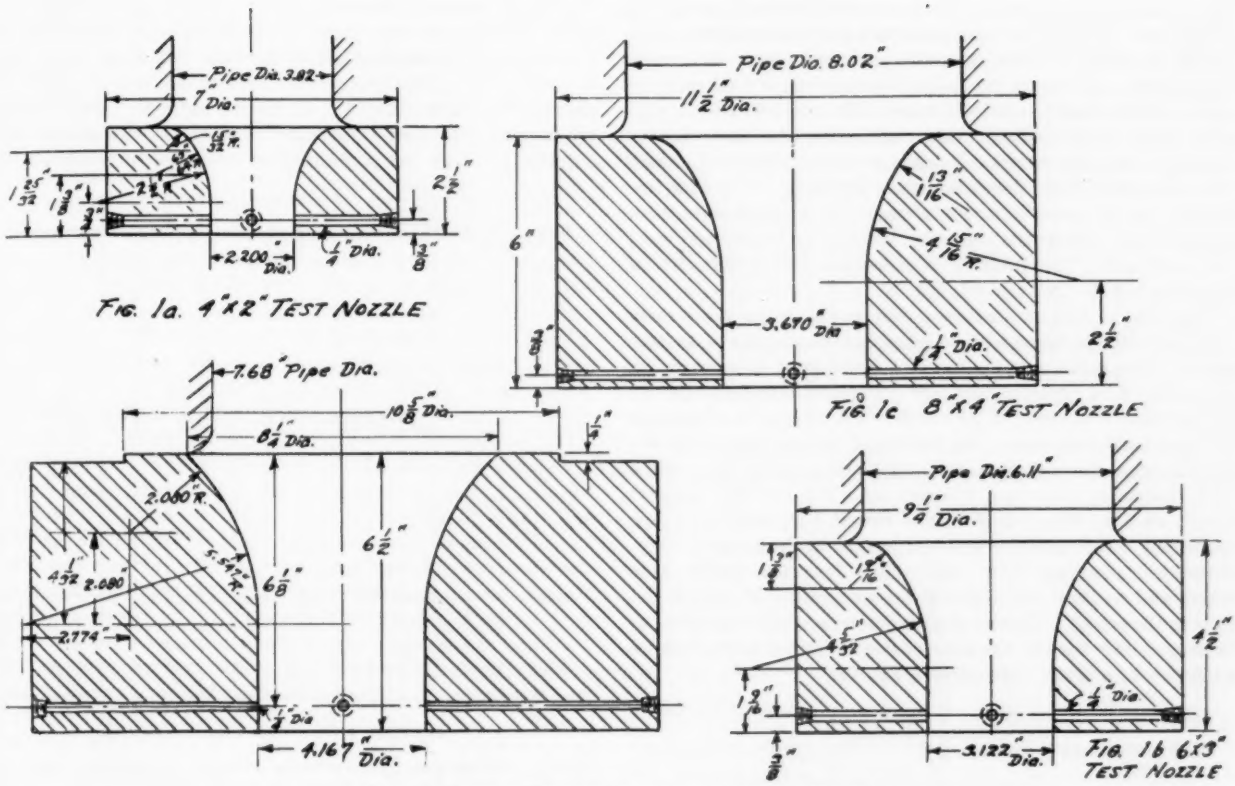


FIG. 1a. 4"x2" TEST NOZZLE

FIG. 1c 8"x4" TEST NOZZLE

FIG. 1b 6x3" TEST NOZZLE

FIG. 8 NOZZLES USED IN LYNN TESTS (THROAT TAPS)

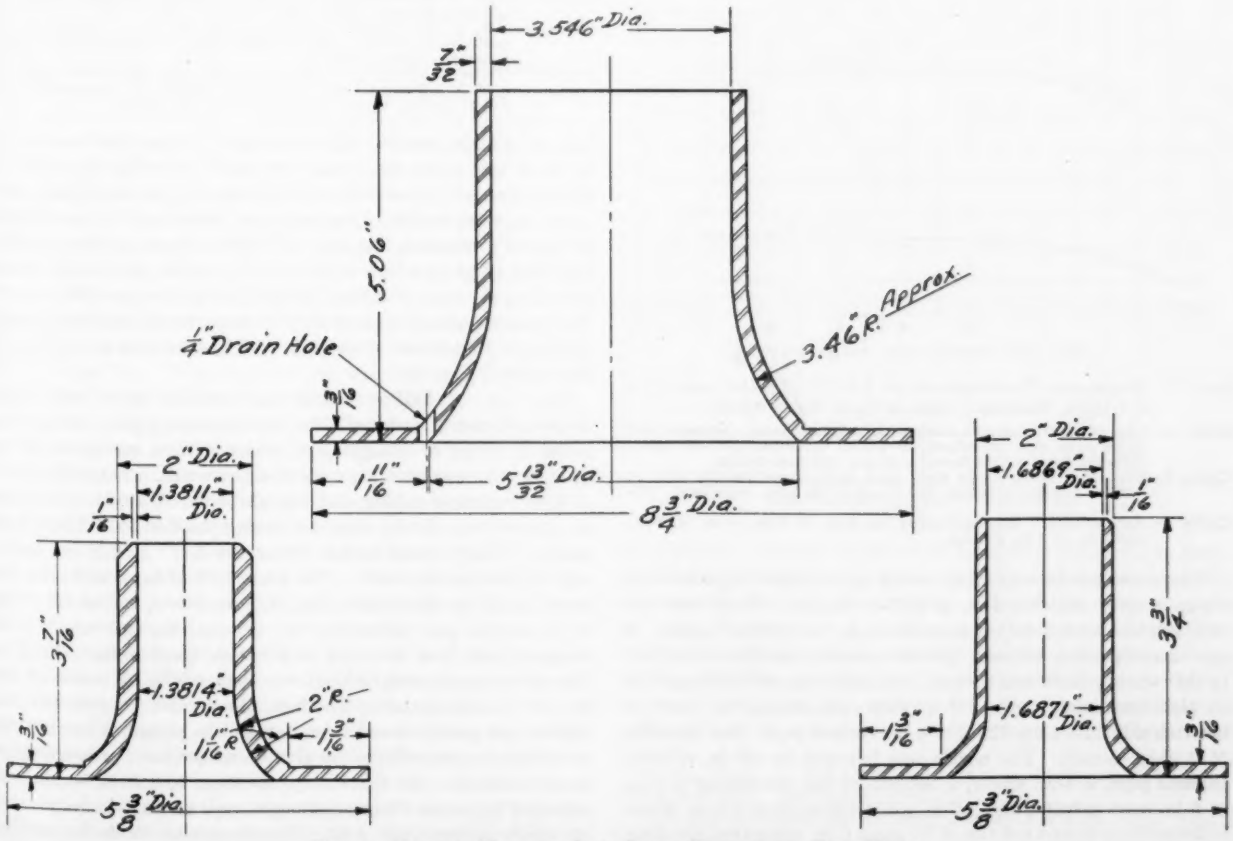


FIG. 9 NOZZLES USED AT OHIO STATE UNIVERSITY (PIPE TAPS)

was adopted as a standard. The steam after passing through the nozzle was condensed in a surface condenser and weighed. Following the flow nozzle and preceded by about 100 in. of straight pipe was a nozzle called the "plate" nozzle, designed for pressure ratios greater than the critical value. Twenty-seven tests were made with weighed flows, thus calibrating the plate nozzle. Twenty other tests were made without weighing the flow, which was calculated from the plate-nozzle readings. The tests extended over a period of about a year. In different tests, two, three, or four differential pressures, usually the latter, were read and averaged. The pressure connections were practically as shown on Fig. 5. See curve 1, Fig. 10.

The 6-in. × 3.12-in. nozzle was first tested in the steam main ahead of a turbine on which weighed-flow steam-rate tests were made. The nozzle was located in a well-lagged downward run of nominal 6-in. high-pressure pipe. Pressure was controlled by a globe valve well ahead of the nozzle. Eighty-two in. of straight pipe preceded the nozzle. At the top of the run was a 6-in. tee containing a thermometer well. Following the nozzle was a 32-in. length of 6-in. pipe, followed by a 6-in. × 8-in. adapter flange, an 8-in. elbow, and a short run of 8-in. horizontal pipe containing a thermometer well and leading to the turbine. Two differential pressures were read. Only four test points were taken, and no time was available for elimination of possible defects in the set-up. The results, therefore, are not considered as reliable as later tests on the same nozzle, described below, but are a fair check on them. See curve 4, Fig. 11.

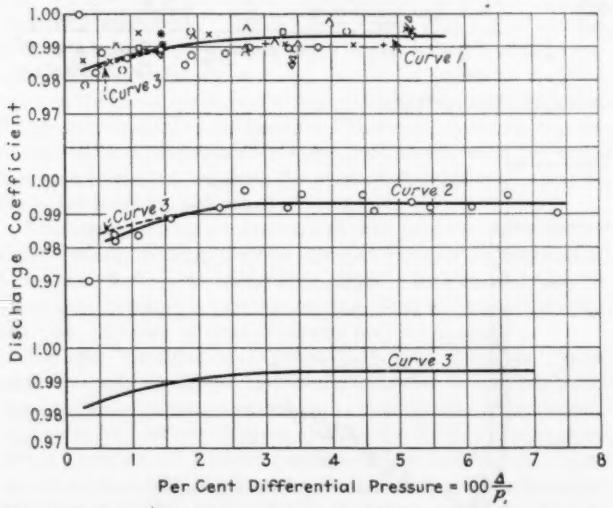


FIG. 10 DISCHARGE COEFFICIENTS OF 4-IN. × 2.20-IN. AND 6-IN. × 3.12-IN. NOZZLES—THROAT TAPS (LYNN TESTS)

Curve 1—4-in. × 2.20-in.-nozzle tests with weighed flows; pressure, 215 lb per sq in. abs; superheat, 50–100 F. Different symbols denote different dates with differences in non-essential details.

Curve 2—6-in. × 3.12-in.-nozzle tests with flows based on the 4-in. × 2.20-in. nozzle; pressure, 110 lb per sq in. abs; superheat, 105–115 F.

Curve 3—Recommended standard curve for 4-in. × 2-in., 6-in. × 3-in., and 8-in. × 4-in. nozzles.

The same nozzle was later tested in a special experimental set-up in series with the 4-in. × 2.20-in. nozzle. Flows were not weighed, but were obtained from the 4-in. × 2.20-in. nozzle. A careful comparison between the two nozzles was thus obtained. In this set-up, there was a short horizontal run of well-lagged 4-in. pipe containing a gate valve (wide open during the tests), a tee with side outlet up, 72 in. of 4-in. vertical pipe, then the 4-in. × 2.20-in. nozzle. The nozzle was followed by 56 in. of 4-in. vertical pipe, a 4-in. elbow, a horizontal run containing a 4-in. × 6-in. cone reducer, and a 6-in. globe valve, then a 6-in. elbow followed by a downward run of 55 in. of 6-in. pipe, then the 6-in. × 3.12-in. nozzle, followed by 80 in. of 6-in. pipe. There was finally an elbow, a 6-in. globe valve, and a pipe run leading to a condenser. Pressure ahead of the 4-in. × 2.20-in. nozzle was controlled by a globe valve in the main steam line about 100 ft ahead of the nozzle, and a $1^1/_2$-in. bleeder in the tee just preceding the nozzle. The 6-in. globe valve between the two nozzles was adjusted for each test point to give the proper pressure on the

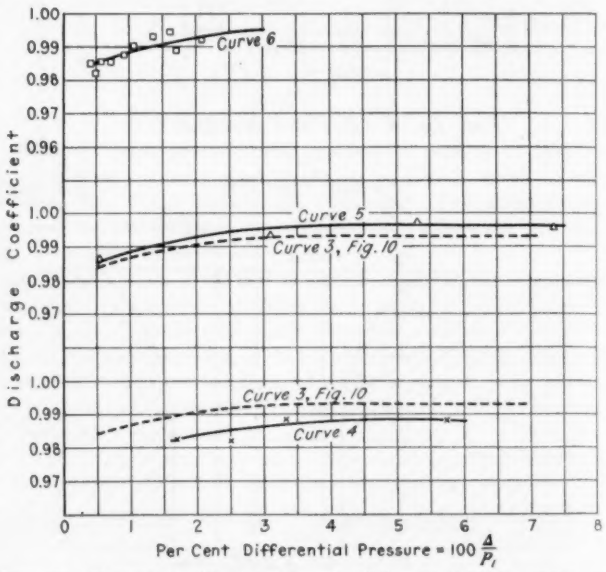


FIG. 11 DISCHARGE COEFFICIENTS OF 6-IN. × 3.12-IN., 8-IN. × 3.67-IN., AND 8-IN. × 4.17-IN. NOZZLES—THROAT TAPS (LYNN TESTS)

Curve 4—6-in. × 3.12-in. tests with weighed flows; pressure, about 210 lb per sq in. abs; superheat, 100 F.

Curve 5—8-in. × 3.67-in. tests with flows based on 6-in. × 3.12-in. nozzle and Curve 3 of Fig. 10. Pressure, 150 to 212 lb per sq in. abs; superheat, 126 F to 88 F.

Curve 6—(Same as Curve 5). 8-in. × 4.17-in. tests with flows based on 8-in. × 3.67-in. nozzle and Curve 3 of Fig. 10. Pressure, about 325 lb per sq in. abs; superheat, 85 F to 100 F.

6-in. × 3.12-in. nozzle. The amount of steam flow was varied for each test point by setting the valve following the 6-in. × 3.12-in. nozzle. Four differential pressures, 90 deg apart, were taken on each nozzle. Temperatures were read in five places: in the tee preceeding the 4-in. × 2.20-in. nozzle, in the pipe just following it, in the elbow at the top of the 4-in. pipe, in the elbow preceding the 6-in. × 3.12-in. nozzle, and in the pipe following it. The pressure ahead of the 4-in. × 2.20-in. nozzle was held at 215 lb abs and that ahead of the 6-in. × 3.12-in. nozzle at 110 lb abs. See curve 2, Fig. 10.

The 8-in. × 3.67-in. nozzle was installed in a well-lagged downward run of nominal 8-in. high-pressure pipe. It was preceded by 69 in. of straight pipe, which in turn was preceded by an 8-in. elbow, containing a thermometer well, a horizontal 8-in. × 6-in. reducing casting and 6-in. globe valve, a 6-in. elbow and an upward run of 6-in. pipe containing the 6-in. × 3.12-in. flow nozzle. Thirty-seven inches below the 6-in. × 3.12-in. nozzle was a thermometer well. The 8-in. × 3.67-in. nozzle was followed by 32 in. of straight pipe, an 8-in. elbow, and about 12 in. of horizontal pipe preceding the turbine stop valve. A thermometer well was installed just before the turbine. Flow in this case was not weighed but was determined by means of the 6 in. × 3.12-in. nozzle. The 6-in. globe valve between the two nozzles was partly closed so as to raise the pressures on the 6-in. × 3.12-in. nozzle and thus to give about the same pressure ratios on both nozzles. In this set-up, the 6-in. × 3.12-in. nozzle was preceded by about 70 in. of straight pipe and a 6-in. gate valve on which pressure was held. At the smaller flows the turbine

stop valve was partly closed so as to avoid excessive throttling on the 6-in. gate valve, since a previous experience had showed appreciable disagreement between the differential pressures due to a gate valve nearly closed. See curve 5, Fig. 11.

In the calibration of the 8-in. × 4.17-in. nozzle by comparison with the 8-in. × 3.67-in., both nozzles were set up in an 8-in. steam loop. An 8-in. horizontal globe valve, used for pressure control, was followed by an 8-in. elbow turned upward, about 93 in. of straight pipe, the 8-in. × 3.67-in. nozzle pointing up, and 24 in. of pipe after the nozzle. An elbow was followed by about 6 ft of horizontal pipe and another elbow pointing down. The downward run consisted of about 174 in. of straight pipe, the 8-in. × 4.17-in. nozzle, and 32 in. of straight pipe followed by an elbow. Four differential pressures were taken on each nozzle. Temperatures were read with mercury-in-glass thermometers inserted in mercury-filled wells, the wells being inserted through bosses in the vertical pipe at an angle of about 45 deg. The thermometer ahead of the 8-in. × 3.67-in. nozzle was 48 in. from the nozzle, and that ahead of the 8-in. × 4.17-in. nozzle was 67 in. from it. See curve 6, Fig. 11.

Discharge Coefficients With Throat Taps—Lynn Tests. The discharge coefficients for the 4-in. × 2.20-in. nozzle are shown as curve 1 on Fig. 10. The plotted points are the results of tests covering a little over a year, during which time several changes were made in the set-up and different ideas described elsewhere were tried out. Separate symbols have been used to represent the tests of each group. Each test in which the dam, shown in Fig. 12, was at least eight pipe diameters away from the nozzle, or removed entirely, has been plotted.

As was expected, based on experience with air nozzles, the average coefficient was very nearly 0.99 over a wide range of pressure ratios. All but one of the points will be seen to fall within a band of plus or minus one per cent from the average. The lowest flows represent differential pressures as small as one inch of mercury. The actual readings were about 2 in. due to the zero deflection, which increased the reading. At very small deflections the mercury columns were practically motionless, while with large deflections there was oscillation up to about one per cent. This tended to equalize the accuracy of the readings at small flows so that there is scarcely any more scattering of points at small than at large flows. Damping owing to a restriction at the bottom of the U-tube, was not considered necessary.

A few readings were taken of impact pressure with a standard impact tube at the center of the nozzle mouth and pointing upward. This was read as a differential against the throat pressure. There was about three inches of vertical length, which caused the readings to be ragged because of boiling in the tube and an unknown water leg. This impact tube was therefore abandoned and no coefficients so derived have been plotted.

The later tests on the 6-in. × 3.12-in. nozzle with the experimental set-up are given as curve 2 of Fig. 10. They show practically the same discharge coefficients as the 4-in. × 2.20-in. nozzle except at very small differential pressures where the discharge coefficient is very slightly lower.

Curve 3 of Fig. 10 has been made up from curves 1 and 2, giving slightly more weight to curve 1. Curve 3 has been shown dotted on curves 1 and 2 of Fig. 10, and on curves 4 and 5 of Fig. 11, for comparison.

The original weighed-flow tests on the 6-in. × 3.12-in. nozzle and the 8-in. × 3.67-in. tests obtained by comparison with the 6-in. × 3.12-in. nozzle, both in commercial set-ups, are shown as curves 4 and 5, Fig. 11. The test points were obtained with but two differential U-tubes, and for this and similar reasons were not given the same weight as the tests shown on Fig. 10. They are, however, a fair check on the standard coefficient curve 3 of Fig. 10.

The comparative tests on the 8-in. × 3.67-in. and the 8-in. × 4.17-in. nozzles are shown on curve 6, Fig. 11. The curve drawn through the points is the same as curve 5. Flows were calculated from the readings of the 8-in. × 3.67-in. nozzle and curve 5. The 8-in. × 4.17-in. coefficients were based on the readings of the 8-in. × 4.17-in. nozzle and the flows calculated from the 8-in. × 3.67-in. nozzle. The tests showed no perceptible difference between the two nozzles, although the ratio of nozzle to pipe diameter was slightly different, being 0.46 for the 8-in. × 3.67-in. nozzle and 0.54 for the 8-in. × 4.17-in.

Ohio State University Tests—With Low-Side Pipe Taps

In these tests, the 3-in. × 1.38-in., the 3-in. × 1.69-in., and the 6-in. × 3.55-in. nozzles (Fig. 9) were tested with both steam and water. The set up and tests are described in two papers, the 6-in. × 3.55-in. in a paper by Professors Beitler and Bucher,[10] and the 3-in. nozzles in a paper by Messrs. Beitler, Bucher, and Barnes.[11]

In the steam tests the 3-in. × 1.38-in. and the 3-in. × 1.69-in. nozzles were preceded by about 135 diameters of straight pipe

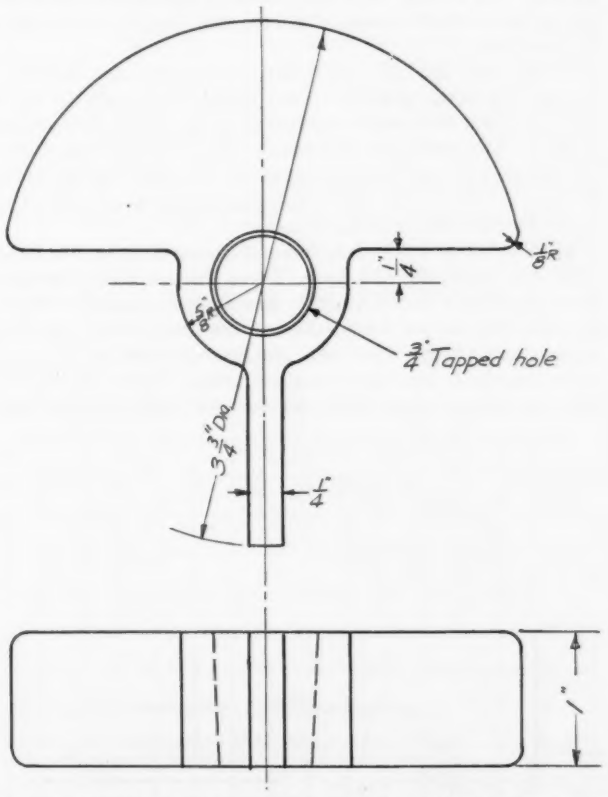


Fig. 12 Dam Used in Tests of 4-In. × 2-In. Nozzle to Simulate Pipe Irregularity (Lynn Tests)

and the 6-in. × 3.55-in. by about 64 diameters, which was in turn preceded by straightening vanes and the hand pressure control, an angle globe valve. The static high-side pressure was measured one pipe diameter ahead of the nozzle, by means of mercury manometers which were connected to the steam line through reservoir assemblies described in the papers and similar to those shown in Figs. 3 and 4. A 50-in. manometer was used for the tests at 19 lb abs, and a special mercury manometer of approxi-

[10] "The Flow of Fluids Through Orifices in Six-Inch Pipes," Trans. A.S.M.E., 1929, paper HYD-52-7a.

[11] "A Study of Primary Metering Elements in 3-In. Pipe," read before the A.S.M.E., Dec., 1931.

mately the same construction but capable of measuring pressures up to 117 lb per sq in. for the tests at 65 lb abs.

The low-side pressure connection was located in the pipe wall in accordance with the method shown in Fig. 1*a*. Owing to the great length of the preceding pipe, and the fact that the low-side taps of this method show a high degree of uniformity, only one differential pressure was taken. The differential pressure was measured by a mercury-filled inverted water manometer, approximately 10 ft in length. The top of the manometer was connected to a high-pressure air line, so that the manometer pressure could be adjusted to approximately the same pressure as that in the steam main. The bottom of the manometer was connected to the static-pressure connections in the steam pipe by means of $^1/_2$-in. O.D. copper tubing and special fittings.

The temperature of the steam was controlled by the rate of firing of an independent superheater. Temperature was measured by a mercury-in-glass thermometer in a finned A.S.M.E. well located 5 pipe diameters ahead of the nozzle. This well could be withdrawn while pressure readings were being taken. Mercury was usually the transfer medium in the well, but cylinder oil was sometimes used. Solder was used at the highest temperatures.

In the steam line after the nozzle, a valve was used to control the pressure in the test line. After passing the nozzle the steam was condensed in a surface condenser having 465 sq ft of cooling surface. The condenser was fitted with a wet vacuum pump. The condensate was first discharged into an open overhead tank for removal of air, and then weighed by means of double weigh tanks on calibrated platform scales.

The steam line was usually warmed up for three or four hours before the beginning of a test. When the line was thoroughly warmed, the flow was adjusted to give approximately the desired pressure drop across the nozzle, and the line pressure adjusted to the desired value. Not until the pressures and temperatures were stabilized, were any readings taken. After the desired pressures were set and the temperature was being adjusted, care

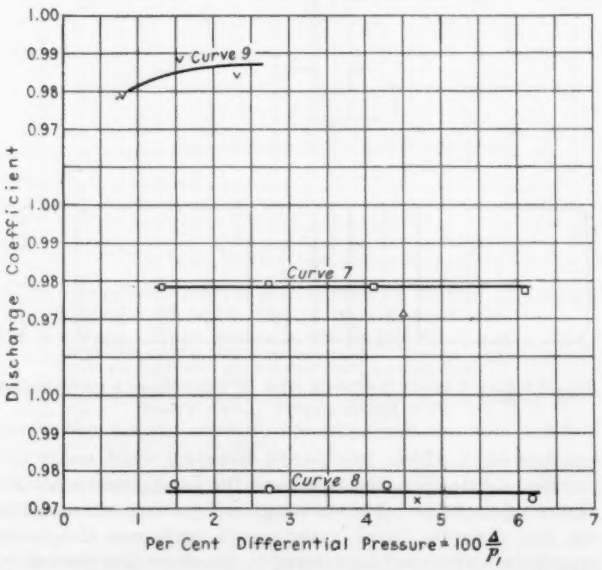


FIG. 13 DISCHARGE COEFFICIENTS OF 3-IN. × 1.38-IN. (46 PER CENT), 3-IN. × 1.69-IN. (56 PER CENT), AND 6-IN. × 3.55-IN. (58 PER CENT) PIPE TAPS (OHIO STATE UNIVERSITY TESTS)

Curve 7—3-in. × 1.38-in. tests with weighed flows. Values at 65 lb per sq in. abs pressure and 49 F superheat indicated by squares; at 19 lb per sq in. abs and 146 F, by triangle.

Curve 8—3-in. × 1.69-in tests with weighed flows. Values at 65 lb per sq in. abs and 76 F superheat indicated by circles; at 19 lb per sq in. abs and 180 F superheat, by cross.

Curve 9—6-in. × 3.55-in. tests with weighed flows. Pressure, 65 lb per sq in. abs: superheat, 153 F.



FIG. 14 8-IN. × 4.17-IN. STEAM-FLOW NOZZLE INSTALLED, WITH LAGGING REMOVED

was taken to remove all air from the manometer connections, and the water in the manometer tubes was cooled to approximately room temperature. The minimum length of run was 30 min of steady readings and the maximum 90 min. In each case the length of the run was sufficient to get at least two tanks of water, each tank containing 225 lb. At least 20 readings were taken during each run. In the longer tests, readings were taken at two-minute intervals. Two observers taking all readings simultaneously were used in all these tests.

Discharge Coefficients With Pipe Taps—Ohio State University Tests. The discharge coefficients for the three nozzles tested are shown in Fig. 13. It will be noted that the coefficients are somewhat lower than those obtained in the Lynn tests. This is due to the use of low-side pipe taps instead of throat taps, as explained earlier in the paper under "Nozzle Discharge Coefficient." Regardless of the absolute value of the coefficients, however, it is felt that they are reproducible within the necessary limits of accuracy.

DETAILS OF THE SYSTEM

Pipe Irregularity—Tests With a Dam. During the Lynn tests on the 4-in. × 2.20-in. nozzle, a dam shown in Fig. 12 was inserted into the pipe ahead of the nozzle through the tee at the top of the run, and readings were taken with the dam at varying distances from the nozzle. The dam closed off one side of the pipe completely, the open side having very nearly one-half the pipe area. During these tests differential pressures were taken from three of the high-side static-pressure holes to three low-side static holes, and from an ordinary and a specially designed impact tube with no vertical leg at the nozzle mouth to the No. 4 low-side static hole. The open side of the dam was directly over the No. 4 low-side hole. With the static differentials the dam caused no perceptible change in the coefficients until it was brought nearer than about 32 in., or 8 pipe diameters. When the dam was brought quite near the nozzle, the coefficients increased to values well over unity, which is easily explained since it is practically certain that the jet leaving the dam does not completely fill the pipe between the dam and the nozzle. Under this condition the average differential pressure is somewhat too small, some of the high-side static-pressure holes being in an eddy and reading lower than the actual pressure in the high-velocity jet, and the actual velocity of approach being considerably higher

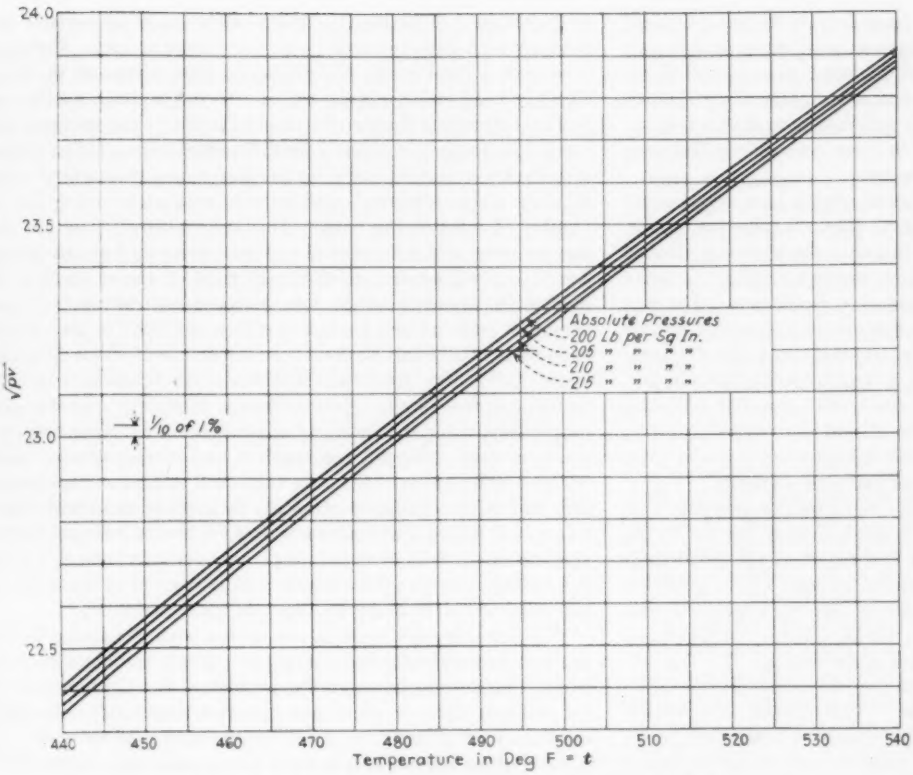


FIG. 15 $\sqrt{pv}$ PLOTTED AGAINST TEMPERATURE IN DEG F
(p = pressure, lb per sq in. abs.; v = volume, cu ft per lb; values from Keenan's "Steam Tables," 1930 edition.)

than the velocity calculated on the assumption that the pipe is full.

With the impact-tube readings the coefficients began to be affected when the dam was about 40 in. from the nozzle, and the coefficients acted irregularly as the dam approached the nozzle, first increasing slightly and then dropping off rapidly. This indicated that when the dam was brought too close the distribution of velocity over the nozzle throat was not uniform, and that the impact tube was affected by stray jets. Since it is more difficult to install than the ordinary static differential-pressure pipes, and is also subject to error due to an uncertain water leg in some cases unless specially designed, the impact tube is not recommended for use with this system. The use of static pressures requires that the velocity of approach be taken into account, which is easily done with the formula given in Appendix No. 2.

The concentration of the flow along one side of the pipe by the dam is considered to be as severe a condition as any ordinary unsymmetrical irregularity such as an elbow or tee, hence the rule of 16D following such irregularities.

A special test showed that a globe valve caused very little disturbance even when nearly closed, hence it is to be treated as an ordinary unsymmetrical irregularity. A gate valve nearly closed, however, caused the two differentials of a pair 180 deg apart to show as much as 5 per cent difference, even when followed by about 20D of straight pipe. A gate valve followed by a bend (in this case a tee) caused from 8 to 12 per cent variation between two differentials and produced an unstable condition. With the same gate valve wide open, the variation was reduced to about 1.5 per cent. A gate valve partly closed is therefore to be avoided unless very remote from the nozzle, or unless followed by special baffles or a steam separator or similar chamber.

Guide Vanes to Prevent Rotary Motion. These are often prescribed in flow-nozzle work in the straight pipe preceding the nozzle. However, it is believed that if the rules here given for lengths of straight pipe are followed, and if gate valves partly closed are avoided, rotary motion or whirls will not often be encountered. Such motion is probably due to some peculiar combination of restrictions and bends, combined with low pipe velocities. Its occurrence will be indicated by results such as large disagreement between the differential pressures, or absurd discharge coefficients. Hence guide vanes may be omitted unless trouble is experienced.

Velocity Reconversion. The present tests show that a considerable amount of the differential pressure is recovered by reconversion of the jet velocity into pressure in the pipe beyond the nozzle. Tests on the 6-in. × 3.12-in. nozzle showed practically 30 per cent recovery at a distance of 12 in. or 2 pipe diameters after the nozzle face, and 40 per cent at 22 in. or 3.7 pipe diameters. At 32 in. or 5.3 diameters the recovery was about 40.5 per cent. It

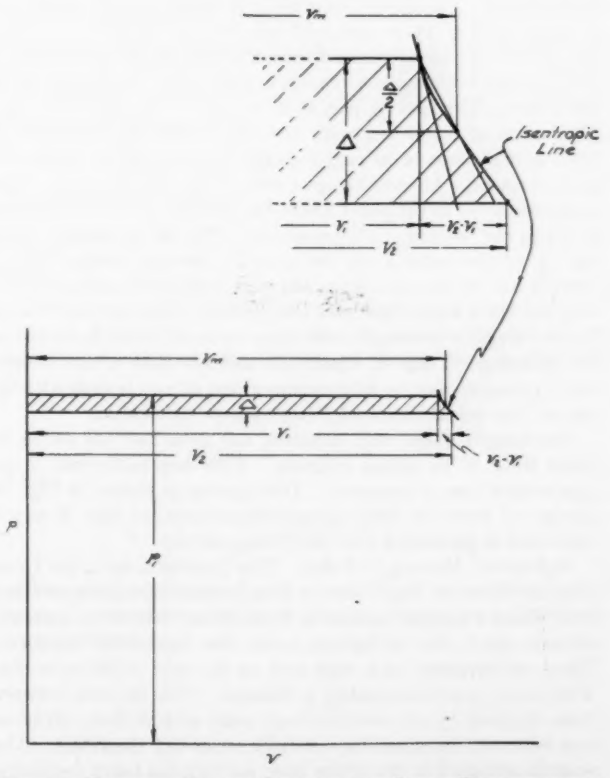


FIG. 16 PRESSURE-VOLUME DIAGRAM FOR STEAM EXPANSION IN A FLOW NOZZLE

is therefore recommended that at least four diameters of straight pipe be placed after a nozzle whenever possible, in which case approximately 40 per cent of the differential pressure will be recovered. This amount seemed to increase slightly at small flows, rising to about 43 per cent. The full-load flow in the 6-in. × 3.12-in. nozzle tests corresponded to 5 per cent differential pressure, so that the net loss was 3 per cent.

The straight pipe beyond the nozzle goes a long way toward serving the purpose of the divergent part of a venturi meter. The jet leaving the nozzle at first fills only the central portion of the pipe. It gradually expands and eventually fills the entire pipe, gaining in pressure as the velocity diminishes. The fact that reconversion is practically complete at a distance of four pipe diameters means that the effect of the symmetrical irregularity formed by the nozzle itself is not noticed after this distance. This shows that the rule given that a nozzle following an ordinary symmetrical irregularity should be preceded by about four pipe diameters or eight nozzle diameters of straight pipe, applies to a restriction of about half the pipe diameter.

Deadweight Gages. The necessity for frequent calibration of an ordinary gage by means of a deadweight gage has led to the use in the Lynn tests described in this paper, of the deadweight apparatus itself for the measurement of the initial pressure. Early research work of this character was accomplished with actual pressure measurement by use of ordinary deadweight-gage testers primarily designed for gage testing. To this apparatus was added an upper stop to prevent excess pressure from lifting the piston from the cylinder. This proved so successful that other forms of home-made apparatus have been used, finally leading to a design of deadweight pressure gage shown set up for a test in Fig. 14. Apparatus of this type has been used for all steam-research work for the last twenty-five years in the General Electric Company's Lynn Works, and has proved to be very satisfactory. The apparatus is made by mounting in stands such as shown in Fig. 14, pistons and cylinders of $^1/_8$ sq in. area purchased from a reputable manufacturer of deadweight-gage testers. In operation the deadweight gage is piped up to the point where the pressure is measured, just as in the case of the Bourdon gage. The reservoir connected with the deadweight-gage cylinder is filled with a grade of oil suited to the temperature at which the gage operates. At room temperature a light grade is used. The pipe connected to the point where the pressure is to be measured is led to the bottom of this reservoir. The oil, of course, keeps on top of the water from the pipe for pressure measurement. During use, an operator spins the gage with one hand and operates the valve which regulates the pressure with the other. The heavier weights are made with large diameter so as to facilitate the spinning. Years of experience indicate that a very nearly exact pressure may be held during a test of any length without any of the uncertainties of Bourdon-gage calibration.

Deadweight gages with constant use soon leak oil owing to slight wear of the piston cylinder. This does not result in an appreciable loss of accuracy. The apparatus shown in Fig. 14 has an oil reservoir with appreciable volume, so that it would only need to be refilled after fairly long periods.

Differential Mercury U-Tubes. The U-tubes used in the Lynn tests are shown in Fig. 7, and as already mentioned, are provided with sliders which have plates in front of and behind the mercury column which can be sighted across the top of the meniscus. These are mounted on a steel post on the side of the tube box. They are of course accurately horizontal. The distance between them is given by an accurate brass scale with 0.10-in. divisions and with 0.01 in. given by a vernier on one of the riders. The scale is attached to the lower rider so that the lower reading is always zero. The glass tube of the U-tubes has an internal bore of $^1/_4$ in., so that with clean glass there is practically no error due to uncertainty of capillarity, which often causes appreciable deflections with glass tubes of $^1/_8$ in. bore, more or less. The glass tubes are a good grade of boiler-gage glass about 48 in. long. This type of U-tube is much superior to the common variety requiring estimation by eye of a scale adjacent to the mercury column. An important item of the differential gage is the provision of a stout wooden box enclosing the glass tubes, with a front door of thick glass reinforced with wire netting and with a liberal number of holes in the back. This is for protection of the observers from scalds in case the glass breaks and steam issues. The $^1/_4$-in. valves close to the main must of course be shut off quickly in case of breakage, but the water contents of the pipes would be emptied and enough steam might issue to give a bad burn before this could be done. In the course of about 20 years' work with steam differential U-tubes, such breakage has only occurred a few times. Nevertheless it is always possible, and proper regard for the safety of the observers demands the precautions cited. Careful construction and careful manipulation of the stuffing boxes of the glass tubes will minimize glass breakage, and with reasonable attention to prevention of leaks elsewhere, will insure that cold water and not steam is always in the glass tube. It is to be noted that it is never safe to clean a glass tube with a wire or other piece of metal, inserted in the inside of the tube, as this is likely to cause the glass to shatter.

The dirt trap, described in connection with the piping to the U-tube, assists greatly in keeping the mercury column clean. It is also desirable to have the upper block of the U-tube made of chrome-iron alloy or other non-corrosive material. Otherwise, small amounts of rust from it will appear above the mercury. In spite of these precautions a little dirt occasionally accumulates in the water immediately above the mercury. This may be removed by forcing water through the U-tube from the pressure pipe above, displacing the mercury and dirt, which collect in the dirt trap and are drained out by removing the cap at the bottom of the trap. The U-tube is then refilled with mercury through holes immediately above the glass, which are afterward plugged. The pipe is then refilled with water.

Appendix No. 2

FORMULAS FOR STEAM FLOW

Notation

w = flow, lb per hr

p_1 = high-side static pressure, lb per sq in. abs

Δ = differential pressure, lb per sq in., or difference between p_1 and absolute low-side static pressure
For most work, Δ/p_1 will not exceed about 0.10. However, where the drop is not objectionable, and the Exact Formula given later is used, it may have any value up to about 0.56.

v_1 = specific volume at high-side conditions, from Keenan's 1930 Steam Tables, cu ft per lb

v_2 = specific volume at low-side conditions with isentropic expansion

t_1 = high-side temperature, deg F

T_1 = high-side temperature, deg F abs = $t_1 + 459.6$

t_s = standard temperature at which micrometer for measuring nozzle diameter is correct, usually 68 F

α = linear coefficient of expansion of nozzle per deg F = 0.0000065 for steel

D = nozzle diameter in inches as average of a number of careful measurements with a micrometer which is at approximately the temperature of the nozzle

D_1 = diameter in inches of pipe preceding nozzle where p_1 is measured

n = exponent for isentropic expansion of superheated steam,

corrected empirically for formulas [3] and [4]. n is to be taken as 1.28

V = jet velocity at nozzle exit, ft per sec

V_1 = approach velocity in pipe preceding nozzle, ft per sec

c = coefficient of discharge. With throat taps this may be taken as 0.99, unless a more accurate value is available

g = acceleration of gravity, to be taken as the standard value, 32.17 ft per sec. per sec. The actual value at any probable test location in the U. S. differs by a negligible amount. For instance, the variation at Lynn, Mass., where the tests were made, would affect the flow by about $^1/_{100}$ of 1 per cent. For any local value g_L, multiply 1890 in the formula by $\sqrt{g_L/32.17}$

Final Formula for Small Differential Pressure, up to $\Delta/p_1 = 0.10$

$$w = \frac{1890cD^2\,[1 + 2\alpha\,(t_1 - t_s)]\,\sqrt{p_1\Delta\,(1 - 1.17\,\Delta/p_1)}}{\sqrt{p_1v_1}\,\sqrt{1 - (D/D_1)^4 + 1.562\,(D/D_1)^4\,\Delta/p_1}} \quad [3]$$

For cases where Δ/p_1 is less than about 0.03, it is sufficiently accurate to neglect small terms, which gives

$$w = \frac{1890cD^2\,[1 + 2\alpha(t_1 - t_s)]\,\sqrt{\Delta(p_1 - \Delta)}}{\sqrt{p_1v_1}\,\sqrt{1 - (D/D_1)^4}} \quad [4]$$

This formula with $[1 + 2\alpha(t_1 - t_s)]$ taken as 1.00 may be used to determine the nozzle diameter in the preliminary work of laying out a test with Δ/p_1 up to 0.10.

These formulas are only for initially superheated steam. If this initial superheat is slight, it makes no difference because "undercooling" always occurs. That is, the expansion continues to follow the superheated-steam laws beyond the saturation line, and condensation does not occur when the saturation temperature is reached.

The value of $\sqrt{(p_1v_1)}$ is to be found by graphical interpolation from steam tables such as Keenan's, using a curve such as Fig. 15. The quantity $\sqrt{(pv)}$ is computed for a number of even values of pressure for values of volume given in the steam tables for a range of temperature and plotted against t. In any set of tests the initial conditions will vary somewhat from test to test, so that the curve such as Fig. 15 should cover the entire region within which p_1 and t_1 occur, with a margin on either side to give fair curves. It might be necessary to cross-plot values of $\sqrt{(pv)}$ against p for a few values of t to insure proper spacing of the curves of Fig. 15 for the various even values of p.

It has been determined by a careful investigation of volume relations in Keenan's 1930 Tables that for steam pressures less than 600 lb per sq in., the error of formula [3] is, for differential pressures of 5 per cent and less, about 0.1 of one per cent, and for 10-per cent differentials about 0.2 of one per cent; for pressures of 600 to 2000 lb the maximum error is, for 5-per cent differentials, about 0.2 of one per cent, and for 10-per cent differentials, about 0.5 of one per cent. In general the maximum errors occur near the saturation line. For high superheats the errors are smaller.

Types of Steam Formulas. The ordinary method of handling velocity due to expansion of steam is by means of a Mollier diagram on which values of total heat are read at points corresponding to the low-side and high-side pressures at the same entropy. For the small differential pressures of the present work, it is impossible to read any existing diagram with anything approaching accuracy, so that this usual method fails completely. This is the reason we must use the above formula [3]. However, this formula is limited to values of Δ/p_1 up to about 10 per cent. There is, therefore given later, beginning with the paragraph "Plan for Deduction of Exact Formula," a somewhat laborious second method of computing theoretical flow which involves fewer approximations than the method of formula [3], by making use of values from the steam tables. It is believed that this method gives accuracy with flows of any magnitude. It also gives the means by which the accuracy of formula [3] was established.

Plan for Deduction of Formula [3]. As pointed out in the paper in the paragraph "Nozzle Pressure-Volume Diagram," an approximation is used to determine the small triangular area under an isentropic curve of the pressure-volume diagram of Fig. 16 which gives no appreciable error for values of Δ/p_1 up to about 0.10. There also must be used in the formula a proper value of v_2, the final specific volume. A very common approximate procedure in this and similar matters is to assume that the isentropic expansion of steam is given by the exponential law,

$$pv^n = c$$

This leads to a well-known, rather complicated formula given later as [6], that involves fractional exponents and was first proposed by Saint Venant. There are also well-known methods of making further approximations, using the first term of an expansion in series. All of these approximations, however, leave a doubt as to the accuracy of the final formula.

We therefore next proceed to deduce from first principles an "Approximate Formula from the Exponential Law." It turns out that this is identical with the previously known formula based on approximations in Saint Venant's formula [6]. However, so many approximations are made both with the original assumption that $pv^n = c$ and in the further deduction, that we cannot be sure that we have a legitimate formula. We therefore use the formula finally found merely as an empirical form, and then proceed to find a value for n which we regard merely as an empirical constant whose numerical value is selected to give agreement with the steam tables.

Deduction of Approximate Formula From the Exponential Law. The mathematical work for this procedure will now be given. In order to find the slope of the tangent at the top of the triangular area, we differentiate

$$pv^n = p_1v_1^n$$

which gives

$$\frac{dv}{dp} = -\frac{1}{n}\,p_1^{1/n}v_1p^{-1-1/n}$$

The minus sign indicates that v decreases as p increases. This value of dv/dp with $p = p_1$ gives the slope at the top of the triangle. Making the approximation that v_2 also is given by this slope.

$$\frac{v_2 - v_1}{\Delta} = \frac{1}{n}\,\frac{v_1}{p_1}$$

Hence

$$v_2 - v_1 = \frac{v_1\Delta}{np_1} \text{ and } v_2 = v_1\left(1 + \frac{\Delta}{np_1}\right)$$

This gives, as the work in foot-pounds, corresponding to the triangular area computed from the tangent to the pv^n curve at the upper corner,

$$144\,\frac{\Delta^2v_1}{2np_1}$$

By adding to this the area of the rectangular portion of Fig. 16 or 144 $v_1\Delta$, we have as the total energy of the isentropic expansion, which, as previously specified, is the change in kinetic energy,

$$\frac{V^2}{2g} - \frac{V_1^2}{2g} = 144\left(v_1\Delta + \frac{\Delta^2 v_1}{2np_1}\right)$$

from which it follows that

$$V^2 = \frac{2g \times 144 v_1 \Delta\left(1 + \frac{\Delta}{2np_1}\right)}{1 - (V_1/V)^2}$$

Since the flow is the same in the jet and the approach pipe,

$$\frac{V_1}{V} = \left(\frac{D}{D_1}\right)^2 \frac{v_1}{v_2}$$

Using the value of v_2 approximated above from the slope,

$$\frac{v_2}{v_1} = 1 + \frac{\Delta}{np_1}$$

We can expand this by the binomial theorem and neglect high powers of the small quantity Δ/p_1 so that

$$(v_1/v_2)^2 = (v_2/v_1)^{-2} = 1 - \frac{2\Delta}{np_1}$$

Substituting this in the value of V_1/V just given,

$$1 - \left(\frac{V_1}{V}\right)^2 = 1 - \left(\frac{D}{D_1}\right)^4 \left(1 - \frac{2\Delta}{np_1}\right)$$

This we later substituted in the above value of V^2.

The flow is given by the expression

$$w = \frac{3600\pi c}{4} \frac{D^2}{144} \frac{V}{v_2}$$

(This assumes that the nozzle has the diameter D, which is not true when it is hot. A correction for this is made in the final formulas.)

From the previous values of V and v_2 we may write

$$\frac{V}{v_2} = \frac{12\sqrt{2g\Delta}}{\sqrt{1 - \left(\frac{V_1}{V}\right)^2}} \sqrt{\frac{v_1\left(1 + \frac{\Delta}{2np_1}\right)}{v_1^2\left(1 + \frac{\Delta}{np_1}\right)^2}}$$

We may expand the denominator by the binomial theorem, noting that Δ/p is a small quantity whose higher powers may be neglected, so that

$$\left(1 + \frac{\Delta}{np_1}\right)^{-2} = 1 - \frac{2\Delta}{np_1}$$

By neglecting squares of small quantities we may write the product of this with the numerator,

$$\left(1 + \frac{\Delta}{2np_1}\right)\left(1 - \frac{2\Delta}{np_1}\right) = 1 - \frac{3}{2}\frac{\Delta}{np_1}$$

By substituting this in the preceding expression for w, we arrive at a formula for flow as the result of making a number of approximations and using the exponential law namely,

$$w = \frac{3600\pi c D^2 \sqrt{2g}\sqrt{\Delta\left(p_1 - \frac{3\Delta}{2n}\right)}}{48\sqrt{p_1 v_1}\sqrt{1 - (V_1/V)^2}} \quad \ldots\ldots [1]$$

This is identical with a formula often deduced by making approximations in the well-known theoretical formula given later as [6] and involving fractional exponents, and which was first deduced by Saint Venant from $pv^n = c$.

In this formula we have used as the value of the small triangular area the value under the tangent at the upper corner, which is obviously less than the area under the isentropic curve. We have also used as the value of v_2 the value to the foot of the tangent which is somewhat less than the actual value to the intersection with the isentropic curve. It would seem that if we could use a value of n which differed slightly from the value corresponding to the index of the isentropic curve, we might make corrections for these inaccuracies and obtain a formula somewhat more exact than [1], with n as the exponent of the isentropic curve. This equation would have the same form as [1], but would use for n such an empirical value as would make the formula more nearly accurate than when the isentropic value of n was employed.

Formula With Steam-Table Volumes. In order to accomplish this we next proceed to find a formula a little more accurate than [1]. In this we use the exact value of v_2 instead of any approximation thereto. We also introduce the specific volume v_m, which is the volume on the same isentropic curve for a pressure drop $\Delta/2$. We then have as a very close expression for the area in Fig. 16,

$$\frac{V^2}{2g} - \frac{V_1^2}{2g} = 144\, v_m \Delta$$

In order to be more nearly exact, this uses as the triangular area under the curve the area under a straight line through the midpoint of the curve, instead of that under a tangent at the upper corner. We also assume the exact v_2 and not that given by the slope of the tangent.

It follows, as in the previous deduction, that

$$w = \frac{3600\pi c D^2 \sqrt{2g v_m \Delta}}{48 v_2 \sqrt{1 - (V_1/V)^2}} \quad \ldots\ldots [2]$$

This involves very little approximation. We proceed now to use it on the basis of values of v_2 and v_m from the steam tables to determine an empirical value of n in a formula of the from of [1].

Empirical Determination of "n." Equating [1] and [2] gives

$$\frac{v_m}{v_2^2} = \frac{1 - \frac{3}{2n}\frac{\Delta}{p_1}}{v_1}$$

Solving for n gives a value which makes [1] give the same result as the nearly exact formula [2],

$$n = \frac{\frac{3\Delta}{2p_1}}{1 - \frac{v_1 v_m}{v_2^2}}$$

Actual values of volumes from Keenan's Steam Tables for even pressures were plotted against entropy, and the values of v_2 and v_m read at constant entropy on curves such as Fig. 15. From these, values of n were computed from the above equation, for different values of Δ/p_1 at different parts of the steam table. It was found that n was very nearly independent of Δ/p_1, up to about 0.10, and differed but little in various parts of the steam table, so that a value of n in formula [1] of 1.28 represented to within a few tenths of one per cent the closely approximate formula [2]. As stated, this comparison involves no assumption that this value of n fits the relation pv^n = constant. However, the value of n corresponding to this equation was found as a separate item of the previous work, and was actually found to be

practically the same as the n which gave equality between [1] and [2]. This means that the approximations made in deducing Equation [1] by either of the methods cited, involve no appreciable error. However, it is important to know that we can find a value of n for Equation [1] which gives the same results as when values from the steam tables are inserted in Equation [2], which has a single evident and slight approximation. This basis is more satisfactory than previous deductions of [1] as the result of a number of approximations beginning with pv^n = constant, whether made by using a tangent to the isentropic as here worked out, or by use of approximations in Saint Venant's formula [6]. A value of n of 1.3 has been used with the latter approximation. The result is affected but little whether 1.3 or 1.28 is used. However, it is felt that 1.28 is somewhat better on the average for all parts of the steam tables.

If the value of the cold-nozzle diameter is D, the value of the square of the diameter to be used in the formula is

$$D^2[1 + 2\alpha(t_1 - t_s)]$$

Substituting all of these values in the expression [1] for w gives the final formulas [3] and [4] presented earlier.

Plan for Deduction of Exact Formula. Formulas [3] and [4] are amply accurate for all usual pressures with Δ/p_1 up to 0.10. However, for exceptionally high pressures such as 1200 or 2000 lb per sq in., where n as determined previously may be as low as 1.22, or for large Δ/p_1 or other unusual cases or for checking the accuracy of formula [3], it is desirable to have a fundamental flow formula, and this is next deduced, based on values of volume to be found for each point from steam tables such as Keenan's. As already remarked, we cannot employ any of the usual Mollier diagrams, for small values Δ/p_1.

The area of the pressure-volume diagram, Fig. 16, is obtained by assuming that the isentropic curve between known values of v_1 and v_2 is practically a curve $pv^n = c$, where n is such an exponent (never actually computed) as produces in any given case a curve corresponding to the given volumes. This method makes the assumption that the pressure-volume area under a curve $pv^n = c$ passing through the given points is practically the same as that under the actual isentropic curve. The legitimacy of this assumption has been demonstrated by computation of actual values of n from isentropic volumes found from values taken from Keenan's Steam Tables, for a number of pressure drops from a given initial condition. The value of n turned out to be the same for all points along a given isentropic.

In the use of the exact formula, the quantities found by graphical interpolation from the steam tables, as explained later, are not actually volumes, but values of p_1v_1 and p_2v_2, because these have the most gradual change. These are inserted in the final formula, the deduction of which is given below,

$$w = \frac{1890\,cD^2\,[1 + 2\alpha(t_1 - t_s)]\,\sqrt{p_2^2(p_1v_1 - p_2v_2)}}{p_2v_2\,\sqrt{1 - (D/D_1)^4\,(v_1/v_2)^2}} \times \sqrt{\frac{\log p_1 - \log p_2}{\log p_1v_1 - \log p_2v_2}} \quad \ldots\ldots [5]$$

Graphical Determination of "pv" Values. From the given values of v in steam tables such as Keenan's for various even pressures and temperatures, there are computed and plotted, values of pv against values of temperature t. Just as described for Fig. 15 in a preceding paragraph entitled "Empirical Determination of n" the value of p_1v_1 is found by interpolation. We next plot values of entropy s given in the steam tables against values of t for the same even pressures and for the same range of temperature. It may be necessary to cross-plot at the initial t_1 values of s against the even values of p to give at p_1 the exact initial s_1, or we may interpolate at p_1 between the curves to get s_1.

We next plot values of pv for values of v in the tables against s for even pressures on either side of the actual final pressure p_2, for a range of values of entropy s on either side of the initial s_1. By interpolating or cross-plotting at the initial s_1 between the curves of the values of pv against the even values of p, we find for p_2 the exact final value of p_2v_2. The resulting values p_1v_1 and p_2v_2 are inserted in [5]. This method is somewhat laborious, but very easy to execute, and yields good values of the two pv's.

Deduction of Exact Formula. Saint Venant's formula for flow of a fluid following the law pv^n = const. has often been deduced. For fluids other than a perfect gas (that is, pv not RT) the formula comes out

$$w = \frac{3600\pi\,D^2\,\sqrt{2g}}{48\,\sqrt{1 - (V_1/V)^2}\,\sqrt{1 - 1/n}}\,\sqrt{\frac{p_1}{v_1}} \times \sqrt{\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{1+1/n}} \quad \ldots\ldots [6]$$

In the usual deduction, n is a given exponent, and values of v are to be obtained from it. We here reverse the process, and suppose that we know v_2 and that n is that exponent which exactly fits with it in any given case, whatever it may be. We do not assume that n is the same for various isentropics but only that it has a constant value along any one isentropic. We never need to find this value of n, however. We next proceed to transform [6] so as to express it directly in terms of v_2. However, the numerical values are most easily found as values of p_1v_1 and p_2v_2, so that we use these terms. Since $p_1v_1^n = p_2v_2^n$,

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^n$$

From this it follows that

$$\left(\frac{p_2}{p_1}\right)^{2/n} = \left(\frac{p_2}{p_1}\right)^2 \left(\frac{p_1v_1}{p_2v_2}\right)^2$$

and

$$\left(\frac{p_2}{p_1}\right)^{1+1/n} = \left(\frac{p_2}{p_1}\right)^2 \frac{p_1v_1}{p_2v_2}$$

These relations make it possible to reduce the radical in [6] to the form given in [5].

By taking logarithms of $p_1v_1^n = p_2v_2^n$,

$$n = \frac{\log p_1 - \log p_2}{\log v_2 - \log v_1}$$

By further algebraic procedure we can express $\sqrt{1 - 1/n}$ in [6]. Since the flow is the same in the jet and the approach pipe,

$$\frac{V_1}{V} = \left(\frac{D}{D_1}\right)^2 \frac{v_1}{v_2}$$

so that we finally have [5] from [6].

Discussion

JOSEPH H. KEENAN[12] AND SAM P. SOLING.[13] The metering orifice and the metering nozzle will doubtless assume positions of greater importance in precise engineering testing as our ability

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to predict their coefficients improves. The paper brings the subject to the attention of power engineers in the operating and designing fields in carefully considered detail. If it serves to concentrate engineering attention on the problems of accurate orifice metering, it will be of great service.

At Stevens Institute of Technology we have undertaken the measurement of the difference in pressure between the throat of a nozzle and the exhaust space following it and the determination of discharge coefficients using air, water, and steam as the flowing fluids. The capacity of our air-flow apparatus limited us to a throat diameter of 0.6 in.

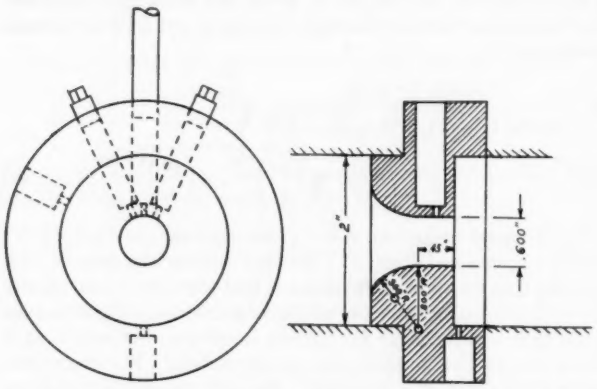


FIG. 17 NOZZLE USED IN STEVENS INSTITUTE OF TECHNOLOGY TESTS

The nozzle walls (Fig. 17) were designed in accordance with the authors' recommendations. It was impracticable, however, to use throat sampling holes as large as $^1/_8$ in., as the authors recommended, because of the small radius of the throat-wall surface. Three $^1/_{16}$-in. throat holes connected in parallel provided the throat pressure to the manometer. These holes were only $^3/_{16}$ in. from the nozzle exit instead of the minimum $^3/_8$ in. recommended, because the entire throat length was less than $^1/_2$ in.

We recognized that these small dimensions would increase the difficulties involved in accurate measurement, but adherence to the minimum dimensions recommended would have forced the nozzle design so far from geometric similarity with those of the authors as to invalidate comparison.

Care was taken to see that the edges of the sampling holes were smooth without excessive rounding. The depth of the $^1/_{16}$-in. drilling was slightly more than $^5/_{32}$ in.

The nozzle was machined out of a solid piece of brass, constituting a flange which could be inserted between standard 2-in. pipe flanges. The exit face of the nozzle was recessed into this flange, giving a short machined length which was continuous with the exit piping when in the operating position. Two pipe taps, each $^1/_{16}$ in. in diameter, opened into the space immediately following the exit face. These were used singly until it was established that no appreciable difference could be found between their pressures, after which they were used in parallel.

This type of pipe tap differs from those shown by the authors, who recommend a pipe-tap set a little upstream from the nozzle exit—an arrangement which is possible only when the nozzle projects into the exit pipe.

The approach pipe in all tests was free from bends or valves for a distance of at least 15 diam. The initial pressure tap was 0.27 in. in diameter and opened into the approach pipe 1 in. before the nozzle entrance in the case of water and 1.6 in. before the nozzle in the case of air and steam.

Pressures were measured with water columns or mercury columns, the drop across the nozzle never giving less than a 3-in. reading on the differential manometer. In the case of water flow, pressures were transmitted to the manometer through water-filled lines fitted with air traps. The steam-test pressure lines were arranged much as shown by the authors.

The results of the tests on the difference between the pressures in the throat and in the exit pipe are shown in Fig. 18. The water and air tests and one steam test appear to scatter about a straight line with about a 3 per cent slope when plotted against the pressure drop across the nozzle. The other three steam tests indicate a higher slope, which was doubtless caused by the accumulation of solid matter from the steam on the nozzle wall. When the nozzle was removed from the steam line, it was found that the brass surfaces had been streaked and discolored by deposited material. The thickness of the deposit was too small to affect measurably the throat area, but probably large enough to alter the stream flow immediately around the throat tap holes.

A similar difficulty but an opposite effect was encountered in the air tests. In the course of several tests it was noted that the difference between throat and pipe-tap pressures, which was being measured directly, was decreasing. The cause of the change was found to be a thin film of compressor oil which was deposited on the nozzle wall from the air stream. Cleaning the orifice with carbon tetrachloride between tests brought consis-

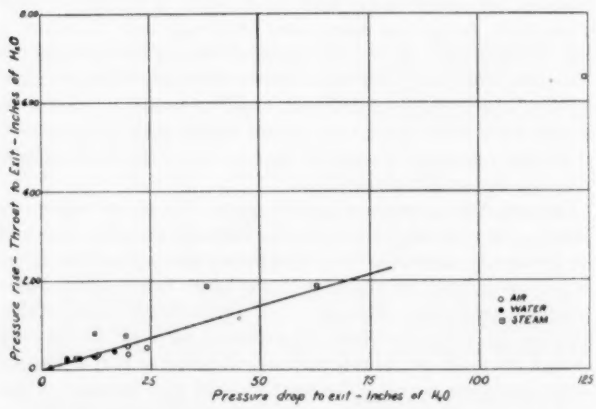


FIG. 18 DIFFERENCE BETWEEN THROAT-TAP AND PIPE-TAP PRESSURES

(The line represents a slope of 2.85 per cent. The high values from the steam tests are probably caused by a deposit of solid matter on the nozzle walls.)

tent results. The least difficulty was encountered in the water tests, during which no material was deposited on the nozzle walls.

Doubtless the difficulties involved in obtaining consistent throat-pressure measurements would have been less pronounced had the dimensions of our orifice and sampling holes been considerably larger. But even in the larger sizes these problems would still be present, though in lesser degree, and it is extremely doubtful whether throat-tap measurements have advantages that justify the difficulties they introduce. It is true that throat taps give a coefficient of discharge fairly near to unity. But the actual magnitude of that coefficient must be obtained from calibration of the orifice in question or of similar orifices regardless of where the low-side pressure is measured.

The statement that the coefficient determined from throat-tap measurements is the same as that of a nozzle freely discharging into the atmosphere or other large region is certainly not true. The jet issuing from the nozzle entrains fluid. That fluid must flow inward from the distant undisturbed atmosphere to the boundaries of the stream. A pressure gradient must exist in the direction of flow of the entrained fluid which causes the pressure surrounding the jet to be lower than that of the undisturbed

atmosphere. It is this pressure immediately surrounding the jet that determines the throat pressure.

Though our nozzle did not discharge into an indefinitely large atmosphere, the ratio of pipe area to nozzle area, 11 to 1, was large enough to approach that condition. Further, the corner position of our pipe tap insures a pressure approximating that of an undisturbed atmosphere. In every case we found the throat pressure lower than the pressure in the exit space. In the more reliable measurements the difference averaged 2.85 per cent of the pressure drop across the orifice, which gives a coefficient from throat pressure approximately $1^1/_2$ per cent lower than the coefficient from exit-space pressure.

Further evidence concerning the direction of this pressure differential is found in the steam-flow test of a freely discharging rounded-entrance nozzle made by Warren and Keenan.[14] Fig. 15 of that paper shows the flow coefficient based on the Marks and Davis steam tables to be about 101.5 per cent at low velocities. Beyond the critical-pressure ratio, where the pressure in the throat assumes the value used in computing the theoretical, the coefficient drops to 99.25 per cent. Thus it is reasonable to assume that the difference between the 99.25 per cent value calculated from the known throat pressure and the 101.5 per cent calculated from measured exhaust-chamber pressure is a measure of the difference between throat and exhaust-chamber pressures. The 2.25 per cent difference corresponds to a throat pressure which is lower than the exhaust-chamber pressure by 4.5 per cent of the pressure drop across the nozzle. The same tests recalculated using the Keenan tables give 101 per cent at low velocities and 99.2 per cent beyond the critical, a difference of 1.8 per cent in coefficient which corresponds to a throat pressure which is lower than the exhaust-chamber pressure by 3.6 per cent of the overall pressure drop. This is a very satisfactory agreement with our measured value of 2.85 per cent.

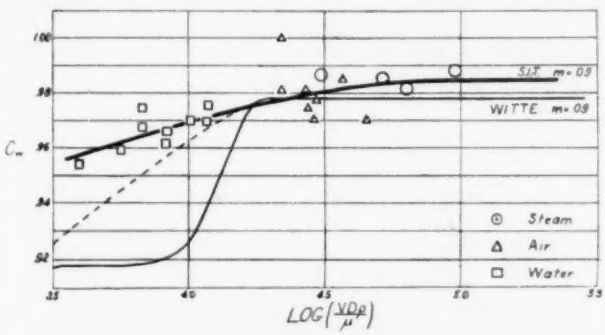


FIG. 19 DISCHARGE COEFFICIENTS FROM S. I. T. COMPARED WITH I.G. NOZZLE TESTS

(The pipe diameter is used in computing Reynolds' number. Coefficients are based on pipe-tap measurements. The dash line bridges over the step in Witte's results.)

TABLE 1 TEST RESULTS ARRANGED IN ORDER OF TESTING

	Initial pressure p_1 lb per sq in. abs.	Initial temp t_1 deg F	Pressure drop to pipe tap Δ lb per sq in.	Pressure ratio $\frac{\Delta}{p_1}$	Throat - pipe differential δ in. H_2O	Differential ratio $\frac{\delta}{\Delta}$ %	Discharge coefficient c_w	log RD
Air	15.30	57	0.464	0.0303	0.33	2.56	0.9816	4.34
	15.67	55	0.718	0.0458	0.35	1.76	0.9750	4.43
	15.08	66	0.460	0.0305	0.37	2.90	1.0039	4.34
	15.29	63	0.712	0.0466	0.53	2.68	0.9817	4.42
	15.43	66	0.868	0.0562	0.50	2.08	0.9772	4.47
	44.26	68	0.440	0.0099	0.30	2.46	0.9845	4.56
	44.26	80	0.698	0.0158	0.76	3.93	0.9703	4.65
	44.06	74	0.282	0.0064	0.25	3.20	0.9704	4.46
Water	16.5	57	0.606	0.0364	0.42	2.50	0.9698	4.06
		57	0.618	0.0371	0.40	2.34	0.9753	4.07
	16.8	57	0.456	0.0274	0.32	2.53	0.9697	4.00
	16.7	57	0.315	0.0189	0.25	2.86	0.9615	3.91
		57	0.311	0.0186	0.25	2.90	0.9660	3.92
	16.7	57	0.202	0.0121	0.20	3.58	0.9678	3.82
	16.6	57	0.206	0.0123	0.25	4.39	0.9745	3.83
	16.5	57	0.144	0.0087			0.9596	3.59
	16.5	57	0.072	0.0045	0.05	2.50	0.9541	3.74
Steam	45.10	332	1.357	0.0301	1.86	4.94	0.9856	4.71
	45.10	355	2.256	0.0500	1.89	3.02	0.9815	4.80
	45.10	292	4.486	0.0995	6.57	5.29	0.9882	4.96
	45.10	296	0.434	0.0096	0.80	6.66	0.9868	4.47

[14] "A Machine for Testing Steam-Turbine Nozzles by the Reaction Method," by G. B. Warren and J. H. Keenan, Trans. A.S.M.E., vol. 48 (1926), p. 33.

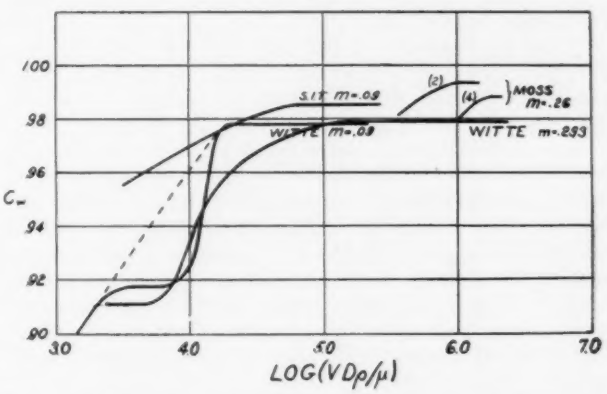


FIG. 20 DISCHARGE COEFFICIENTS COMPARED ON REYNOLDS' NUMBER BASIS

(m = nozzle area/pipe area. Numbers on the Moss and Johnson curves represent the curve numbers in the paper.)

A pipe-tap measurement made in a recessed region upstream from the nozzle exit might give, as the authors claim, a pressure lower than the throat pressure. If so, it is doubtless because of high-velocity local eddies in the constricted space between the nozzle and the pipe.

There is of course no one right place to measure the low pressure in a nozzle test. Since the coefficient used must be determined by calibration in any case, reproducibility of result is the only requirement. Consequently, pipe-tap measurements, which are in general more reproducible with ordinary care, are preferable to throat-tap measurements. It is highly desirable, nevertheless, to standardize the method of measuring low-side pressure in order to facilitate the correlation of results obtained by various experimenters. Only after correlation of comparable experimental results from many sources shall we be able to predict with confidence and with precision the discharge coefficient of a nozzle in any specific case.

Dimensional reasoning indicates that for geometrically similar nozzles in geometrically similar piping systems the discharge coefficient is a function of Reynolds' number and the ratio of nozzle area to pipe area:

$$c_w = f(VD\rho/\mu, m)$$

The validity of the Reynolds number as a criterion has been experimentally verified for a given nozzle in a given pipe by the experiments of Witte. In an extensive series of measurements with water, oil, and steam sponsored by the I.G. chemical industries, Witte[15] showed that the coefficient of discharge of any one nozzle or orifice in any one pipe system was a unique function of Reynolds' number regardless of the fluid used. In the case of the rounded entrance nozzle, the coefficient of discharge remains constant for all Reynolds numbers higher than about 10^5. Be-

[15] Witte, *Z.V.D.I.*, vol. 72 (1928), pp. 1493–1502.

iow that value the coefficient decreases gradually as Reynolds' number is reduced.

If the pressure drop across a nozzle is large enough to cause a marked change in the fluid density during the expansion, then it can be shown dimensionally that a compressibility criterion must be added to Reynolds' number and area ratio in the expression for discharge coefficient. A pressure ratio, final pressure to initial pressure or pressure drop to initial pressure, is usually used as the compressibility criterion. Witte's tests, which covered pressure drops in steam flow up to about 5 per cent of the initial pressure, showed the coefficient to be independent of pressure ratio and consistently dependent on Reynolds' number.

At Stevens Institute of Technology we have obtained discharge coefficients (Fig. 19) using air, water, and steam over a wide range of Reynolds' numbers. Initial conditions and pressure ratios are shown in Table 1. The steam and water flows were determined by weighing. The air flow was measured by determining the course of pressure and temperature with time in a supply tank of known volume. Three temperatures measured at different heights above the tank bottom were averaged to obtain the temperature of the air remaining in the supply tank at any instant. The pressure was held constant ahead of the orifice as the pressure dropped in the tank. The slope of the ratio p/T against time determined the rate of flow. Uncertainties in instantaneous tank temperatures were caused by radiation to the thermocouples from the tank walls during the test. Steps have been taken to eliminate this difficulty and to reduce the spread of the test results.

The points plotted in Fig. 19 show, as do Witte's tests, a good correlation against Reynolds' number, but no consistent trend against pressure ratio. Witte's curve for the same area ratio is compared in the figure with the values from Stevens Institute. The agreement in maximum value (within $^3/_4$ per cent) is fortuitous because the two orifices are not geometrically similar; the I.G. orifice has a smaller entrance radius. The two curves begin their decline at approximately the same Reynolds number, though Witte's curve falls off much more steeply, levels off for a space, and then continues downward. Similar steps appear in Witte's other curves in unsystematic fashion, representing, perhaps, a peculiarity of a given set-up which could hardly be reproduced even in a close copy of it. The dash line bridging the step is doubtless more representative of the general case.

In Fig. 4 the curves from Witte, S. I. T., and Moss and Johnson are shown plotted against Reynolds' number. The Moss and Johnson data are inconsistent with the other two sources in two ways: First, they show variations in coefficient with pressure drop in a region where the other experimenters have found all variation to have ceased; second, they show two separate curves for the same nozzle and pipe combination when operated with steam at two different initial pressures. Our own tests with different fluids and Witte's tests covering a wider range of steam pressures than Moss and Johnson's both show the discharge coefficient to be a unique function of Reynolds' number. Change in pressure, temperature, density, viscosity, etc. caused no departure from the single curve which represents all tests when they are plotted against Reynolds' number.

The authors express the results of all their tests on several nozzles at various pressures (curve 3, Fig. 10) as a single function of pressure ratio. This is in direct contradiction to dimensional reasoning and to recent experimental studies of nozzle coefficients. The agreement shown in Figs. 10 and 11 in the droop in coefficient at low-pressure drop is more probably an indication of systematic error than an indication of the influence of compressibility.

The test data from Stevens Institute should be considered as tentative only. Since the work has been undertaken only recently, there are many opportunities available for improving test equipment and technique. We hope to carry out a further study of the variation of discharge coefficient with Reynolds' number, with area ratio, and with pressure ratio, concentrating for the present on the type of nozzle recommended by the authors.

R. E. Sprenkle.[16] Two nozzle shapes or designs appear in Fig. 1, also in Figs. 8 and 9. The first of these designs in Fig. 1 is of the shaped or flared type, with the outlet connection placed into the wall of the pipe back of the nozzle throat. The second design is formed of a solid block of metal and has the outlet connection placed into the inner throat of the nozzle. Since both of these designs are recommended, the writer presents comparative data showing the relation between these two designs in terms of coefficient of discharge, ability to detect disturbances in the flow, as well as stability in performance. It must be remembered that there are hundreds of nozzles in commercial use today of the flared type using the outlet connection made into the pipe wall back of the nozzle throat, as shown in the upper portion of Fig. 1, whereas there are but comparatively few of the solid-nozzle shapes using the throat connections, in either commercial or testing use.

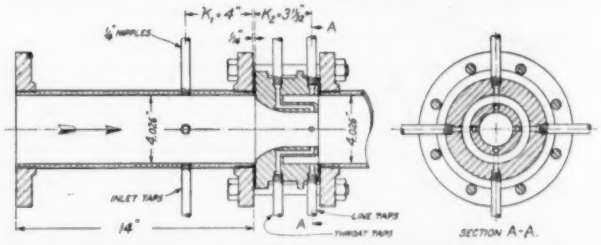


Fig. 21 Special Bailey Test Nozzle

In order to make such comparison tests between the two different designs of nozzles, the Bailey Meter Company made up a special 4-in. by 2-in. test nozzle, as shown in Fig. 21. This nozzle contained the basic characteristics of both designs of nozzles, in that it possessed not only connections going through the throat of the nozzle, but also connections which opened out into the annular space back of the nozzle throat. These latter connections simulated pipe taps precisely, inasmuch as the larger diameter of this annular space corresponded to the inside pipe diameter, whereas the smaller diameter of the annular space corresponded to the normal outside throat diameter. The only deviation in the construction of this special nozzle from the throat-tap kind of nozzle, as shown in the lower part of Fig. 1, is that the throat connections changed direction and ran through the wall of the nozzle instead of straight into the throat as recommended. Four throat and four pipe connections were used, each of these being at 90 deg with each other. The purpose was to detect any disturbances around the inner pipe surface which might be created by improper piping conditions on the inlet side of the nozzle.

In making up the installation of this special nozzle, a short section of 4-in. bored brass pipe was placed immediately in front of the nozzle, this section having a diameter of 4.026 in. Into this pipe section at a distance of one pipe diameter preceding the outlet flange were placed four inlet static-pressure connections, 90 deg apart. The nozzle and this inlet section were then so connected that each inlet static-pressure connection lined up with one throat and with one pipe connection. To each pressure connection was connected a water manometer open at the top

By this hook-up it was possible to take simultaneous readings

[16] Mechanical Engineer, Bailey Meter Company, Cleveland, Ohio. Assoc-Mem. A.S.M.E.

of all the static pressures, not only on the inlet side of the nozzle, but on the outlet side, as measured both along the pipe wall and in the throat surface. Thus it was possible to obtain a direct comparison not only of the coefficients of discharge, but also of the ability of each set of throat and pipe connections to detect disturbances in flow through the nozzle.

All tests made on this nozzle in the company laboratory were run with water. This special nozzle and the short inlet brass section were installed initially as shown in Fig. 22, piping arrangement No. 1. For subsequent tests to determine the relative effect of disturbances on the two kinds of outlet pressure connections, the piping was arranged as shown in Fig. 22, piping arrangement No. 2.

General Electric engineers were invited to witness the tests, and Mr. Johnson responded, and helped to collect most of the data taken in these tests.

Fig. 23 shows the coefficients of discharge obtained on this special nozzle, using both sets of outlet connections. The lower and solid group of lines show the coefficients obtained from the pipe connections, whereas the upper and broken lines indicate

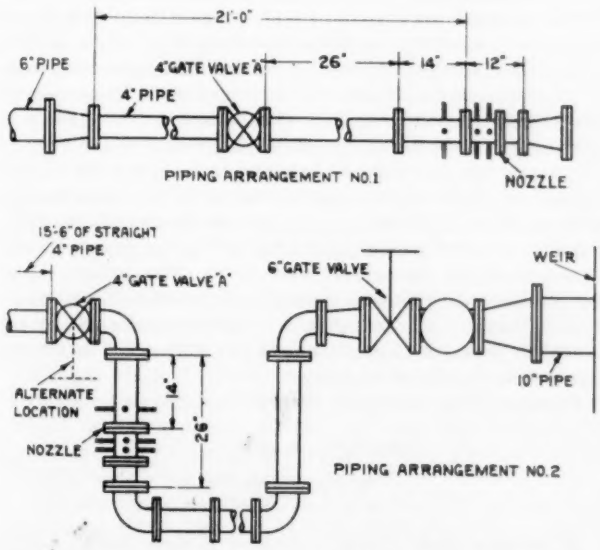


Fig. 22 Piping for Tests on Bailey and G.E. Nozzles (Using both throat and line taps.)

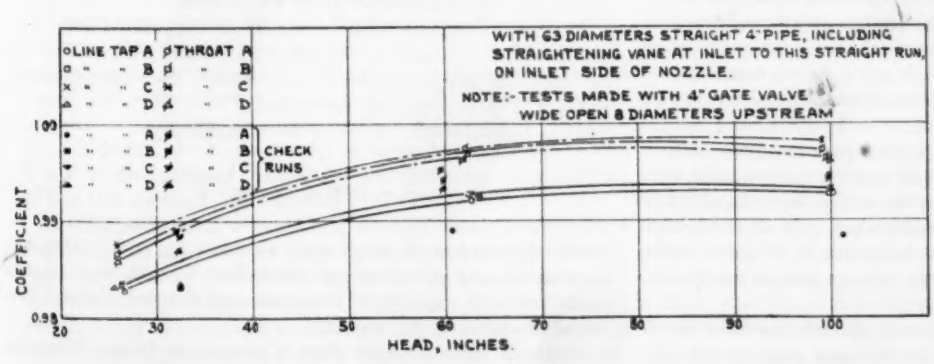


Fig. 23

those obtained from the throat connections. By reference to Fig. 10 in the paper, it will be noticed that the coefficients obtained from the throat connections checked very closely those obtained in the Lynn tests on steam. These data were obtained using piping arrangement No. 1.

Fig. 24 shows the comparative results obtained from the throat and the pipe connections when the special nozzle was installed in piping arrangement No. 2 with the gate valve wide open. It will be noticed that these coefficients check closely those shown in Fig. 23, thus indicating that there was no disturbance produced by this particular piping arrangement.

Again using piping arrangement No. 2, but with the gate valve throttled so that it was only two turns off the seat, the

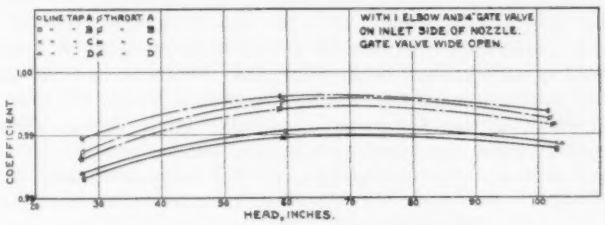


Fig. 24

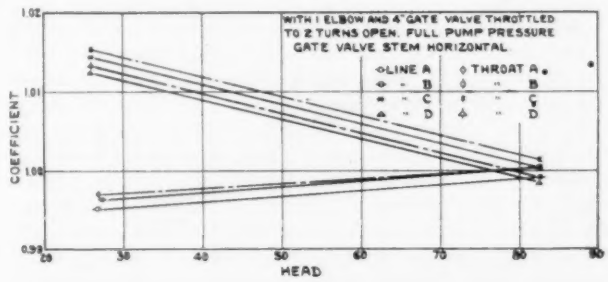


Fig. 25

data as shown in the curve of Fig. 25 was next obtained. It will be noticed that the coefficients of both kinds of outlet connections increased considerably, thus indicating that considerable swirl or disturbance was present. It will be further noticed, however, that there was no appreciable increase in spread of coefficients between either the four-throat or the four-line connections. In this curve as well as in the two previous ones, the upper set of curves, or the broken lines, show the throat-connection results, whereas the lower curves. or the solid lines, show the pipe-connection results.

Mr. Johnson brought with him another special nozzle which had been made in Schenectady, of the same size and general characteristics as the one which we had made up, with exception that the pipe-connection design of the nozzle was omitted. This nozzle was so placed in the set-up that simultaneous readings could also be taken, not only of the differential pressures as measured by the pipe connections, but also by the throat connections, so as to obtain a direct comparison between these two kinds of outlet connections. The results obtained from this nozzle indicated approximately the same differential-pressure variation between the throat connections and the pipe connections as was obtained from the special Bailey nozzle.

In all three of the curves shown it will be noticed that there is a slightly greater spread between the four throat-connection

readings than between the four pipe-connection readings. This is true regardless of whether there is any disturbance in the flow, as is evidenced by the fact that there is approximately the same spread between the throat connections in the case of the installation in piping arrangement No. 1 as in piping arrangement No. 2 with the gate valve throttled, the latter arrangement being that which produced the marked disturbance, as evidenced by the change in coefficients. While this spread between the four throat connections is small, in magnitude, we were still unable to reconcile these results, particularly in the case of piping arrangement No. 1, where we were sure there was no disturbance, until we rotated the nozzle axially in the set-up. We found by so doing that the connections which were giving the highest static readings in the initial position gave the same highest static readings when moved around to positions of 90 deg and 180 deg from their original location. This demonstrated the fact conclusively that the

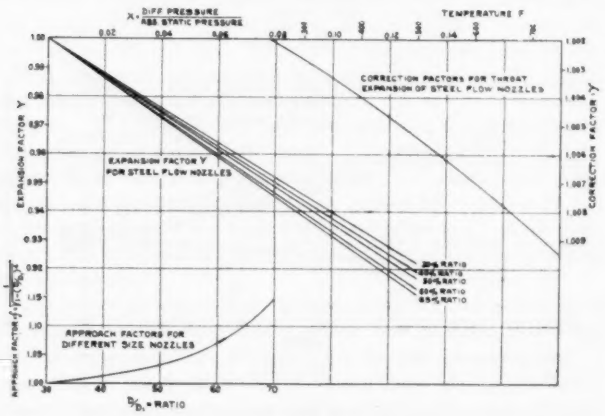


FIG. 26

reason for this spread in coefficients was due not to any disturbances in the pipe, but to the construction of the individual pressure connections into the throat. That is, the edge condition of the throat holes was found not to be exactly the same in all four cases. This meant that the differential as measured between one inlet connection and its corresponding throat connection would not be quite the same as the differential measured between any other inlet connection and its corresponding throat connection. Considerable time was spent in an effort to work down the finish of the openings into the throat, but it was found that it was impossible to get quite as close an agreement between the four different throat connections as between the line connections, even after the nozzle throat had been ground out very carefully and the edges of the throat pressure connections dressed down with crocus cloth and a countersink tool. On the other hand, no difficulty at all was experienced with the pipe connections back of the throat of the nozzle. These four pipe-connection holes were not touched from the beginning to the end of the tests, and thus was demonstrated their marked ability not only to check each other, but also their freedom from fussy details of construction.

As a result of this series of tests, certain definite conclusions can be drawn, as follows:

1 Nozzle-outlet connections placed directly into the throat gave a slightly higher coefficient of discharge than do the pipe connections placed back of the nozzle throat, but that both kinds of outlet pressure connections repeat very well and can be used, therefore, for obtaining a reliable measurement of the differential across the flow nozzle.

2 Pipe connections placed back of the nozzle throat into the pipe wall are just as effective in detecting swirls or other disturbances created by various arrangements of piping set-up on the inlet side as are the connections made into the throat of the nozzle.

3 Throat connections have been found considerably more sensitive to any imperfections of manufacture than pipe connections. This is due to the condition of the edge of the pressure hole into the throat, as the condition of this hole must be precisely the same in every case or the static pressures as measured by the different throat connections will be different. Therefore, great care must be taken in the manufacture of nozzles having throat connections so as to be sure the hole-edge conditions are as uniform as it is possible to make them. Even with this close attention, there is apt to be somewhat more spread or difference between the various throat connections placed circumferentially around the pipe than between the pipe connections.

4 A 50 per cent ratio nozzle is not very susceptible to irregularities in flow conditions except under very adverse conditions. The throttling of a gate valve in the straight horizontal run of pipe preceding it seems to have very little effect on accuracy. In fact, only when in combination with an elbow in such a way that the flow is pushed out to the outer edge of the pipe wall and must then travel through a bend, thus producing a swirl or other disturbance, will adverse calibration be secured.

The formulas given in this paper, while entirely correct, are somewhat difficult in application. They can all be greatly simplified by using the term Y as the expansion factor, the value of which can be found from the curve in Fig. 26 for any set of conditions. As will be noticed, the abscissas on the upper part of this curve are ratios of the differential pressure to the absolute inlet static pressure, with the ordinates at the upper left the corresponding values of the expansion factor Y for steam flow for different ratios of D/D_1, as shown by the diagonal lines. This graphic method of measuring in terms of flow the change of state of steam checks the method used by Dr. Moss within 0.1 to 0.2 per cent, and therefore is amply accurate for all test conditions. Further, it is the scheme as now recommended by the A.S.M.E. Research Committee on Fluid Meters for calculating steam flows through flow nozzles.

Equation [5] of this paper would thus resolve to

$$w = \frac{1890\ cD^2\ (1 + 20\ (t_1 - t_s))\sqrt{Y}}{\sqrt{1 - (D/D_1)^4}\sqrt{v_1}} \quad \ldots\ldots\ldots [7]$$

If we now allow $\frac{1}{\sqrt{1-(D/D_1)^4}} = f$, these values can be obtained from the lower left-hand part of Fig. 26, in which the values of f for each ratio of D/D_1 are plotted.

Further, we can plot values of the throat-expansion factor $(1 + 20\ (t_1 - t_s))$, as shown in the lower right-hand part of Fig. 26, for different temperatures of the steam. The corresponding value of $[1 + (t_1 - t_s)]$ we have called y.

Thus Equation [7] can be further simplified to

$$w = 1890\ cf\ D^2 y \sqrt{\frac{\Delta}{v_1}}\ Y$$

which allows of much more rapid calculations, and further can be used for any calculation of steam flow through steel nozzles regardless of the quality of the steam and the size of the differential produced by the nozzle.

Professor Keenan's tests show a reversal as to the difference between the pipe-tap and the throat-tap coefficients shown by data of this paper and by our own data. The reason for this discrepancy apparently is due to the fact that the throat connection was taken beyond the end of the throat section of the nozzle and in a region where there were apt to be swirls or other disturbances. We have found it necessary to place the outlet connection to any flow nozzle immediately back of the straight

throat section of the nozzle in such a position that no part of the pressure opening projects beyond the edge of the nozzle throat end.

Professor Keenan's graph plots coefficients for different pipe size flow nozzles on the basis of Reynolds' number, and by so doing he finds some discrepancies in the results. Reynolds' number alone is not the proper medium for expressing flow-nozzle or orifice coefficients on the basis of transferring data obtained from one pipe size to another without the use of some auxiliary correction factors to compensate for the pipe-size effect. The necessity of this has been proved by the various types of work which we have done, not only in Cleveland, but at Ohio State University. However, when it comes to the transfer of coefficients from one fluid to another in the same size pipe and using the same primary device, then Reynolds' number does have a decided field of usefulness. In other words, in attempting to obtain steam-flow coefficients from water-flow test data where there is such a large change in viscosity between the two fluids, there must be some correction made to compensate for this factor. In the metering of vapor flow, the differential produced for the lowest rate of flow or Reynolds' number is equivalent to the largest differential head equivalent to liquid-flow metering. The proper coefficient, therefore, to use for steam flow is that which is obtained from the highest differential pressure across the primary when metering water flow. Were coefficients for these primary devices unchangeable with the rate of flow or with Reynolds' number, there would be no reason for this change in transfer, but inasmuch as there is a decided variation in coefficient with head or with Reynolds' number, readjustment must be made in order to obtain the most correct analysis.

G. B. Warren.[17] The authors have, together with the research work and experience of the last few years of the manufacturers of flow meters, obtained sufficient data and developed a technique which should make the flow nozzle, when used with sufficient precautions, a precision instrument of high enough accuracy to permit making turbine tests upon which reliance can be placed. This procedure should not be used to supplant weighed tests when such can be obtained.

There are some aspects of the testing of steam turbines which this paper did not aim to cover and which should be brought to attention.

In connection with large turbines, we have for at least ten years been conscious of the importance of having available, to facilitate proper turbine design, a large body of tests under widely varying conditions of turbines of the most modern design. We have of course carried out extensive research relating to the detail design and performance of the parts of turbines, but after all, the criterion of the value of a design is how it performs in the customer's plant. Such test results on large turbines must be obtained in the customer's plant, as no manufacturer has facilities to test this economy in his factory.

It is necessary that tests made in the field be not only accurate and complete, in covering a wide variety of operating conditions, but that searching analysis determine the meaning of the results in terms of the turbine design. The results must be studied over a period of years and a running study must be made of the relationship of the results on each machine with the results obtained in preceding tests. This we have been doing, and the results so obtained have been embodied in the underlying knowledge upon which turbine design is based.

As to the smaller turbines, prior to the development of this flow-nozzle method of testing, the only accurate system involved the use of weighed flows. On non-condensing and extraction machines this either meant that no tests could be obtained or that expensive and wasteful tests would have to be made in the builder's factory. The short time required for shipment of such smaller machines, together with the highly competitive condition in the market, precluded doing this on more than a few turbines. Further, most of these smaller turbines are in use in industrial applications or small power plants where their owners have rightfully considered that the cost of testing equipment was not a charge which they should be asked to bear, and so such equipment has not been installed. The result has been that manufacturers of small turbines have not been able to obtain the assistance from test results in customers' plants that the manufacturers of large turbines have been able to obtain. It is only fair to assume that the design of the smaller machines has inevitably suffered as compared to what it might have been had ample test information been available, excepting as they have been able to profit by the results of tests on large turbines in the field and tests on small machines in the factory, where these results have been available.

In connection with the use of flow nozzles for testing turbines, the writer would point out the results of certain work which we have done in connection with the testing of larger machines. A few years ago it became quite apparent that in the older stations the condensate-weighing apparatus would not be sufficient to test the larger machines going into these stations and that the pressure for economy in first cost was forcing the builders of new stations to eliminate the installation in many cases of testing equipment. We turbine manufacturers felt rather good about this in many respects, inasmuch as it reflected a changed attitude on the part of our customers who were beginning to have sufficient confidence in our meeting our guarantees so that they could eliminate the making of accurate tests. It is quite a contrast to the European situation, where every customer assumes that the machine has failed to meet its guarantee until the manufacturer proves otherwise. On the other hand, the situation had certain dangers for us in that it meant that, unless some action was taken on our part, we would not be able to get a sufficient body of test information on the large, modern machines. Accordingly we developed a system of flow-nozzle measurement which we could install in the condensate line following the condensate pump on such installations. The pressure differences were measured by a combination of water and mercury U-tubes, and a very careful calibration over a wide range of flows with seven different sizes of nozzles was undertaken with the help of Professor Pardoe at the University of Pennsylvania. These tests and the subsequent experience which we obtained with the use of these instruments and that which was recently obtained at the Hudson Avenue Station on one of these flow nozzles by the Brooklyn Edison Company convinced us that such a system of measurement could be made, and if used with the proper precautions could obtain very accurate results. During the last four years we have tested about ten turbines with this equipment.

When we plotted the flow coefficient of these nozzles, ranging from $1^1/_4$ to 5 in. in diameter against the Reynolds number, we found that the flow coefficient of a group of geometrically similar nozzles is a function only of the Reynolds number. As a result of this we feel quite confident that we can predict the coefficient of nozzles when made geometrically similar to these and used to measure water, to within plus or minus 0.2 per cent. We are, however, still testing any new nozzle either before or after each important test.

The ease and inexpensiveness of making turbine tests should now permit the users of turbines, both large and small, to obtain performance tests on their machines. The results of such tests if made available to the manufacturers should be of material assistance to them in continuing to make progress in turbine

[17] Turbine Engineering Department, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

design, and in addition should serve as a healthy stimulant toward making them continue to live up to guarantees.

FRANCIS HODGKINSON.[18] A reasonably accurate and standardized method of determining a steam quantity by means of flow through an orifice is needed. The Steam Turbine Test Code of The American Society of Mechanical Engineers, as it stands today, discriminates against the determination of steam quantities by orifice-flow measurement, while, on the other hand, the Steam Turbine Rules for Acceptance Tests of the International Electrotechnical Commission permit this determination by this means. The A.S.M.E. code is now undergoing revision, and in the revised code this discrimination will probably have been removed. Of course, there will be appropriate precautions, made mandatory, to secure accuracy, with perhaps some qualifications as to the expected accuracy. Such measurements, however, are not to be confused with those by means of commercial flow meters, and turbine manufacturers have been justified in the past when they have declined to recognize acceptance tests if carried out by any such means.

Steam-quantity measurements by orifice-flow means have made more progress in Europe than in this country. The standardized nozzles of the V.D.I. are commonly employed. It is suspected that this greater progress in Europe is due to the reason pointed out by Mr. Warren, i.e., that in many European countries the burden of proof that the machine under consideration conforms to the contract warranties is on the shoulders of the manufacturer, which is contrary to the principles of common law. In Europe, the manufacturer has to carry out the test with little or no assistance from the purchaser beyond the purchaser exercising vigilance to assure himself that the test results represent the facts. Frequently, the arrangement of the plant is such that a test cannot be carried out by means of the weighing of condensate, except at an inordinate cost. Hence, the great need of the less costly method of means of measurement.

As far as the I.E.C. Rules for Acceptance Tests of Steam Turbines are concerned, while determinations by means of flow measurement are permitted as at present written, they are non-mandatory because a special committee of the International Standards Association has been appointed to study the subject and to formulate standardized nozzles and rules for the carrying out of such measurements. It is expected that the I.E.C. rules will remain non-mandatory until such time as the I.S.A. completes its work, when its reports will be considered, and then the I.E.C. rules will be revised and made mandatory. Beyond doubt, the A.S.M.E. test code is bound to permit steam quantity measurement by some such means as described in the paper.

DAVID J. BLOOMBERG.[19] The paper has properly given detailed treatment to the "test set-up." However, even with a perfect test set-up, there are many chances for slip-ups with respect to obtaining the correct result. In this discussion an attempt will be made to set down the results of our experience in this respect.

In any test system for the determination of a quantity, the difference between the derived test value and the true value is affected by three factors; namely: (1) test setup, (2) test observations, and (3) computation of test results.

While it is possible that differences due to each of the three factors may offset each other, still the width of the band between the true value and the derived test value may be the numerical sum of the individual differences due to all three factors.

It is obvious that for the precision called for in this flow measurement the differences or the errors contributed by each of the three factors should be kept to as low a value as consistent with the time-accuracy characteristic. In the paper the "test set-up" is well taken care of. There is, however, a certain relation between the test set-up and the test observations for precision testing. The test set-up should be planned to permit the test observations to be taken by the observers with the minimum fatigue.

The total time duration and the time interval between observations will depend upon the steadiness of the steam conditions. The greater the time duration and number of observations, the greater the opportunity for the law of averages to balance out observational errors.

Thus while the errors due to the test set-up are fixed, those set up by test observations can be partly controlled by the arrangement of the test instruments and the time duration and number of observations. The statement is often made that the precision of the computation of the test observations need not be any greater than that of the observations themselves. This may be true if we were dealing with a large number of independent tests at the same conditions, or conditions that would lend themselves to graphical presentation permitting an average characteristic curve to be drawn through the points. However, in the case of flow-nozzle tests there usually is only one point, and therefore the width of the band of the deviation of the derived test values from the true values will be that of the arithmetical sum of the mean deviations resulting from the three factors—test setup, observations, and computations.

The magnitude of the error contributed by the flow computation is under more positive control than that of the other two factors, and therefore there is no reason why this should not be kept down to $^1/_{10}$ of 1 per cent.

In our experience we have found it desirable to rearrange Equation [3] so as to permit the use of curves and thus facilitate the flow calculations. The equation as rearranged is:

$$w = \frac{1.664\ cD^2\ [1 + 2\alpha(t_1 - t_s)]}{\sqrt{[1 - (D/D_1)^4 + 1.562\ (D/D_1)^4\ \Delta/p_1]}} \times p_1 \times 1136 \sqrt{\left(\frac{1}{p_1 v_1}\right)} \times \sqrt{\left(1.17\ \frac{p_2}{p_1} - 0.17\right)\left(1 - \frac{p_2}{p_1}\right)}$$

The term $1.664\ D^2\ [(1 + 2\ \alpha(t_1 - t_s)]$ for a given nozzle is plotted against t_1 as abscissa. The value of α varies with different materials.

The term $\sqrt{[1 - (D/D_1)^4 + 1.562\ (D/D_1)^4 \Delta/p_1]}$ is plotted against D/D_1 for different constant values of $\sqrt{(\Delta/p_1)}$.

The value of c, being a function of Δ/p_1, can be incorporated in these curves.

The term $1136\ \sqrt{(1/p_1 v_1)}$ is taken from the steam-flow equation used in our steam-turbine calculations. Mr. E. C. Perry, of the General Electric Company turbine engineering department, first introduced the use of the curves of $1136\sqrt{(1/p_1 v_1)}$ plotted against p_1 for different constant values of initial temperature and superheats. These curves can be plotted on small curve sheets with a fairly open scale, permitting the reading of the values with accuracy well within $^1/_{10}$ of 1 per cent.

The values of the final term $\sqrt{[(1.17\ p_2/p_1 - 0.17)\ (1 - p_2/p_1)]}$ are best presented in form of a table, for even values of Δ/p_1.

The application of the flow nozzle in measuring the inlet flow in connection with turbine efficiency tests has one advantage, which has not been previously stressed. That is, an observational error in the steam temperature will cause comparatively little

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[19] Research Engineer, Turbine Engineering Department, General Electric Company, River Works, West Lynn, Mass. Assoc-Mem. A.S.M.E.

error in the calculated turbine input, since the effect of this temperature error on the derived flow will be to a large extent balanced by the effect of the same temperature error on the available energy used in determining the turbine input. To retain this beneficial feature, the relation of the turbine-inlet temperature to nozzle-inlet temperature should be correctly determined. Our experience has been that with well-lagged pipe between the flow nozzle and the turbine inlet, the heat losses are relatively small and the temperature differences will be that due to steam throttling.

N. E. Funk.[20] It is pleasing to learn that the steam-turbine manufacturers are at last satisfied that it is possible to meter the flow of steam in pipes with sufficient accuracy to be acceptable for turbine tests. For many years the writer has felt that it was desirable to find some substitute for the expensive and clumsy method of determining the condensate from a turbine with weighing or measuring tanks.

In 1920 the Philadelphia Electric Company arranged for metering the condensate on the turbines which were installed in its Delaware station. This was done by inserting a removable section of pipe containing an orifice in the condensate line. This removable piece of line could be inserted in a system of piping containing a V-notch meter in which water was circulated by means of the pump so that the pipe and orifice combined could be accurately tested. Later, in 1925, a similar arrangement was used with the installation of the 60,000-kw turbo-generators at the Richmond station, together with a new V-notch measuring equipment necessary because of the increased amount of flow. The V-notch meters in this installation were accurately calibrated at the University of Pennsylvania. A description of this equipment can be found in the technical publications.[21]

The turbine test results obtained from the use of orifices so calibrated have been extremely satisfactory in determining the initial turbine performance and for the purposes of closely following the economic operation of the stations. Although turbine tests were always made using a special mercury column for accurate measurements, the regular operating procedure is to connect a standard recording flow meter to the calibrated orifices. This combination is sufficiently accurate to permit closely following daily plant performances.

This method of course has left much to be desired because it was only applicable to the measurement of water. It is very gratifying that equipment has been developed to permit the measuring of steam with the accuracy that we have been able to obtain for a number of years in the measurement of water.

Edward C. Perry.[22] It seems worth while to call to the attention of those who may try out this system certain precautions in the set-up and condition of the differential U-tubes (Fig. 7) which, if neglected, may nullify the refinements of theory expressed in the formula and apparatus:

(*a*) The illumination of the U-tube should be carefully studied. It must be adequate. It should be so arranged that no shadows or reflections make it difficult for the operator to take his readings quickly and easily. There must be a good light under the sliders at all positions; hence a movable light is usually desirable. Sometimes this can be supplemented or replaced by a beam reflected from a small mirror in the hand of the operator. Avoid direct light from windows or skylights, which may cause troublesome reflections on the glass in front of the tubes.

(*b*) In addition to good light, it is important to have the brass scale bright, the interior of the box white, and the glass in front clean.

(*c*) The position of the U-tube above the floor should, if possible, be such that the operator can always keep his eyes on a level when taking readings without getting into difficult positions. In the case of large deflections, the operator can sometimes use a small mirror, lining up the slider with the meniscus by means of the reflection. This is usually necessary only on the lower slider, and often helps in checking the upper and lower levels without moving the eyes from one level to the other.

(*d*) The U-tube should of course be placed where there is no hindrance from surrounding objects or from other persons.

S. A. Tucker.[23] The use of the flow nozzle where the fluid to be measured is steam is a problem not met often in central-station testing where the quantities are very large and the condensate is usually available for flow-nozzle or weighed-water measurement. Because of the trend in central-station design toward minimizing interconnections between units, centralized weigh tanks are becoming an increasing burden on the already high cost of acceptance tests.

The cost of revising our weigh-tank equipment was recently emphasized by modifications necessary to permit testing 160,000-kw units. We therefore have recently made two series of tests at Hudson Avenue with the cooperation of the General Electric Company on a flow nozzle of their design similar in shape and characteristics to those described in the paper.

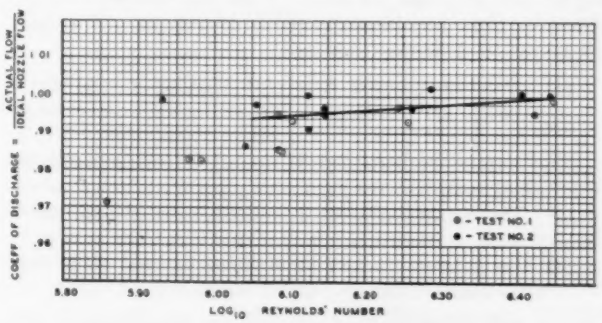


Fig. 27 Calibration of Flow Nozzle, Hudson Avenue Station

A 5-in.-throat-diameter nozzle was installed in the 12-in. condensate return line and calibrated during acceptance tests on two condensers, at which time the condensate was being weighed.

The curve of Fig. 27 shows the results of 20 1-hr test runs plotted against the logarithm of the Reynolds number. Before starting the test it was recognized that two such nozzles would be desirable to cover the range corresponding to the range of test points with relative throat areas in the ratio approximately 2 to 1. The curve, therefore, has been drawn only for the region in which the larger nozzle would be used even though measurements were actually made at the lower flows at small deflections. Full-scale deflection on the manometer was about 30 in. of mercury. Some pulsation of the mercury was observed, but the motion was so slow that fair accuracy of reading was obtained, particularly in the second set of runs, where the frequency of reading was increased from 2 min to 1 min.

These test runs under commonly met service conditions hold

[20] Vice-President in Charge of Engineering, Philadelphia Electric Company, Philadelphia, Pa. Mem. A.S.M.E.

[21] "Accuracy of the V-Notch-Weir Method of Measurement," by D. Robert Yarnall, *Mechanical Engineering*, 1927, vol. 49, no. 1, p. 21. A paper covering the calibration of the flow meters and the description of the apparatus at Richmond, "The Philadelphia Electric Company Calibrates Its Meters," by Frederic G. Ely, *Power*, 1927, vol. 65, no. 10, p. 352.

[22] Turbine Engineering Department, General Electric Company, West Lynn, Mass.

[23] Mechanical Engineering Department, Brooklyn Edison Company, Brooklyn, N. Y.

out a hope of better accuracy for nozzle measurement of condensate flow than has heretofore been generally accepted.

E. D. DICKINSON.[24] Engineers have recognized for a number of years the ever-increasing necessity for a simple means of correctly and accurately measuring the steam used by turbines. There are several reasons that have tended to emphasize this necessity:

(1) The necessity for low-cost production to meet present-day competition has emphasized the need of higher efficiency of prime movers, which are an integral part of manufacturing process.

(2) An increase in the proportion of non-condensing and extraction turbines. It is impracticable to test these by condensing and weighing the steam from the exhaust.

(3) The increase in size of the non-condensing and extraction units, which in consequence emphasizes the importance of saving fuel.

The existing commercial situation would be almost ludicrous were it not so serious. Efficiencies are discussed to a fraction of 1 per cent when it is known that in many cases there has been no recognized practicable means for positively determining the efficiency of the turbine after it is built. The number of non-condensing and extraction turbines accurately tested in the field is negligible.

The paper describes the results of many years of research. This work was the natural outcome of the development of the nozzles used in steam turbines. What the authors have described can in no way be considered an instrument. They have endeavored to show the following:

(1) A nozzle inserted in a pipe carrying steam can be used to give readings of flow with a high degree of accuracy
(2) Nozzles proportioned as described will give reliable and consistent readings with pipes of different sizes
(3) The nozzle coefficient is uniform over an appreciable range
(4) Accurate results can be obtained without calibrating the nozzle when using throat taps.

It goes without saying that all necessary care and precaution must be taken in conducting a test to assure accurate results. The system cannot be considered in any way as competing with commercial flowmeters, but rather might be considered as a natural adjunct. It is not the intention that the nozzle would remain in the steam pipe at all times, for the reasons that it should be inspected before and after each test and that the pressure drop would result in considerable loss if used continuously.

While the system has many compensating advantages, it is not adapted to measuring fluctuating flows. The advantages are:

(1) It can be readily installed at relatively small cost
(2) It can be used to measure superheated or dry steam
(3) Test can be run quickly, reducing the cost of testing
(4) Many points can be run on a curve in a relatively short time
(5) All the steam to the turbine is measured. There is no question of loss in drips or drains. No corrections have to be made for condenser leakage.

S. R. BEITLER[25] AND T. C. BARNES.[26] This paper represents a very important step in the right direction, in suggesting the use of a metering device for acceptance tests of steam turbines. Modern metering practice, however, makes use to a great extent of square-edged orifices in measuring flows of all kinds, and these are believed to be as accurate a metering element as any other. In addition, the venturi tube has long been used as a primary element, and the characteristics of it are very well known. Any method adopted for a standard should, then, not be limited to any particular metering device, but should permit the use of any device of sufficient known accuracy.

It has been the writers' experience that the nozzle with low-side pipe taps is the more desirable for general use as it is much less sensitive and easier to construct. Since the throat taps are placed at the point of maximum velocity, they must be so carefully constructed that it is almost beyond the capabilities of the ordinary mechanic. Any good pipe fitter with proper supervision can install the other type, however, without much difficulty.

In the manufacture of any nozzle, care must be taken to see that the throat is round and that the sides of the parallel portion are actually parallel. It is better to have a slight convergence in the section than any divergence whatsoever. In installing the nozzle care must be taken to have the center of the nozzle in the center of the pipe, and the axis of the nozzle must be coincident with the axis of the pipe. This means that the parallel section of the nozzle must be perpendicular to the flange of the nozzle.

J. C. HOBBS.[27] In order to obtain an accurate picture of the results obtained, the writer would ask two questions:

First, if the desired sizes and arrangement for the measurement of any steam flow can be obtained in each practical case, what accuracy in the final results should be expected?

Second, will the results obtained be acceptable on turbine acceptance tests by the General Electric or other turbine companies without allowance for unknown errors? In other words, if the steam-flow meter after applying all coefficients and corrections still shows a difference from the guarantee, will the meter figure be used as a basis for payment of bonus and penalties?

AUTHORS' CLOSURE

The discussion indicates an interest in this proposal for accurate measurement of steam by means of flow nozzles that is very gratifying. There have been brought out many differences of detail with regard to use of flow nozzles, which reflects the present chaotic situation with respect to them, since nearly every person who measures flow uses a different arrangement of nozzle shape and other details. This presentation of so many differences of detail in the discussion further emphasizes the need for standardization of a single definite system which will thereafter be adhered to by all those who desire to use a flow nozzle for precise measurement of flow. There is presented in the paper a single set of details, mostly selected from the diverse systems in use, which the authors believe most satisfactory for a standard system. These are such as to reduce losses and irregularities to a minimum, so as to give a flow which approaches the theoretical as nearly as possible and so that the discharge coefficient to be used with a rational formula is as nearly unity as possible. It is felt that this is safer than the introduction of items used merely for the sake of convenience, which lower the coefficient, even though a calibration is made to allow for them.

The authors reiterate that the paper discusses only the case of flow-measuring apparatus especially installed for an acceptance test or the like, where certainty of accuracy is the primary consideration. Commercial flowmeters are giving excellent satisfaction in a wholly different territory. Commercial flowmeters

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[27] Painesville, Ohio. Mem. A.S.M.E.

must be suitable for permanent and simple installation in the usual commercial pipe line with small pressure drop, must give a direct reading, and must be easy to maintain. They usually depend upon a vendor's guarantee for accuracy. (If a guarantee of a vendor be acceptable, the guarantee of the apparatus tested by the nozzle might as well be accepted at once, which would eliminate the necessity for a test.) None of these considerations can be allowed to influence nozzles to be used for precise measurement with certainty of accuracy, which is the subject of the paper. The authors feel that a test on the basis described in the paper, conducted by experts, furnishes a flow measurement which may be used with complete confidence in investigating the performance of a guarantee, so that the results, no matter what they may be, will be accepted without hesitation and without question of the accuracy of the test. Attempts to measure flow without the precautions cited in the paper have in the past usually resulted, not in any useful information, but only in discussion as to the accuracy of the test.

Fig. 1 shows to scale the particular case where the nozzle diameter is half that of the preceding pipe. In Fig. 1 the pipe diameter referred to is the actual inside diameter and not the nominal diameter. In Fig. 1, with pipe diameter $2D$, the axial distance from the center of the small radius to the intersection of the inlet curve with the inside pipe wall comes out to be $^{15}/_{32}D$. For pipe diameters of $2D$ or more, it is proposed to have this distance establish the inlet face. The small radius will therefore not be continued to tangency. The intersection with the nozzle face at $^{15}/_{32}D$ shall be rounded with a radius of about $D/20$. In this case, the overall length of the nozzle comes out $1^{23}/_{32}D$, or $1^{15}/_{32}D$ plus $^{3}/_{8}$ in. for nozzles with D less than $1^{1}/_{2}$ in.

With pipe diameters between $1^{2}/_{3}D$ and $1^{9}/_{16}D$, the preceding shape does not give sufficient flange face. Then the overall length of the nozzle shall be reduced on the inlet end to make the nozzle inlet diameter (the intersection of the inlet curve with the face of the nozzle) equal to 1.05 times the actual inside pipe diameter.

In recent use of the system of the paper, and in confirmation of statements in the Keenan and Soling discussion, it has been found desirable to have a cleanout immediately following the nozzle, for inspecting it, and for cleaning it if necessary. This cleanout must be designed so as to keep at least four pipe diameters of unchanged pipe section immediately following the nozzle. In pipe sizes of 4 in. or less, a run of 4 diameters followed by a removable elbow or tee is satisfactory. In larger sizes a special pipe or casting is necessary, with a hand hole and cover, such that the inner pipe contour is not altered. A tee with a blank over the side outlet might create a disturbance of the flow.

It is interesting to note from the Keenan and Soling discussion that the Stevens Institute of Technology is investigating this matter of precise flow measurement. However, their nozzle with diameter only 0.6 in. is very much smaller than anything which would occur in usual flow measurement. It is to be hoped that the Stevens work will be continued with larger nozzles and with direct comparison of the exact details proposed in the paper with others which may be thought better. Their $^{1}/_{16}$-in. pressure holes are, as they state, easily affected by dirt deposits or burrs. Their three throat taps were connected together, and were all drilled into a short arc on one side of the nozzle. This is an unusual arrangement.

This discussion rightly states that a discharge coefficient determined from a throat tap is different from that of a nozzle discharging freely into the atmosphere or other large region. However, the difference is of so small a magnitude that the effect on the discharge coefficient is negligible. This fact was the basis of the statement in the paper that the coefficients for the two cases are identical.

The authors have made a number of tests with a nozzle discharging freely into the atmosphere, with measurement with a draft gage of the difference between the static pressure shown by a throat hole and the pressure of the atmosphere, and have found that with properly formed throat holes the difference was so small that it could not be detected with the draft gage which was used.

The fluid entrained by the jet and flowing toward it from a distance is a very gentle breeze, as is known to any one who has been near a jet discharging into the atmosphere. We have had a great deal of experience with a 20-in. nozzle discharging into atmosphere with differential pressures of about 15 lb per sq in. The entrained fluid had a velocity certainly less than that corresponding to a breeze of about 10 mi per hr. Such a breeze would give a pressure gradient of about $^{1}/_{16}$ in. of water, which is certainly insignificant. It is to be noted that there is a very slight pressure drop between the plane of the throat taps and the end of the nozzle, as well as the very slight pressure gradient mentioned between atmosphere at a distance and the region around the jet. Any difference in coefficients between nozzles using throat taps and nozzles discharging into the atmosphere is due to the difference between these two effects, which seems very small indeed.

Contrary to the statements in the discussion, the region into which the Keenan and Soling small nozzle discharged, or the region into which was discharged the nozzle of the Warren and Keenan A.S.M.E. paper to which they refer, certainly was not a large region, and there probably existed in both cases powerful eddies. In the case of the nozzle in the Warren and Keenan A.S.M.E. paper, it is known that there was a great deal of circulation in the chamber, and screens were introduced to eliminate the effect, as shown in Fig. 13 of the Warren and Keenan paper. Furthermore, this nozzle had a comparatively small entrance radius, so that the jet probably did not fill the throat at high velocities. This gives a coefficient which decreases at high velocities, as is shown in Bureau of Standards Research Paper No. 49, Fig. 12, nozzle A-4.

The area ratio of 11 to 1 of the low-pressure pipe and the nozzle of the Keenan and Soling tests corresponds to a diameter ratio of 3.3 to 1, which certainly gives back flow due to entrainment much more vigorous than with a nozzle discharging into an infinitely large region. Furthermore, the Keenan and Soling low-side pipe taps were located in a corner where they undoubtedly received impact from the back flow of the entrained stream. This is in contrast with both the throat taps and pipe taps recommended in the paper for low-side taps, which are located in regions where the flow is parallel to the walls.

This impact effect, due to the particular location of the Keenan and Soling low-side pipe taps in a region certainly not large, was undoubtedly the reason why they gave a pressure higher instead of lower than that of the throat taps.

For the case of a nozzle located in a pipe of ordinary size, it is of course true that the arrangement of the low-side taps is of the utmost importance. After a careful review of all known methods, the authors proposed the two alternate arrangements given in the paper—viz., throat taps, and pipe taps located according to an exact specification slightly upstream from the plane of the nozzle mouth. These are the only places where a true static pressure can be taken—i.e., by a tap located at right angles to the direction of flow. The only objection to the throat taps—i.e., the high velocity of the jet at that point—is easily taken care of by having good pressure holes and a clean nozzle. This arrangement also requires the minimum amount of calibration to permanently establish coefficients which may thereafter be used with confidence with nozzles following the given specifications.

The authors are definitely opposed to any attempt to measure static pressure in a corner or other region of curving flow, such as

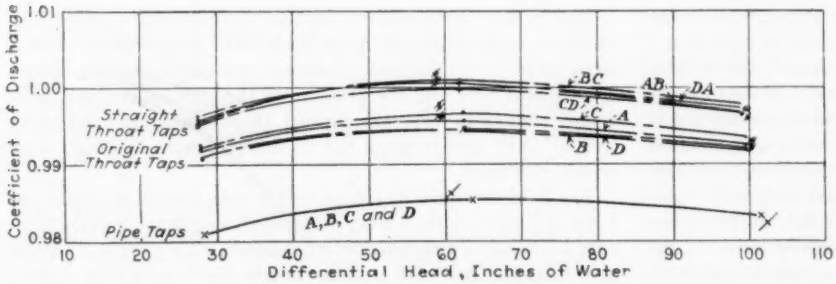


FIG. 28 STRAIGHT THROAT TAPS COMPARED WITH THE THROAT TAPS WITH A RIGHT-ANGLE BEND OF NOZZLE OF FIG. 21

the pipe taps of the Keenan and Soling nozzle. Any pressure so measured is a combination of pressure and velocity effects, and any coefficients based on such pressures have no real meaning, and are also easily affected by inevitable irregularities.

This discussion reiterates the well-known theory that the proper abscissa for plotting of discharge coefficients for nozzles with rounded approach is the Reynolds number. It is not the purpose of the present paper to discuss the pros and cons of the Reynolds number controversy. It is certainly true that no discredit can be cast upon our own tests because of their failure to coincide on a Reynolds number basis with other tests made with wholly different nozzle shapes and pressure-hole arrangements such as the IG nozzle used by Witte. However, Mr. Sprenkle has pointed out that the Reynolds number is not a complete criterion. There have also been a great many other objections to the use of the Reynolds number, previously printed in our own Transactions, such as in the paper by Mr. J. M. Spitzglass, A.S.M.E. Trans., Vol. 52, paper No. HYD-52-7c, and the joint discussion of this paper and papers Nos. 52-7a and 52-7b.

The authors used pressure ratio as the abscissa for discharge coefficient curves merely because it was a convenient way to bring together various tests on a single nozzle with moderately varying pressures and temperatures. In order to bring together tests made on air, steam, water, etc., with a given nozzle, it has also been proposed to use "head" as the discharge coefficient abscissa, which is nearly the same as pressure ratio for a single compressible fluid. This matter of the best abscissa for flow-nozzle discharge coefficient curves certainly needs considerable further investigation.

Mr. Sprenkle gives in Figs. 23, 24, and 25 some comparative, simultaneous readings of throat taps and pipe taps. However, his throat taps, as shown in Fig. 21, had a right-angle bend immediately adjacent to the high-velocity jet. As shown by Professor Allen's classical paper read at the last Annual Meeting ("Piezometer Investigation," Trans. A.S.M.E., paper No. HYD-54-1), a static hole for pressure measurement should be straight and of uniform size for a length from the wall adjacent to the fluid equal to at least two hole diameters. This condition was not satisfied by the throat taps shown in Fig. 21. Fig. 28 shows a third set of comparative readings taken as part of the tests which Mr. Sprenkle cites, but which he does not include in his discussion. The throat taps in this third set of tests satisfied Allen's criterion, and were straight holes of uniform size extending away from the nozzle throat for a distance of several hole diameters. After the tests given in Figs. 23, 24, and 25, posts were inserted in the nozzle of Fig. 21, across the space from nozzle to pipe wall midway between the original short throat taps. There were drilled in these another set of straight throat taps satisfying Allen's criterion. The nozzle throat was then ground and the hole edges dressed. Fig. 28 shows coefficients from simultaneous readings with the three systems of low-side pressure taps. The four upper curves of the figure show the results with the four new straight throat taps. The four middle curves show results with the original short throat taps. Both show the spread which Mr. Sprenkle cites, but this spread has now been reduced to a very small amount by the extra effort to get good holes. The average of the four upper curves, however, is now definitely different from that of the four middle curves, which illustrates the value of Allen's rule. The fact that the discharge coefficient is practically unity shows that the weir used for measuring flow in these tests actually had a calibration coefficient.

The lower curve of the figure is that obtained by Mr. Sprenkle's pipe taps and is appreciably lower than that obtained with proper throat taps, on account of the evacuation and reversed current previously cited. However, the values are lower than those obtained before the insertion of the posts for the straight throat taps. These posts undoubtedly affected the reverse current, the pressure of which is measured by the pipe taps.

Mr. Sprenkle proposes a rearrangement of the terms in the final flow formula to bring it under a general formula for any fluid and any nozzle, including an orifice in a thin plate. Mr. Sprenkle does not give the mathematical basis of the curves he mentions. If, however, these curves are so constructed that they give results identical with the formulas of the paper, there is of course no objection to their use.

FSP-55-11

Stoker Developments at The Detroit Edison Company's Delray Power House No. 3

By PAUL W. THOMPSON[1] and FRED J. CHATEL,[2] DETROIT, MICH.

Experience gained from nearly three years' operation of the large single-ended stokers at Delray has led to developments and improvements in design, the application of which has made them comparable with other types of coal-burning equipment. Improvements have been made with two things in mind—namely, an increase in efficiency of combustion and a reduction of maintenance by decreasing burned stoker parts. All improvements possible, particularly those to improve combustion, have not as yet been completed; however, changes that have been incorporated have materially improved performance. Automatic control of preheated air to the various parts of the fuel bed has been sufficiently developed to establish it as a desirable and valuable aid to combustion, particularly with long stokers burning high-volatile coal. One of the stokers has been operated successfully for a short period with preheated-air temperatures as high as 500 F. Development of a similar air-control system, manually operated, under two boilers in the Beacon Street Heating Plant, handling cold air, has resulted in an increased steaming capacity of 31 per cent and an increase in calculated boiler efficiency of from ½ to 2 per cent. The paper is a description of the Delray installation, but reference is made to the large double-ended stokers, supplied with cold air, and the manually operated metered air control in the Beacon Street Heating Plant.



P. W. Thompson



F. J. Chatel

At the time Delray Power House No. 3 was placed in operation, the latter part of 1929, the five stokers installed were the largest of their type then to have been built. The initial installation consisted of five underfeed stokers, 15 retorts wide and 57 tuyères long. They were rated to burn 34,000 lb of coal per hour with a projected grate area of 611 sq ft when supplied with preheated air at 350 F. Drives were through cog belts from 35 hp d-c motors having a speed adjustment of from 500 to 2000 rpm with drum-type controllers having 21 points of speed control. Each retort of the stoker was equipped with six coal pushers and a movable extension grate. The two-roll clinker grinder was driven by a 5-hp adjustable-speed d-c motor. A cross-section of the boiler unit and the stoker is shown in Fig. 1.

The design of this power station contemplates the ultimate installation of ten boilers to supply steam to six and possibly seven 50,000-kw turbo-generators. The present generating capacity consists of two 50,000-kw units and a 10,000-kw experimental high-temperature unit. A third 50,000-kw unit is now being erected and will be supplied with steam from the present boiler plant.

Selection of Equipment

Fuel-Burning Equipment. The single-ended boilers installed in Delray were a distinct departure from The Detroit Edison Company's present practice. Boilers with double-ended firing have been installed in all the older plants, with the exception of Delray Power House No. 1, which became obsolete several years ago and has been dismantled.

This selection was made to simplify the boiler house arrangement and to economize in building investment rather than because of any dissatisfaction with the double-ended design. With the single-ended boilers arranged in pairs, one coal-conveying system, one row of bunkers, and one firing aisle replaced the three firing aisles, the three rows of bunkers, and the cross-conveyors for coal distribution which would have been necessary with the double-ended type. It was thought that by placing the economizer and air preheater at the back of each boiler, there would be a saving in building height.

These considerations which led to the selection of single-ended boilers indicated a lower first cost. However, a study, made later, on a comparable double-ended boiler plant indicated that such was not the case. The operating convenience of the single-ended design is its only advantage.

The selection of stokers for the initial installation was based on an item which normally would not receive consideration—that of ash disposal. Operating experience with pulverized coal at Trenton Channel and stokers at Marysville had indicated a fairly close comparison between the two methods of firing as to efficiency and ease of operation. Estimates of required investment for the two types of plants indicated total costs so nearly

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[2] Assistant Engineer at the Delray Plant, The Detroit Edison Company. Mr. Chatel received the degree of B.M.E. from the University of Michigan in 1914, and after a short period in the research department of the Detroit Edison Company was made boiler-room engineer at the Connors Creek Plant. In 1917 he entered the U. S. Naval Service as an officer in the Engineering and Ordnance Divisions. Upon completion of his war service he assumed the duties of technical engineer at the Delray Plant, The Detroit Edison Company. During the last two years he has been directly in charge of the stoker development at that plant. He is the author of a paper on "Steam Condenser Practice and Performance" presented at the Annual Meeting of the A.S.M.E. in 1927.

Contributed by the Fuels and Steam Power Division and presented at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

alike as to suggest the question whether the difference was of the same order as the possible errors of estimate.

The absence of any low areas on the Delray site which could have been filled in with powdered-fuel ash and the presence of a local building market which would pay a return of 6.5 cents per ton of coal burned, for stoker ash, formed the basis of the final decision.

The coal-handling equipment and building steel have been

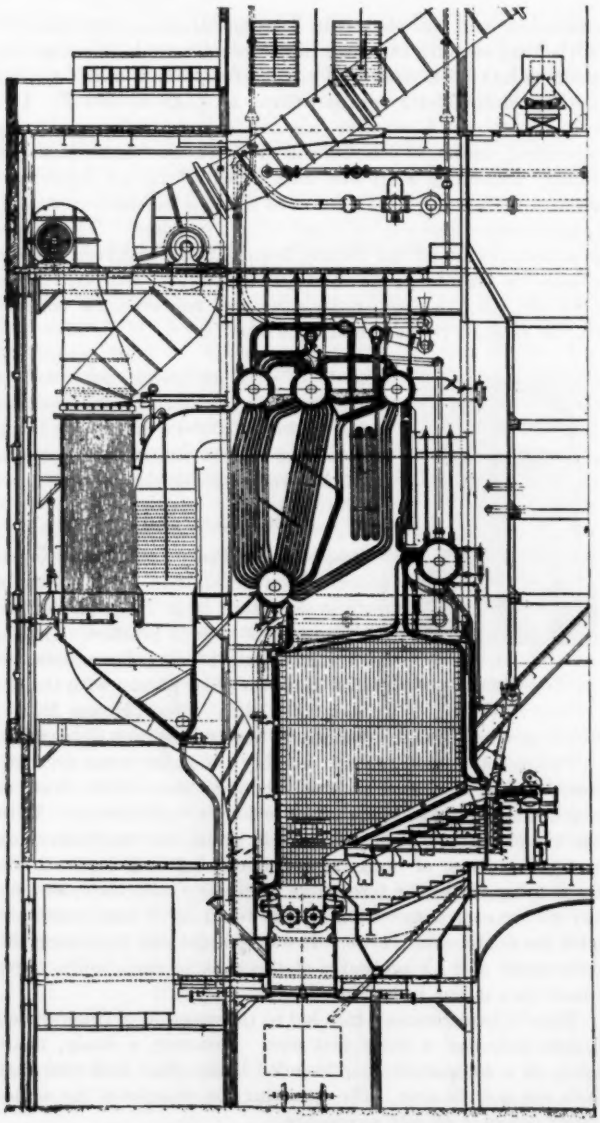


FIG. 1 CROSS-SECTION OF BOILER UNIT AND STOKER, DELRAY POWER HOUSE NO. 3

designed so that pulverized-fuel-burning equipment can be installed should future developments warrant such a change.

Air Supply and Furnace. An upper limit of preheated-air temperature of 350 F was selected as being conservative, although other installations were operating at the time at 400 F or higher. A recent test run, during which the feedwater was bypassed around the economizer, indicated that satisfactory combustion of the general run of coal burned at this plant can be obtained with temperatures closely approaching 500 F. This test was of too short duration (299 steaming hours) to give a definite indication of the effect of this higher preheat temperature on stoker maintenance. An examination of the stoker at the completion of the test showed no burning of parts which could be attributed to the high air temperature, although numerous tuyère plates were found cracked—evidently a result of insufficient allowance for the additional expansion above the normal operating air temperatures.

The problem of proper air distribution, particularly with preheated air under a stoker of this length, was much more complex than had been encountered with previous installations. It was realized that it would be impossible to observe the fuel bed, with the accuracy required, to maintain proper air flow with the hand-operated dampers previously used. The flames from the high-volatile coal would also obscure portions of the fuel bed from any observation doors that might have been installed.

These difficulties were appreciated sufficiently far in advance of this installation to permit an experimental automatic underfire damper control being installed under a stoker at the Marysville Power Plant. Developments of this equipment had proceeded far enough to indicate that a satisfactory arrangement could be worked out for the long stoker, and it was therefore decided to install the automatic air-control dampers under first one, and then the remaining four stokers at Delray.

The combustion chamber was designed with water-cooled walls to eliminate maintenance and increase the steaming capacity. These walls formed the radiant or screen boiler with its water-circulation system entirely separate from the main convection boiler. Above the main furnace chamber and separated by a slag screen was a refractory-lined, secondary-combustion chamber.

The water-wall tubes were covered with various kinds of blocks to temper the cooling effect of the tubes at points where reflected heat was thought to be essential to good combustion. All the front wall tubes, which were bent to form an arch to promote mixing of the lean gas formed at the rear of the stoker with the richer gas liberated at the front, were covered with refractory-faced blocks.

The side-wall tubes were covered with cast-iron blocks to the slag line, and above this point the inner half of the tubes facing the fire was left exposed. The rear wall tubes were covered in the same manner except that the blocks were of the slag-coating type. The upper ends of these tubes were extended over the top of the furnace to form the slag screen previously mentioned.

OPERATION

The 15 retorts are divided into five sections, each section being driven through a two-speed power box. This speed control, combined with a 1-to-4 motor speed and a double-voltage power supply, provides a 12-to-1 crankshaft speed range. This arrangement permits of an accurate coal-feeding rate over a wide range. It is possible with this flexibility to burn only sufficient coal to maintain a floating bank of approximately 50,000 lb per hour, or to generate 430,000 lb of steam per hour. This rate of change of evaporation can be accomplished within a period of less than an hour, provided the boiler is fairly clean and the fuel bed is not too light or irregular at the start.

Rapid changes in coal-burning rates or changes required in sections of the fuel bed are facilitated by the ease with which adjustments can be made to the stroke of the pushers and extension grates. These parts are driven through a bell-crank mechanism in conjunction with each ram, and adjustments are made at the stoker front, outside the plenum chamber. Little time or effort is required on the part of the fireman to make whatever changes are desired.

The individual adjustments of the pusher and extension-grate strokes provide a very flexible means of control. Because of the various combinations afforded by this number of adjustable

units, it is essential to good economy that the operators be able to detect faulty conditions readily and apply the proper adjustments to coal feed, pushers, and fans. Under the present day-to-day load carried on this station only minor and infrequent adjustment of pusher strokes is necessary to obtain good fuel-bed conditions and combustion efficiency. Should there be a sufficient change in the coarseness or moisture content of the coal to affect appreciably the rate of coal burning in a retort or section, it may be necessary to adjust the pusher strokes and the coal-feeding rate to that section. This operation is required, as the automatic dampers which control the air flow over a rather limited range cannot be expected, under such conditions, to keep a uniform fuel bed with an improper fuel-feeding rate or improper movement of the fuel from zone to zone. Appreciable time to correct such defects may be required if allowed to progress, particularly if the correction must originate at the stoker front. The ability to detect faulty conditions at the early stages by observation of the fuel bed or from the operating instruments is quite essential, as irregularities originating at or near the stoker front require more time to correct than on a shorter stoker.

The expected difficulties of fuel-bed observation due to the long flames have more than justified the installation of the automatic air control. This equipment has passed the general development stage, and although refinements are still being made it is now considered a valuable aid to operation.

Under test conditions with and without the air control, a slight advantage is found with the air control in operation. However, under normal operating conditions with similar stokers burning the same coal and operated over a long period, those equipped with air control show a decided improvement. The small difference obtained, with and without the air control during test periods, is the result of the operator's maintaining definite pusher adjustments for any given rating and observing conditions more closely, particularly over the extension grates, than can be expected during normal operation.

The coal, normally burned, is a high-volatile, free-burning West Virginia and southeastern Kentucky mine-run bituminous with an ash-fusion temperature above 2500 F. A typical analysis of this coal is given in Table 1.

TABLE 1 TYPICAL COAL ANALYSIS, DRY BASIS

Proximate analysis:	
Volatile matter, per cent	35.9
Fixed carbon, per cent	56.1
Ash, per cent	8.0
Heating value, Btu per lb	13,750
Ultimate analysis:	
Carbon, per cent	77.4
Hydrogen, per cent	5.1
Nitrogen, per cent	1.2
Oxygen, per cent	7.2
Sulphur, per cent	1.1
Ash, per cent	8.0
Ash-softening temperature, deg F	2500 or above

Little trouble is experienced with clinkers in the fuel bed when proper pusher strokes are maintained. However, improvements have been made to cope with the sheets of slag formed on the side walls and clinkers at the bottom of the tuyère stacks as well as on the extension grates. Special heat-resisting refractories to deflect the slag, formed at high ratings, have been placed in the side walls above the half-tuyères to prevent the formation of slag sheets. The slag runs down over the deflectors and is continuously carried away with the burning coal instead of being built up into large sheets which eventually break loose and foul the fuel bed at the end retorts. The bottom square-ended pushers were replaced with longer beveled pushers which are more effective in cleaning the clinkers from the coal-plate extension and grates.

Stokers are now operated from six to eight weeks before being shut down for stoker or boiler repairs, and it is not unusual to have them on the line for periods as long as three months. Since the plant was started in 1929, it has carried its share of the system load and has continually operated with an average evaporation of 218,800 lb of steam per hour and with a ratio of banked to steaming hours of 24.5 per cent.

Fig. 2 shows the load-duration curve of the stokers for the twelve-month period ending June, 1932. During this period the ratio of banked to steaming hours was 24 per cent.

The movable extension grates, with the parts immediately above and below, have been more susceptible to burning than any other section of the stoker. Redesign of these grate bars to permit freer flow and better distribution of air has reduced the burning appreciably.

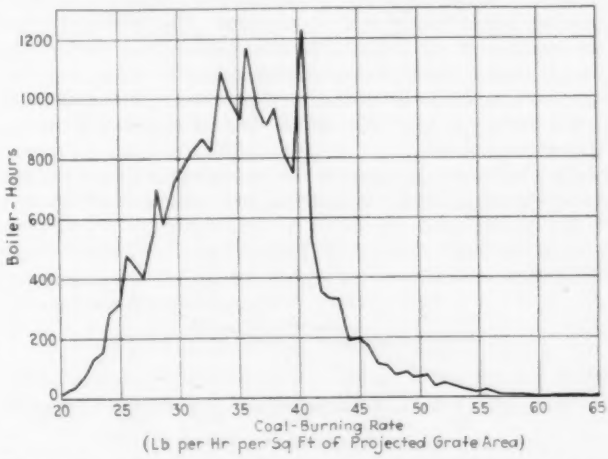


FIG. 2 LOAD-DURATION CURVE FOR STOKERS AT DELRAY POWER HOUSE NO. 3

(This covers all boilers for 12-month period from July 1, 1931, to June 30, 1932.)

STOKER DEVELOPMENTS

The question of stoker developments may well be divided into two parts for discussion. After preliminary operation, it became evident that changes were necessary; first, to improve the combustion and, second, to reduce maintenance, with a resultant increase in continuity of service.

Combustion Developments. The stokers were originally supplied with six square-nosed adjustable-stroke pushers per retort, operated in conjunction with the ram, to push the burning coal, coke, and ashes toward the movable extension grates. With this arrangement the coal was not sufficiently packed at the stoker front and in the retorts, and excessive agitation resulted in smoke. As a result of improper packing, combustion often took place deep in the retorts, which resulted in the burning of iron and the formation of large clinkers over the bottom coal plates and extension grates.

To improve these conditions, lower ram box caps were installed and the bottom pushers were beveled and the nose extended forward as far as practical so that the ends would not extend past the tuyère line on the forward stroke. These changes were an improvement, but it soon became evident that fewer pushers would deepen the retort, produce better coal packing, and enable the operator to maintain a uniform fuel bed with less agitation. Inasmuch as the exact number was questionable, the sectional side cover plates were replaced with one-piece steel plates to obtain straight retort sides, allowing the number and location of pushers to be altered without further retort changes. One stoker was arranged with five pushers and another with four pushers per retort to obtain a comparison.

The five-pusher arrangement showed slightly improved conditions over the original, indicating that it was a step in the right direction, while the four pushers per retort was a decided improvement over the other two. Poorer grades of coal could be handled efficiently. There was less tendency to smoke, and improved construction resulted in a reduction of coal siftings from approximately one-half of 1 per cent to less than one-tenth of 1 per cent. This reduction in quantity of siftings is accounted for by a decrease in the number of joints in the retort construction resulting from replacing sectional castings with one-piece steel side cover plates as well as by the reduction in the number of pushers. Further reduction resulted from a decrease of clearance between stationary and moving parts. The decreased power consumption of the four-pusher arrangement over that of the original design, due to decreased stoker friction and a reduction in excess air, amounted on test to 30 per cent at a steaming rate of 200,000 lb per hour and 38 per cent at a steaming rate of 350,000 lb per hour.

Fig. 3 shows a comparison of the percentage of CO_2 in the flue gas at the boiler outlet obtained on tests of the three different

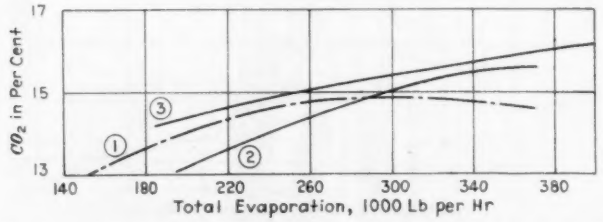


FIG. 3 COMPARISON OF PERCENTAGE OF CO_2 IN FLUE GAS AT BOILER OUTLET

(Test 1, six pushers; test 2, five pushers; test 3, four pushers.)

designs. These data are directly comparable except that better air distribution was provided to the fuel bed and there was less leakage past the control dampers during test No. 3.

Four of the stokers are now equipped with the four-pusher arrangement, while the fifth with the original five-pusher arrangement is still in service.

Design Improvements. The lack of any prior experience with stokers of this size left the designer with only the experience gained from smaller units as a guide. It was to be expected that changes would be required in order to bring the performance of these larger stokers to the same standard of perfection as has been reached through years of development of the smaller units.

After the first few months of operation it was decided to install larger power-box and pusher shearing pins. The forces transmitted through the stoker mechanism had been underestimated and the change was made so that the pins would withstand the normal shearing forces and still give protection to the stoker.

Machine fits and methods employed in driving the pusher mechanisms which had proved satisfactory in smaller stokers were soon recognized as unsatisfactory in this larger stoker. A change was made employing closer tolerances and a clevis-type bell crank instead of the original bar link with cantilever pin. Nearly two years' experience with the improved arrangement indicates that the improved drive needs little attention and the wear on the moving parts is not excessive.

Excessive wearing of the crankshaft journals indicated that the main bearing material was too hard for the material of the crankshafts. Certain gears in the power box also showed excessive wear, and in a few cases pinions were found cracked or broken.

The results of a study to determine the proper material and heat-treatment requirements has been used as a guide in the selection of more adequate materials. The absence of wear on these new parts leads to the belief that there will be no recurrence of this trouble.

The original stoker support plates, made from steel plates 20 in. wide, did not, under the service imposed upon them, provide sufficient strength without sagging. These were replaced with 24-in. steel plates, which were reinforced with the one-piece retort side plates secured to them with close-fitting bolts. This change was made in order to obtain a satisfactory installation of the air-distribution boxes.

Several different types of movable extension grates have been tried, mainly with a view to reducing the amount of iron burned. Better air distribution and cooling of the grates, combined with the ability of the long beveled lower pushers to remove clinkers more effectively, have shown a reduction in replacements.

Changes, other than those made to the extension grates, were incorporated to prevent excessive burning of iron. The stokers were originally supplied with high side-wall air backs which were susceptible to burning. Inasmuch as it was not necessary to supply air at the side walls higher than the plane of the tuyères, these air backs were replaced with standard half-tuyères. Replacements of these half-tuyères are now no greater than that of the full tuyères. Further protection to the side tuyères and the end retorts is afforded by the slag deflectors in the side walls.

The stoppage of coal due to holding up in the supply chutes to any retort has been another serious condition causing burn-outs. Failure of the coal feed to retorts of such depth and length, together with the earlier ignition due to preheated air, has resulted in burn-outs at the stoker fronts. This has been remedied by taking all precautions possible to prevent any foreign matter from being entrained with the coal supplied to each retort. The individual coal chutes to each ram are being replaced with chutes of greater depth which feed to three rams instead of one.

Automatic Air Control. The automatic air-control dampers, which were installed at Marysville as an experiment, were not perfected when the Delray stokers went into operation. Progress was such that it was believed, however, that a design could be worked out that would perform satisfactorily at Delray. This original mechanism was attached directly to the stoker support plates beneath the tuyères, but the coal siftings readily fouled the damper mechanisms, making them inoperative. Correction of this condition was obtained by isolating the moving parts from the siftings with a system of offset air ducts.

This control provides regulation of the air to four zones of each tuyère section. These zones correspond to the sections of the fuel bed controlled by the four pushers, as shown in Fig. 1. Air is supplied from the plenum chamber through an individual control box and air duct to each of the four sections of each tuyère stack, making 56 such boxes per stoker. Neither the side rows of half-tuyères nor the extension grates are equipped with air-control boxes. Additional tests are to be made to determine the advisability of applying such control.

Each control box is equipped with an orifice, a balanced cylindrical damper, and a motor mechanism which regulates the damper opening by power supplied from the drop in pressure across the orifice.

The pressure differential across the orifice, a measurement of the air-flow volume, is maintained practically constant throughout the operating range of the boiler by varying the orifice area. This is accomplished by a separate motor mechanism to which all the orifices are linked.

With the controls in operation and a fuel bed of the desired thickness, the dampers remain balanced. Should a portion of the fuel bed become heavier or thinner, the controls under this particular section will operate to give an increase or decrease of the air quantity. This change will enable the fuel bed to regain its proper thickness, provided the pusher strokes approach the

proper adjustment for the rate and grade of coal being burned.

It has been developed from test that approximately 50 per cent of the total air for combustion passes through the control units, and of this quantity the dampers have a controlling effect up to 20 per cent.

The pressure loss through the dampers amounts to approximately 1 in. of water, which corresponds to a power input of 20 kw to the forced-draft fan.

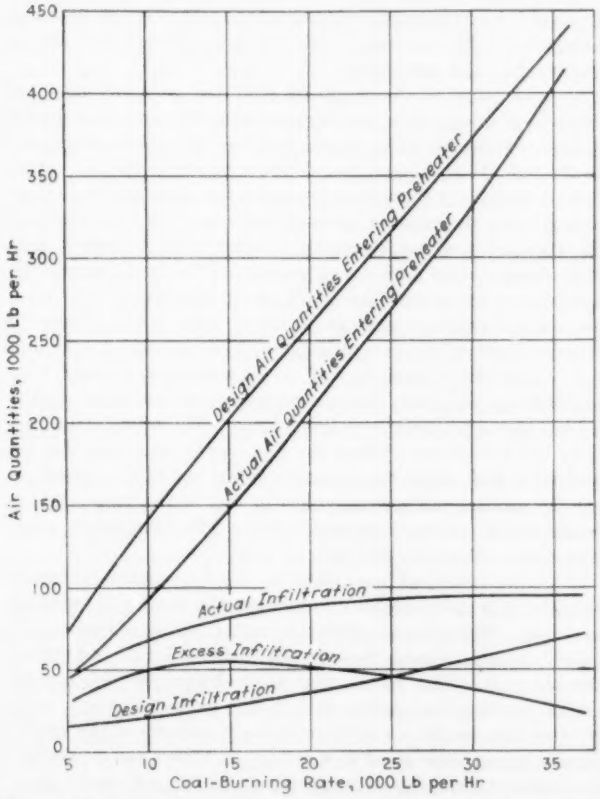


FIG. 4 PREHEATER AIR FLOW AND INFILTRATION-AIR QUANTITIES

STOKER PERFORMANCE

Although the stoker and boiler were designed for a continuous coal-burning rate of 34,000 lb per hour, this rate has been exceeded. It has been possible for intervals of approximately 10 hours, with proper fuel-bed conditions, to burn as high as 45,000 lb per hour.

In Table 2 are given the performance data for maximum and continuous coal-burning rates, based on a coal analysis as given in Table 1. Continuous operation at ratings higher than those given is limited by the accumulation of ash on the slag screen and in the superheater pass. This ash cannot be removed with a hand lance, without reducing the steaming rate.

The overall-stoker, boiler, economizer, and preheater efficiency increased from a yearly value of 82.3 per cent in 1930 to 85.4 per cent in 1931. This improvement is a result of the many changes in design of stoker and air control as well as improvements in operating methods. For the first eight months of 1932 the efficiency is 84.8 per cent, slightly lower than the 1931 figure. Although changes have been made this year that have improved the steaming efficiency, the overall efficiency is less because of the poorer load conditions, with resulting higher banking losses.

Although the gas temperatures leaving the boiler are lower and the CO_2 is higher than predicted, the temperatures of both the flue gases and the air leaving the air preheater are higher than

TABLE 2 MAXIMUM COAL-BURNING RATES

	10 hr	24 hr	Continuous
Boiler:			
Heat output, M Btu per hr per sq ft	15.6	14.5	12.4
Steam output, M lb per hr	430	400	340
Stoker and furnace:	(1)	(2)	(3)
Coal burned, M lb per hr	45.0	41.0	34.2
Coal burned, lb per hr per sq ft of projected grate area	74.0	67.0	56.0
Heat release, M Btu per cu ft per hr	43.0	39.2	32.7
Air to stoker, deg F	378	375	365
Stoker and fan power, kwh per ton	30.3	26.5	21.0

NOTE: (1) Obtained on a stoker having the six-pusher arrangement. It is believed that with the four-pusher arrangement the Beacon Street coal-burning rate of 77.4 lb per sq ft per hour can be obtained. Recent plant load has not permitted a demonstration. (2) This coal-burning rate can be obtained under any expected condition. With favorable conditions this rate may be 45,000 lb per hour. (3) Limited by rating at which boiler tubes may be hand-lanced.

those predicted. This condition appears to be caused by a larger quantity of combustion air being supplied by infiltration and leakage through the ashpit than was estimated for purposes of design of the preheater. The actual air quantities passing through the preheater and through the setting, together with the expected quantities used in the design, are shown in Fig. 4. These quantities were obtained from design data and from a computation of the actual ratio of gas-temperature drop to the air temperature rise taken from operating logs.

AIR CONTROL AT BEACON STREET HEATING PLANT

The stokers under two of the boilers in the Beacon Street Heating Plant have been equipped with manually operated

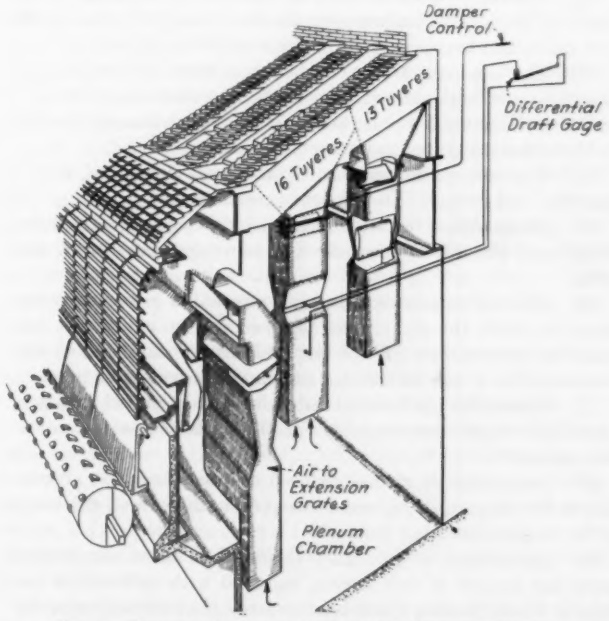


FIG. 5 DAMPER CONTROL UNDER ONE SIDE OF STOKER

metered air control, the general principle of which is quite similar to that of the Delray installation. These stokers are of the underfeed type, 14 retorts wide, with 29 tuyères per retort. There are no air preheaters. Each boiler is served from two stokers which have a combined projected grate area of 636 sq ft, including the common ashpit.

Each stoker is divided into three zones, extending the width of the stoker, for the control of the air. The upper zone is the upper part of the tuyère stack; the intermediate zone is the lower part of the tuyère stack; and the extension grates comprise the lower zone. Each zone of each tuyère section is pro-

vided with a venturi-metering throat and control damper. The extension-grate zone is divided into four sections. A group of differential draft gages, one connected to each venturi-metering throat, is provided as an indication to the operator of the condition of the fire. The control of each individual damper is brought to the boiler gage board so that adjustments can readily be made. Fig. 5 shows the arrangement of this control under one side of the stoker.

The results of tests on these boilers show an increase of maximum steaming capacity of 31 per cent, a maximum coal-burning rate of 77.7 lb per hour per sq ft of projected grate area, including the pit, and a calculated increase in overall efficiency of $^1/_2$ per cent at high loads and about 2 per cent at low loads over a similar boiler without the air control.

At a steaming rate of 400,000 lb per hour, 83 per cent of the total air supplied by the forced-draft fan is metered and controlled by the air-control apparatus. The remainder represents leakage into the furnace through the walls, around pushers, and up through the ashpit. The greater portion is supplied through the lower tuyère section, while about equal amounts are supplied through the upper tuyère zone and the extension-grate zone.

Conclusions

Developments, as a result of operating experience, have led to the following improvements in the design and operation of the Delray stoker:

(1) A change from a six- to a four-pusher arrangement has made it possible to burn efficiently a wider range of fuels.

(2) Clinker formation has been reduced by a change in the shape of the bottom pushers, installation of slag deflectors in the side walls, and better packing of the coal in the retorts.

(3) The amount of iron burned has been reduced by the elimination of high side-wall air backs, the installation of side-wall slag deflectors, more certain coal supply, and improvements to the extension grates and other parts of the stoker.

(4) The automatic air control has been developed into a practical and useful aid to efficient combustion.

(5) Maintenance has been reduced by changes in mechanical design and the use of adequate and properly heat-treated material.

(6) The continuous coal-burning rate of 56 lb per sq ft per hour for which the stoker was designed can be easily met and could be exceeded were it not for the limitation imposed by the accumulation of ash on the slag screen and superheater tubes.

(7) One stoker has been operated for a short period with preheated-air temperatures as high as 500 F without loss of combustion efficiency.

(8) Infiltration of air has reduced the expected heat absorption in the air preheaters, and consequently the overall efficiency of the steam-generating unit.

(9) Installation of manually operated metered air control under the stokers of two boilers, supplied with cold air, in the Beacon Street Heating Plant has increased the steaming capacity 31 per cent and the boiler efficiencies from $^1/_2$ to 2 per cent.

Discussion

Bert Houghton.[3] That the authors' opinions differ radically from mine as to the proper design and operation of stokers does not in any way detract from the excellence of their paper. The basic disagreement is the stress laid upon the deep retort and densely packed coal in the retort, which they claim enables the operator to maintain a uniform fuel bed.

While they have been operating these stokers for three years, all improvements possible, particularly those to improve combustion, have not been completed. The writer does not believe these improvements ever will be completed until they shallow the retorts and do not pack the coal so tightly. With a stoker of large area, the thin, even fuel bed is best for normal operation, and it can be brought up to high continuous ratings much quicker than a thick fuel bed.

The writer has run two boilers side by side—one with a thick fuel bed and deep retorts and the other with shallow retorts and a thin fuel bed. The results vindicated the reliability at maximum ratings of the thin fuel bed and the ability to maintain high ratings continuously and uniformly.

Another point of disagreement with the generally accepted methods of firing is the maintenance of a balanced draft in the furnace. Experience has taught that the manner of getting the air through the fuel bed is best done by pulling it through rather than by pushing it through. To obtain the best results we must have an even thickness of the fuel distributed over the fuel bed. The air must be drawn through the bed in order to keep it light and porous. Deep retorts give an uneven fuel bed, causing the burning coal to separate into dark and bright lanes. The dark lanes in the retorts form logs of seared coke, which appear to be huge clinkers, but in reality have an inner core of coke and coal. After these dense logs of seared coke have formed, they tumble down irregularly over the rear end of the stoker, giving the fuel bed the appearance of a brush heap. The bright lanes just above the tuyère rows where the fuel bed is relatively thin let the greater part of the air through the fuel bed at that point, so that the air does not pass through the fuel bed in proportion to the amount of coal to be burned. These difficulties can be overcome with a thin, even fuel bed.

To obtain increased quantities of steam or higher ratings of the boiler it is only necessary to increase the draft and speed up the stoker. The point at which the draft should be placed at zero referred to atmospheric pressure lies within the fuel bed rather than above it. This means that air is drawn through a good part of the fuel bed rather than being pushed through, with the attendant trouble of uncovering large sections of the grate.

It is common experience that ratings on boilers can be materially increased when the boiler passages and furnace can be scavenged of the hot gases. Where underfire pressure is depended on, the furnace and boiler passages choke up, and with increased pressure under the fire there is actually a decrease in boiler output.

Some people have argued that the maintenance of a high draft in the furnace would require a large increase in the size of the induced-draft fans to handle the larger quantity of hot gases. However, for an increase of minus $^1/_2$ in. to minus 1 in. in draft in the furnace, the increase in gas volume at the induced-draft fan inlet is rather small, ordinarily not amounting to more than a fraction of a per cent, and while there is an increase in the induced-draft fan power, this is more than offset by the consequent reduction in the forced-draft fan power. The thin fuel bed also cuts down the auxiliary power of both induced- and forced-draft fans due to the lesser resistance in the fuel bed. The writer has also been operating stokers equipped with preheated air, with high induced draft, and finds it not only lowers the stoker maintenance, but in this case also cuts down the amount of auxiliary power.

There are two very definite schools of thought on stoker design and operation. This divergence of opinion is a healthy condition. It will lead to a greater understanding of the design and operation of the larger stokers which we have today and of those of the future, which may be even larger. The stoker has come back into its own after having been handled rather roughly for a time by pulverized-fuel installations. The cost of coal prepara-

[3] Operating Superintendent, Brooklyn Edison Company, Brooklyn, N. Y. Mem. A.S.M.E.

tion and maintenance of the mill is far in excess of stoker maintenance per ton of coal burned.

H. D. FISHER.[4] Our experiences have been similar to the authors', and we trust will be improved by similar remedies. We have operated an underfeed stoker equipped with clinker grinder (not of the make described in the paper), of 160 sq ft projected area of underfeed and overfeed sections and 207 sq ft total projected area, for about 3½ years, burning Beckley seam New River coals at rates of 5000 to 6500 lb per hr, the load being practically constant day and night for three- to six-day periods, the variation in capacity being mostly seasonal.

We have found that by far the largest part of the stoker maintenance is required in the high-side tuyères, in the overfeed grate tuyères at the bottom of the retorts, and in the agitator grates at the edge of the clinker pit. During the first 22 months of operation the replacement of sidewall tuyères averaged 0.62 set per year and agitator-grate fingers 1.8 sets per year, while during the next 13 months' operation, at a slightly higher rating, these were, respectively, 1.0 set and 1.9 set per year. The replacement of the overfeed-grate tuyères was too irregular to allow of averaging, but was at a lower total rate than either of the foregoing. Gear-box bushings and shafts have been replaced once, but the gears and jaw clutches showed negligible wear, and no trouble has been found with crankshaft or wristpin bearings. Cost of repair parts was for the first period $0.0453 and for the second $0.0526 per ton of coal burned.

The paper confirms the writer's opinion that sidewall tuyères should be eliminated, and he would appreciate it if the authors will describe in more detail the slag deflectors in the sidewalls, only casual mention of which is made.

J. S. BENNETT.[5] The authors' paper touches on two subjects of vital interest to the power-plant designer; namely, fuel-burning rates per unit of area and the use of highly preheated air with underfeed stokers. With the whole tuyère area of the stoker supplied from a common plenum chamber, the greatest air flow takes place through the spots offering the least resistance to the air flow. This greater air flow burns coal in these areas rapidly, still further reducing the resistance to air flow. This condition may be exaggerated by uneven coal flow from the bunkers.

With air control, the whole stoker area is divided into a number of rectangles. The quantity of air flowing through each rectangle is metered, and if the flow through any section is greater or less than any other section, the rate of flow through that section is modified to bring it in line. The uniform flow through each band across the stoker insures uniform burning and freedom from uneven spots.

The air flow through each section with manual air control is indicated on a gage board. It is a simple matter to increase or check the air flow to any area by control buttons. Automatic control employs the same impulse that operates the gage with manual control to increase or check the air flow.

Air is controlled across the stoker in bands. It is not necessary and is frequently undesirable to have the same air flow through adjacent zones going down the stoker. For example, the air flow through the top 17 tuyères in the stoker may be only half of the air flow through the next lower 17 tuyères, but the air flow through each row of the top 17 tuyères on the stoker should be the same. The need for some form of air regulation has been apparent, but there have been many practical difficulties to be overcome.

[4] Plant Engineer, New Haven Pulp and Board Company, New Haven, Conn. Mem. A.S.M.E.

[5] Mechanical Engineer, American Engineering Company, Philadelphia, Pa. Mem. A.S.M.E.

Air control of the types described by the authors has passed the experimental stage and is now in commercial use in three large central stations under eight boilers. Its use will undoubtedly spread rapidly, as its benefits become fully appreciated, not only in new plants, but in gaining increased capacity in old ones.

Air control has a double purpose; first, to raise the average fuel-burning rate for the whole stoker, while limiting the maximum fuel-burning rates in small spots, and, second, to distribute the air in such fashion that those sections of the fuel bed capable of very high fuel-burning rates are not limited by other sections of the fire, which cannot produce correspondingly high fuel-burning rates.

A conventional stoker is limited in capacity by small spots of excessive fuel-burning rates, equaling 250 lb per sq ft per hr, although the average rate may be only 60 lb per sq ft for the whole stoker. Air control holds the burning rate in any spot close to the average for the whole stoker.

It is hard for a man that has not seen the performance to realize that a stoker burning from 70 to 80 lb per sq ft per hr as an average with air control is not working any harder than the same machine without air control at fuel-burning rates of from 40 to 50 lb per sq ft per hr.

This company has been conducting an exhaustive study of fuel-burning rates using zoned air control. We have found that an average fuel-burning rate for the whole stoker in excess of 90 lb per sq ft per hr can be maintained. This study was conducted on a unit equipped with an air preheater, which supplied air for combustion at from 500 to 570 F. The maximum air temperature which could have been used successfully is not known, as the construction of the heat-recovery equipment did not permit the use of a higher air temperature.

The uniform fuel bed produced by air control would be expected to produce uniform temperatures in a stoker structure. This is confirmed by the fact that a study made by placing 24 thermocouples in the tuyères and other parts of the stoker exposed to the greatest temperature showed average metal temperatures of 600 deg with air temperatures averaging 530 deg and with normal peak temperatures of the metal running from 800 to 900 deg, when the stoker was burning 70 lb of coal per sq ft per hr. These temperatures are considerably lower than similar temperatures which have been obtained on cold-air stokers without air control.

Because it limits the maximum fuel-bed temperatures by limiting the fuel-burning rate in spots, air control is of great value in burning coals containing ash with low fusing temperature.

The authors' statement that it has been found that double-fired units are no more costly than single-fired units should be of great interest to engineers contemplating very large steam-generating units. The authors seem to be supported in this by present practice in large steam-generating units, as it is a notable fact that all units generating over 600,000 lb of steam per hour are fired from both ends.

It is interesting to note that the Delray unit, if double-fired, would give at least 860,000 lb of steam per hour.

JOSEPH G. WORKER.[6] It might be of interest to go back into the history of the development of the boiler and stoker units at Delray No. 1 and No. 2, and then to Delray No. 3 in order to see just what progress has been made.

The conception of the large 2365-hp boiler units in Delray No. 2 design came from a suggestion of Dr. Patchell and experiences at Delray No. 1. This design of Delray No. 2 established a principle of engineering that was persistently adhered to in later plants. These were the largest boilers in the world at that time.

[6] American Engineering Company, Philadelphia, Pa.

This principle of double-ended firing has been followed with all methods of fuel burning for large-capacity units. None of the large units are what might be termed single-fired units; they all follow the Delray No. 2 duplex-firing method.

In the year 1911 a series of tests were conducted on the boilers and stokers at Delray No. 2. These tests were under the supervision of Mr. Alex Dow, past-president of the Society and president of the Detroit Edison Company, aided by Dr. Jacobus and a corps of engineers of national prominence.

Delray No. 2 tests do not exceed 40 lb of coal per square foot per hour or a steaming capacity of 150,000 lb of steam per hour, while Delray No. 3 burned 74 lb of coal per square foot per hour at 430,000 lb of steam per boiler per hour.

The authors state that the boiler-plant cost was as low as any other method within their experience.

The manufacturer of the stokers offered the air control as described by the authors in order to speed up operation to burn 70 or 80 lb of coal continuously per square foot of grate per hour instead of 35 or 40 lb.

The tests in 1911 show that a daily average of about 6000 kw was carried by one of the 2365-hp boilers with 7000 kw, or 8000 kw as a peak. Each boiler at Delray No. 3 carries 34,000 to 40,000 kw, even though the boilers at Delray No. 3 are not much larger in heating surface than the boilers at Delray No. 2.

The figure that the authors mention of 85.4 per cent as the average yearly efficiency is about the predicted guaranteed efficiency at the average operating steaming capacity for the year.

The authors have shown rare judgment and courage in using designs that have in their experiences seemed best to suit their needs, irrespective of engineering fashions.

E. L. Hopping.[7] The paper's treatment of the development in stokers, both in the Delray Power House No. 3 and in the Beacon Street Heating Plant, shows that there has been a marked increase in the coal-burning capacity per square foot of stoker area. This development should add to the possibilities of lower investment cost where stoker capacity is a limiting factor. The zoned air control with manual operation was first applied at the Beacon Street Heating Plant and offers a solution to the control of the fuel bed, which is particularly useful with very wide and long stokers where the usual operating methods depending largely on personal observation are inadequate. At high burning rates the flame density prevents observation of the fuel bed for more than a few feet beyond the observation doors.

There has always been an opinion among power-plant engineers that stokers would not successfully burn the same types of coal that could be used in pulverized form. Stoker manufacturers advised this company's engineers early in 1931 that this was not the fact, and after a visit to the Detroit Edison Company's Beacon Street Heating Plant it was decided to make an experimental installation with zoned air control manually operated on No. 8 boiler at Chester Station. It was hoped to demonstrate not only that the cheaper coals which are available to pulverized-fuel installations could be burned satisfactorily on stokers, but that they could also be burned at higher rates than was the usual practice with better coals. The results of tests made in 1932 on this experimental stoker are offered in discussion of the paper, as they supplement data given therein and it is believed they will be of value to the industry.

No. 8 boiler, which is a 14,217 sq ft Stirling boiler with a 50,276 sq ft tubular preheater, was installed in 1925. The original furnace was entirely of refractory construction. The unit was fired with a 15-retort, 21-tuyère stoker of 310 sq ft projected area and having an ashpit 3 ft 8 in. wide and 5 ft 3 in. deep. A paper presented by Mr. N. E. Funk to the Power Division of this Society at its May meeting in 1926[8] describes tests conducted at that time.

Because of the effect of the high preheated-air temperatures, which reached a maximum of about 600 F, maintenance costs were very high with the refractory furnace walls, and to correct this condition the bridge wall and portions of the two sidewalls were rebuilt with bare steel water-cooled blocks. Tests were made on the unit with this construction and with refractory surfaced blocks, the results of which were published in the Prime Movers Committee Report on Stoker Equipment and Furnaces, N.E.L.A. Publication No. 068, in June, 1930. This showed some improvement in performance, particularly at what was then the maximum continuous boiler output of about 150,000,000 Btu per hr.

The experimental stoker previously mentioned, which was placed in operation in March, 1932, is only 11 retorts wide and 33 tuyères deep, with a projected area of 295 sq ft, and has an ashpit 3 ft 2 in. wide and 6 ft 10 in. deep. It has a diverging slope from the throat to the crusher rolls, which are 18 in. in diameter instead of the 10-in.-in-diameter rolls used with the old stokers. It was necessary to restrict the width of the stoker in order to permit operation at high burning rates because of the limitation in steaming capacity of the boiler unit, which was fixed at approximately 200,000 lb per hr, equivalent to an output of about 240,000,000 Btu. To accommodate this narrow stoker it was necessary to construct a considerable bench in the side wall. A slag lip was provided on each of the side walls and at the top of the bridge wall. These slag lips minimized slag formation along the side and bridge walls. The stoker is provided with sectionalized air control in three zones. The first zone takes in approximately the upper half of each tuyère bank and provides 11 individual air inlets with adjustable throats and manually operated dampers. The intermediate bank is similar to the first, but is applied to the lower part of each tuyère bank. The third zone takes in the extension grates and is divided horizontally into three compartments, each with its own adjustable throat and manually operated damper. The damper in each air inlet is controlled by a pushrod located on a control panel. The panel is also equipped with differential pressure gages that indicate relative flow through all the air inlets.

One ram and two pushers were originally installed in this stoker, but after experimental runs the two pushers, each of which was 10 in. in height, were replaced by three pushers, each only 6 in. in height. This alteration resulted in considerable improvement in the fuel bed and materially reduced burning of tuyères and side plates.

In the first experimental runs there was a substantial amount of soot deposited at the rear of the boiler and at the bottom of the preheater. A considerable quantity of the particles blown from the fuel bed also deposited against the bridge wall. In an effort to improve the stoker efficiency, soot was removed from the rear of the boiler and the preheater and deposited in a hopper above the boiler; then fed by gravity back into the stoker hopper. To prevent the piling up in the fuel bed at the bridge wall, air was introduced between the water-cooled blocks in this wall about at the normal level of the fuel bed. This air permitted almost complete combustion of the material and improved operating conditions at this point. So far as could be determined by test, the introduction of air in the bridge wall had no effect on the CO_2.

A series of tests indicated that the best operating conditions were obtained when the relative air ratios through the controlled zones were 15 per cent through the first zone, 50 per cent through

[7] Mechanical Engineer, Philadelphia Electric Company, Philadelphia, Pa.

[8] "Comparative Performance of Air Preheaters," by Nevin E. Funk, Trans. A.S.M.E., 1926, vol. 48, p. 337.

the second, and 35 per cent through the third, or extension-grate zone. Analysis has indicated that about 65 per cent of the total air for combustion entered the furnace through these controlled zones, 15 per cent through the bridge wall air boxes, and 20 per cent by leakage through the stoker.

The tests for the new stoker were all made with the foregoing air ratios with soot returned to the hopper and with the air admission through the bridge wall. The soot returned was not included as part of the fuel.

Attention is called to the 500-deg air temperature leaving the preheater, which is somewhat unusual for good performance with stoker firing. The tests have indicated that no serious increase in maintenance cost would result from continued operation with this temperature.

With the old stoker the maximum output was about 150,000,000 Btu per hr, which may be considered as a continuous rating, since the tests were 24 hours in duration. On the new stoker the maximum output was 250,000,000 Btu per hr, but the conditions in the fuel bed indicate that these ratings could be maintained satisfactorily only for periods of about 4 or 5 hr. At the end of that time there was an excessive amount of smoke emission from the stack. Because of load conditions at the plant it was not possible to run the tests on the new stoker for greater duration than 10 hr, but from the condition of the fuel bed at the end of the lower rating tests, it is believed that a continuous output of at least 215,000,000 Btu can be maintained with satisfactory operation. This represents a dry-coal burning rate of 60 lb per sq ft of projected grate area, which is a substantial increase over the 45 lb burning rate possible on the old stokers.

It appeared that feeding soot with the coal to the grate tended to increase CO in the furnace and also appeared to slightly increase the fly refuse loss up the stack. This increase in CO and fly refuse loss, however, was greatly overbalanced by the saving accruing from returning the soot. The CO observed at the boiler outlet varied from 0.03 to 0.3 per cent, while the CO_2 was fairly constant in the tests at about 15 per cent.

Since the test runs were for periods of 10 hr or less, the radiation and unaccounted-for losses were estimated. The estimates, however, were based on former tests conducted on this boiler, and the efficiencies thus secured were checked by efficiencies based on measured coal and water.

The maintenance cost as indicated from the test period was not excessive as compared to any of the other stoker installations on the system. Carborundum plates have practically eliminated the burn-out trouble previously experienced on extension grates. Some tuyères were burned at the point immediately over the division between the first and second air zones, where it is possible that the higher air pressure in one zone caused a down draft into the adjacent zone. The manufacturers believe that trouble at this point, although not serious, can be reduced.

This stoker demonstrated its ability to burn some of the cheaper coals carrying ash with low-temperature softening characteristics, which are available in this territory but heretofore could not be satisfactorily burned on stokers. The coal selected and used in the tests was a Central Pennsylvania coal having the following proximate analysis:

Moisture, per cent	4.5
Volatile (dry basis), per cent	20.1
Fixed carbon (dry basis), per cent	74.0
Ash (dry basis), per cent	5.9
Sulphur (dry basis), per cent	1.4
Heat value of coal (dry basis), Btu	14,800
Initial softening point of ash, deg F	2100
Final fusion point of ash, deg F	2550

This coal was selected from among a number that were satisfactory for slag-tap pulverized-fuel furnaces but not for stoker firing, and has a price range of 25 to 40 cents per ton lower than that for good stoker coals. During experimental runs on the stoker, this particular coal gave a great amount of trouble. It produced clinkers which fused to the tuyères and the side plates of the stoker, resulting in excessive burning of stoker parts, and it was difficult to maintain a satisfactory fuel bed. It differs from good stoker coals principally in the initial softening point of the ash, which is about 2100 F, whereas the coals which are burned successfully on standard stokers have ash with an initial softening temperature which is usually above 2400 F, or within 100 to 150 deg of the fusion temperature of the ash.

The experimental runs were made with this coal and with two good stoker coals, resulting in an efficiency at 240,000,000 Btu output of about 70 per cent with the coal used in the final test and 74 per cent with the good stoker coals. In the final tests there was a substantial improvement in the performance where at a rating of 240,000,000 Btu an efficiency of about 82 per cent was obtained.

Summarizing the improvements in the stoker and related apparatus, it is found that the efficiencies at high ratings are the result of the following changes as compared to the older stokers:

1 Use of carborundum plates on the extension-grate section
2 Use of larger clinker grinders with deeper ashpits having diverging slopes and with positive crushing between the extension grates and the bridge wall
3 Zoned air admission with control and indicated performance on a panel in front of the boiler
4 Deeper ram boxes and better fitted interior parts
5 Improved pusher design with ability to increase or decrease the stroke of each individual pusher from the front of the stoker
6 Reclamation and burning of soot which normally is thrown away with the ash
7 Introduction of air at the bridge wall which completes the combustion of particles blown to this section of the fuel bed
8 Use of slag lips on the side and rear walls to prevent accumulations along the side and back of the fuel bed
9 General improvements in mechanical strength of stoker parts.

P. Nicholls.[9] This discussion will be confined to the fuel bed and the authors' description of the changes made to the stoker. In common with the majority of papers that deal with underfeed stokers, this one describes the changes made and gives the relative over-all results obtained before and after the changes, but does not tell the reason why they were made or the line of argument that suggested that improvement in operation would result.

If one is dealing with overfeed or hand-fired furnaces, the changes are related to the burning action in the fuel bed, to the gases arising from it, or to the mixture of these gases with secondary air, which factors again the authors may relate to the fuel being burned; also the logic of the changes may be more or less explained. Problems connected with gas producers and with powdered fuel are usually treated in a similar manner.

The reason for this difference is that we have a fairly clear conception of the process of combustion in these types of burning, but there is no corresponding knowledge of underfeed burning. It may be that those connected with the design and operation of underfeed stokers know more than they tell, or is it that they have to depend on guess and on trial and error? At least the lack of reasoning makes it difficult to follow and to classify the data given in papers, and leaves one in the dark as to whether

[9] Supervising Fuel Engineer, Pittsburgh Experiment Station, U. S. Bureau of Mines.

the lessons taught can be applied to other conditions and other coals.

It is granted that the actions in a fuel bed of an underfeed stoker are much more complex than those in an overfeed and that the burning is a mixture of overfeed and underfeed. In other types of burning the travel of the pieces of fuel can be followed, but the writer has not yet seen more than an approximate suggestion of the paths followed in an underfeed stoker. As previously stated, the actions could only be completely covered by tracing the life history of the various streams of coal from the time they leave the pusher to their discharge as ash; this would include their ignition and the loss of combustible as related to their location.

The U. S. Bureau of Mines investigated the process of combustion in overfeed burning, and the results, published in Technical Papers 137 and 139, have been referred to as classics by a recent writer.[10] We have desired to supplement this work by a similar study of underfeed burning; the complexity of the problem and its many graduations were a deterrent, but an investigation has been under way for some time, and it is expected that a first report will soon be presented. Because of its complexity, a start was made by studying pure down-burning without side flow of the fuel; combined with this work has been a study of the effect of preheated air on the combustion of both over- and under-feed beds—another subject on which data are lacking.

It is evident that the factor of rate of ignition of the fuel is all-important; a fuel cannot burn any faster than it is ignited, and the rate of ignition is not proportional to the rate of air supply, whereas the rate of burning is approximately so until such a high rate is reached that the fuel bed becomes so thin that the oxygen is not consumed and excess air appears in the gases. Although the actions in an underfeed stoker are far from being a pure down-burning, yet this principle underlies them and an understanding of it should be a guide in interpreting observations and planning changes.

It would be helpful if the authors could explain the line of reasoning followed in making the changes to the Delray stokers. Was it possible to predict that the motion of and the burning of the fuel could be improved by modifying the motion of the entering coal?

FRANK S. CLARK.[11] Any one who has visited the Delray Station cannot but recognize the care and thought put into the design of the plant and also appreciate its scrupulous cleanliness. Where there is good housekeeping, there is invariably good operation.

There are a few questions which the writer would like to ask. It is stated that "considerations which led to the selection of single-ended boilers indicated a lower first cost. However, a study made later on a comparable double-ended boiler plant indicated that such was not the case. The operating convenience of the single-ended design is its only advantage." Were the comparisons made using a Stirling-type boiler only or was the sectional header boiler considered? It is appreciated that the Delray Station was designed about four years ago and that considerable advance in the art of combustion has been made since that time. The writer would like to know, however, if a low rate of heat release in the furnace was assumed or if the analyses were made on the basis of the higher heat-release rates such as are now feasible.

In relation to the method of fuel burning, whether in pulverized form or on stokers, it is stated that "Estimates of required investment for the two types of plants indicated total costs so nearly alike as to suggest the question whether the difference was of the same order as the possible errors of estimate." Were these comparisons made using the bin and feeder system for pulverized coal and, if so, was the preparation plant housed in a separate building as is the case at Trenton Channel? With the pulverized coal, was an extremely moderate rate of heat release considered, as is the case at Trenton Channel, or were the comparisons made on the basis of some of the more modern forms of forced-draft pulverized-fuel burners with high furnace-heat release?

Fig. 2 shows a load-duration curve for the stokers for all boilers for the 12-month period from July 1, 1931, to June 30, 1932. The continuous rated capacity of the stokers is given as 56 lb per sq ft of grate surface per hour. Referring to Fig. 2, one sees that the stokers were run at this rating for a total of only about 40 hr during the year. Were any measurements taken of the amount of coal blown from the grates when burning coal at the rate of 56 lb per hr; also, was the very small use of the continuous fuel-burning capacity of the stokers due to the conditions of the load or to some others such as the accumulation of ash on the slag screen and superheater tubes as is mentioned in item 6 of the conclusions? What measures do the authors consider necessary to take in order that the continuous fuel-burning capacity of the stokers might be attained over long periods of time?

I. E. MOULTROP.[12] The authors' study of the relative advantages of stokers versus powdered-coal equipment for their conditions shows that the two types of fuel-burning equipment are about on a par. This is not surprising to some of us who have been carefully watching the developments in both powdered-coal equipment and stokers.

It is of great value to have on record the statement of the engineers of so successful a large utility as the Detroit Edison Company that there is very little difference between the two types of fuel-burning equipment and that the decision is governed by the disposal of the by-products, This is most gratifying to those engineers who have had faith in the stoker and have had the courage to carry on with it in the face of a popular hysteria for powdered fuel.

Now at a time when no one is increasing his plant or putting in new equipment, we can sit down and think over the advantages and disadvantages of the various types of fuel-burning equipment in advance of future needs. The opinion seems too general that each type has its advantages and disadvantages and that the choice will have to be governed largely by local conditions.

The efficiency of the stoker when properly designed and operated has always been good, but there has been ground for improvement in maintenance cost. Apparently the Detroit Edison Company has done much in this direction. The most notable improvement, and one which will have far-reaching effects, is the air control to the fuel bed. When one considers that the modern stoker having an area of 200 sq ft or more is placed above a plenum chamber with a uniform pressure of air in it, is it any wonder that adequate control of the fire is lacking? The slightest variation in the thickness of the fuel bed will immediately change the air distribution in the wrong way. If there is a spot in the fuel bed that is a little thinner, then much more air will rush through, whereas the supply should be reduced to avoid burning a hole. Conversely, the same thing is true when the fuel bed becomes thickened in a locality.

With this condition it is not surprising that maintenance of stoker parts in contact with the fire is heavy. Proper air control should not only practically eliminate this defect, but should

[10] "On the Mechanism of the Combustion of Fuel in Industrial Furnaces," Prof. Morris W. Travers, Jour. Inst. of Fuel, vol. 6, Oct., 1932, pp. 45–48.

[11] Consulting Engineer, Boston, Mass.

[12] Chief Engineer and Superintendent of Construction Bureau, Edison Illuminating Company, Boston, Mass. Mem. A.S.M.E.

improve the economic performance. The thin spot tends to permit high excess air and lower the CO_2.

Manual control of the air is a long step in advance, but what is really needed, and it is going to come in time, is an automatic control which will insure that every square foot of fuel bed will have automatically supplied to it the exact amount of air required. When this is done, we will have a very much improved stoker performance.

There has been much discussion lately as to the relative advantages of the thin versus the thick fire on the stoker. The manufacturers and most of the users believe in the fairly thick fuel bed.

The tendency of the present day to install very large boiler units has forced the engineers to either use powdered fuel or to put in abnormally large stokers. There has been a tendency to increase the size of the stoker too rapidly, as materially lengthening the stoker brings in factors of unknown value and difficult to meet in operating.

The alternative is to put in short stokers, one at each end of the boiler. When the first installation of this kind was made at the 59th Street Station of the I.R.T. by the late Mr. H. G. Stott, the writer thought the principle was a mistake and would lead to increased investment costs. Mr. Stott undoubtedly did the only thing he could to get increased output from his boilers, but there have been very few double-end installations made unless there has been some controlling factor demanding it. The writer wonders if Mr. Stott was not looking farther ahead than we realized, and if, in place of some of these tremendously long stokers recently installed, it would not have been better to put in two much shorter stokers, thereby keeping the fire under better control, even though the fuel-supply system was a little complicated and the installation cost perhaps a little higher. This is a subject that should be carefully studied by engineers where they have a problem of furnace design for a very large boiler.

It is noticed in the technical press that engineers on some of the recent power plants across the water have expressed the thought that the stoker or the powdered-fuel equipment, as the case may be, was the controlling factor of the evaporating plant and that they are showing a tendency to put the responsibility for the boiler-room installation on the manufacturer of the fuel-burning equipment. This is going a little too far in taking the responsibility off the boiler manufacturer and placing it on the stoker men, but the idea behind the thought is good.

Power-plant engineers have been too prone in the past to not sufficiently consult the stoker builder before determining on the boiler, superheater, and economizer. The stoker manufacturer is vitally interested in what is going to be put above his stoker to evaporate the water. The proportions of the evaporating apparatus and their relative location have a bearing on the fuel-burning equipment, and the advice of the stoker manufacturer should be sought and given equal weight with that of the boiler manufacturer.

H. E. KLEFFEL.[13] Can one at the present time sell the stoker ash at a rate of 6.5 cents per ton of coal burned?

Referring to Table 2, the coal-burning rate in pounds per hour per square foot of projected grate area for 10 hr is 74 lb, for 24 hr 67 lb, and for continuous operation 56 lb; the continuous coal-burning rate is limited according to item 6 of the authors' conclusions by the accumulation of ash on the slag screen and superheater tubes, and as this accumulation is due to the fine particles of coal, coke, and ash being carried up through the furnace by the gas stream, have the authors considered screening the coal and providing the stokers with fuel of a given size range and using the fines in their pulverized-fuel-fired plants? This is a question of balancing the gains, if any, against the cost of screening, handling, and transportation.

[13] New York, N. Y. Assoc-Mem. A.S.M.E.

At the Delray Plant the maximum continuous burning rate is 56 lb per sq ft of grate per hr, with a heat release of 32,700 Btu per cu ft per hr, whereas at the Beacon Street Heating Plant the maximum continuous burning rate is 77.7 lb of coal per square foot of grate per hour, with a heat release of 41,100 (see p. 283 of December issue of *Power*). It would be interesting to have a comparison of the two furnace designs and an outline of the arrangement of the furnace heating surface and the boiler heating surface exposed to the furnace of the Beacon Street Heating Plant boilers, and the authors' suggestions as to how the Delray furnaces could be rearranged to overcome the limitations imposed on the stoker.

AUTHORS' CLOSURE

The preference expressed by Mr. Houghton for a thin, porous fuel bed with the combustion air drawn through it, rather than forced through it, indicates that he may have in mind an extremely heavy fuel bed. The authors agree that a fuel bed can be too thick and too densely packed for best results, but it is apparent that a thin fire with high gas velocities is the other extreme.

The thickness of a fuel bed for best economical and operating results depends largely upon local conditions. The type and size of stoker, kind of coal burned, fan capacity, and the excess-air requirements are items which have a bearing in determining the proper fuel-bed thickness. Although the deepened retorts at Delray result in more coal packing, there are no "dense logs" and control of the fuel bed has been definitely improved. The time required to obtain the maximum boiler rating has not been decreased and the efficiency has been increased.

With a normal fuel bed the rate of evaporation of the Delray boiler can be increased to the maximum desired quantity practically as fast as the stoker and fans can be brought up to the corresponding speeds. The data in Table 3, taken from heat-loss tests on one of the Delray Power House No. 3 boilers, show the effect of thin fires on excess-air and total fan-input quantities. The pressure at the top of the combustion chamber was maintained practically constant, so that the air infiltration quantities through the setting up to that point would be the same. The draft loss through the gas passes of the boiler increased with the excess-air quantities, which undoubtedly resulted in a greater amount of air infiltration through the second and third passes of the setting.

TABLE 3

Test No.	Evaporation, 1000 lb per hr	Excess air, per cent	Drafts: Air pressure under fire, in. of water	Drafts: Top of combustion chamber, in. of water	Drafts: Boiler outlet, in. of water	Total fan input, kw
1.....	250.4	19	1.4	0.07	2.5	141.2
2.....	251.5	25	1.4	0.08	2.6	143.1
3.....	258.0	32	1.2	0.06	3.0	169.3
4.....	252.1	43	1.2	0.05	3.5	213.8

Although the settings have been made reasonably tight, tests show that a considerable portion of the air for combustion throughout the normal operating range does not go through the air preheater. Lower furnace pressures would undoubtedly further increase the air infiltration unless the settings are absolutely tight.

It is the authors' contention that atmospheric pressure is reached in the fuel bed when operating either with balanced draft or a relatively high underpressure in the combustion chamber. With balanced draft, this point is at or near the surface of the fuel bed, while with higher drafts it is probably somewhat below the surface in a fuel bed of equal thickness.

Since the stack action of the hot gases begins at the fuel bed in either case, the only essential difference between the two methods is the fact that with balanced draft the atmospheric point is again reached in the combustion chamber, while with high drafts it is not. Since experience at Delray and other stoker-fired plants of The Detroit Edison Company indicates that operation with high drafts increases the air infiltration and puts more load on the induced-draft fan, the balanced-draft method is employed.

Those changes which were made to the stokers and described in the paper were to correct certain obvious deficiencies, the two most common being the inability of well-trained operators to control the fuel bed properly and the excessive formation of clinkers. Although the original installation of six square-nose pushers resulted in satisfactory combustion at moderate ratings when burning coal with an ash-softening temperature of 2500 F or above, the clinker formations at the stoker front, on the tuyères, and coal and tuyère extension plates were excessive when poorer grades of coal were burned. The clinkers at the front wall and the long clinkers which invariably capped over the tuyères indicated clearly that the coal was coking too low in the retorts. Various pusher-stroke combinations were unsuccessfully tried in an effort to keep the burning fuel off the grates. With six pushers, the stroke reduction required on successive pushers in the retort to obtain the necessary packing and control was such that it would have required exceptionally long top-pusher strokes. Since ignition takes place early on this stoker, it would have been difficult to maintain the proper fuel bed at the stoker front with longer pusher strokes. It was evident that an increased reduction of successive pusher strokes should be obtained by reducing the number of pushers rather than by making changes to obtain longer strokes.

The fuel bed, controlled with long strokes and a large number of pushers, was subjected to excessive agitation. With certain grades of coal which required long strokes, the clinker formation on the tuyères was rapid, and it became difficult, and at times impossible, to prevent the emission of smoke from the stack. It was apparent that fewer pushers would disturb the coal and coke less frequently and the deepened retort would furnish means of packing the coal, so that with proper pusher strokes the desired coal movement and shape of fuel bed could be obtained at all times.

Evidently these predictions were correct, as the entire performance has been improved. Pusher-stroke combinations can readily be found to properly handle coals that were difficult to burn efficiently before the change. Clinker formations at the stoker front are practically eliminated, and the clinkers over the tuyères are reduced in size so that the flow of air is not obstructed. Replacement of internal stoker parts has been reduced, particularly the tuyère side plates, which were formerly burned due to the burning of coal in the retorts. The long beveled pusher at the bottom improved the disposal of clinkers that adhered to the bottom tuyères over the coal and tuyère-plate extensions.

In reply to Mr. Clark's question regarding the cost comparison of different boiler plants, the authors desire to point out that the comparison was made between actual costs of a plant as constructed containing single-ended boilers and a plant containing double-ended boilers. This latter type of boiler used in the comparison is one which has been installed in four earlier plants of The Detroit Edison Company, and considering the many years of experience obtained and the estimators' familiarity and experience with costs applicable to such design, the authors are confident that these cost comparisons represent the true facts as stated in the paper—i.e., that there was no appreciable saving in investment by adoption of the single-ended boiler. All comparisons were made on boilers of the Stirling type. In both cases a high rate of heat release was contemplated, and the furnace was designed accordingly.

The small use of the continuous fuel-burning capacity of the stokers was due entirely to the conditions of the load on the plant. Were conditions such as to require a continuous coal-burning rate in excess of 56 lb per sq ft per hr, special soot-removal equipment would be needed, because, with the average run of fuel, slag would accumulate on the screen tubes during a period of from 10 to 20 hr, requiring removal by hand-lancing. Such procedure is very difficult to accomplish without reducing the rating to a point where operators can do this cleaning effectively. The character of the load on the boilers, however, does not demand continuous steaming at high rates, and during normal day-to-day operation advantage is taken of the low-load periods to do the necessary cleaning with a hand-lance. As far as the stokers are concerned, a continuous coal-burning rate of 75 lb per sq ft per hr or more can be maintained.

During the past three years the market for stoker ash has fallen off to such an extent that whereas the figure of 6.5 cents per ton of coal burned represents the average selling price of stoker ash over a long period of time at or near Detroit, the present price averages only about 1.0 cent per ton. There is reason to believe, however, that as the demand for building materials revives, the price will increase.

The complications and additional costs which would be involved in screening the coal used on the stokers rule out such procedure, as the removal of slag from the tubes is not a costly or difficult operation and is performed at times when the maximum capacity of the boilers is not needed.

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Utica	Public Library
Yonkers	Public Library

North Carolina

Chapel Hill	University of North Carolina (Engineering Library)
Raleigh	North Carolina State College

North Dakota

Fargo	North Dakota State College
Grand Forks	University of North Dakota

Ohio

Ada	Ohio Northern University
Akron	Public Library
	University of Akron
Canton	Public Library
Cincinnati	University of Cincinnati
	Public Library
	Engineers Club of Cincinnati
Cleveland	Public Library
	Case School of Applied Science
	Cleveland Engineering Society
Columbus	State of Ohio Library
	Public Library
	Ohio State University
Dayton	Engineers Club of Dayton
Toledo	Public Library
	University of Toledo
Youngstown	Public Library

Oklahoma

Norman	Oklahoma University
Oklahoma City	Public Library
Stillwater	Oklahoma Agricultural and Mechanical College
Tulsa	Public Library

Oregon

Corvallis	Oregon State Agricultural College
Portland	Portland Library Association

Pennsylvania

Allentown	Free Library
Bethlehem	Lehigh University
Easton	Public Library
	Lafayette College
Erie	Public Library
Lewisburg	Bucknell University
Philadelphia	Engineers Club
	Drexel Institute
	University of Pennsylvania
	Franklin Institute
Pittsburgh	University of Pittsburgh
	Engineers' Society of Western Pennsylvania
	Carnegie Institute of Technology
	Carnegie Library (Schenley Park)
	Carnegie Free Library of Allegheny
Reading	Public Library
Scranton	Public Library
State College	Pennsylvania State College
Swarthmore	Swarthmore College
Villanova	Villanova College
Wilkes-Barre	Public Library

Rhode Island

Kingston	Rhode Island State College
Providence	Brown University
	Providence Engineering Society
	Public Library

South Carolina

Clemson College	Library, Clemson College

Tennessee

Kingsport	Public Library
Knoxville	University of Tennessee
Memphis	Goodwin Institute
Nashville	Vanderbilt University

Texas

Austin	University of Texas
College Station	Agricultural & Mechanical College of Texas
Dallas	Public Library
El Paso	Public Library
Fort Worth	Carnegie Public Library
Houston	Rice Institute
	Public Library
Lubbock	Texas Technological College (School of Engineering)
San Antonio	Carnegie Library

Utah

Salt Lake City	University of Utah
	Public Library

Vermont

Burlington	University of Vermont

Virginia

Blacksburg	Virginia Polytechnic Institute
Charlottesville	University of Virginia
Norfolk	Carnegie Public Library
Richmond	Virginia State Library

Washington

Pullman	State College of Washington
Seattle	Public Library
	Engineers Club
	University of Washington
Spokane	Public Library
Tacoma	Public Library

West Virginia

Morgantown	West Virginia University

Wisconsin

Madison	Library, University of Wisconsin
Milwaukee	Public Library
	Board of Industrial Education, Vocational School Library
	Marquette University

Wyoming

Laramie	Wyoming University

Progress in Hydraulics

Hydroelectric Plants

AN IMPORTANT gain during the year has been the fuller realization of the importance of hydro in public-utility systems, as well as the dependence of hydro upon having such load systems to feed into. This emphasizes the interdependence between steam and hydro, and as an evidence of interest in this respect, there should be cited the excellent paper by Irwin and Justin presented at the Semi-Annual Meeting of the A.S.M.E. at Bigwin last June.[1] This paper, although dealing with the economics of hydroelectric development in general, confined itself more or less to two factors subject to evaluation, the energy and capacity values of hydro development, but did not deal extensively with the operating advantages in the way of accommodation to systems and the maintenance of frequency.

A great amount of study is being given by companies that are faced with such problems to making the proper balance in everyday operating practice between the sacrifices that have to be made in steam and hydro in order to secure greater over-all economy. The carrying of peak swings on hydro results in reducing kilowatt-hours of output that could have been generated in operating the plant at a uniform discharge just equal to the river flow. At the same time, this sacrifice is more than made up by the assistance which it renders to the steam system, not only in the saving of investment generally associated with the problem of economics, but also in the daily peak prepared for, as well as the adjustment of the weekly steam schedule to favor particularly the boilers and to keep uneconomical plants off the line as much as possible. This advantage is becoming so important that in the Eastern part of the country many of the large-load centers supplied primarily by steam have important supplements of hydro.

For example, the seaboard cities of New England are receiving energy from the water powers located inland where the rivers are developed at many available sites, Philadelphia derives a large amount of power from Conowingo, while Baltimore likewise receives benefits from Susquehanna plants at Holtwood and Safe Harbor. The trend of development in this way is clearly shown by the interconnections under way for bringing hydro power from upper New York State to New York City and the connection soon to be placed in service between the high-tension line from Safe Harbor to Baltimore and Washington.

In general, it may be said that interconnection is favorable to hydraulic development because it widens the opportunity of making use of the mutual advantages between two sources of power.

The outstanding large development in the West is the Rock Island Plant of the Puget Sound Power and Light Company (Washington Electric Company) on the Columbia River near Wenatchee, Wash. Fig. 1 shows a shop assembly of one of the Rock Island turbines.

The outstanding large development in the East is Safe Harbor, which is unique in many respects, in addition to the installation there of the most powerful Kaplan turbines in the world. Engineering studies were made here and in Europe preliminary to this development. These investigations led to such innovations as the use of double leaf gates, testing by current meters, streamline rack bars, etc.

The contract for these turbines was carried out jointly by the Baldwin-Southwark Corporation, I. P. Morris Division, and by the S. Morgan Smith Company. Fig. 2 shows a shop assembly of one of the Safe Harbor runners.

The official opening of the hydroelectric power station on the

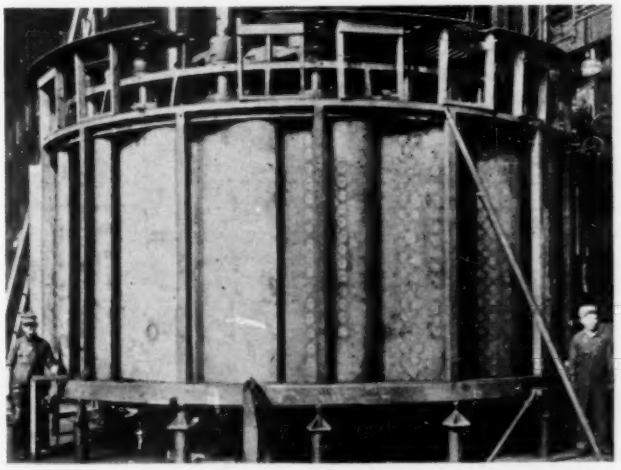


Fig. 1 One of Four 21,000-Hp, 32-Ft-Head, 100-Rpm, Single, Vertical Concrete Spiral-Cased Turbines With Cast-Steel, Six-Blade, Adjustable Nagler-Type Runners Plate-Steel Guide Vanes, for Washington Electric Co., Rock Island Plant

(Photograph by courtesy of Allis-Chalmers Manufacturing Co.)



Fig. 2 Kaplan Adjustable-Blade Runner for Safe Harbor; 42,000 Hp, 55-Ft Head, Blades in Open Position

(Photograph by courtesy of Baldwin-Southwark Corporation, I. P. Morris Division.)

[1] K. McH. Irwin and Joel D. Justin, "Economic Balance Between Steam and Hydro Capacity."

Presented at the session of the Hydraulic Division, Wednesday afternoon, December 7, 1932, during the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Personnel of the Executive Committee of the Hydraulic Division for 1932: Blake R. Van Leer, Chairman; Carroll F. Merriam, Secretary; L. F. Moody, Byron E. White, E. M. Breed, and D. J. McCormack.

Dnieper River, known as Dnieprostroi, took place on October 1. The first unit was put into operation on May 1 last and another four have been finished and gradually tested since then. At the official opening there will be five turbo-generators ready for operation. It will be recalled that the engineering on this project was done by engineers from the United States and that the turbines were built in this country by the Newport News Shipbuilding and Dry Dock Company.

The improvement in the development of propeller-type turbines and the adaptation of the movable-blade idea of Mr. Kaplan into American practice is outstanding. The efficiency characteristic obtainable with this form of construction has opened up a new opportunity for hydro development, in cases not hitherto economically feasible for the Francis turbine or even for the propeller-type turbine with fixed blades.

Practically all of the manufacturers have been interested in attractive variations of the movable-blade turbines. Some of these have blades adjusted by hand from time to time, some propose regulating entirely by the movement of the blades, thus eliminating the usual turbine gates, and another would have the

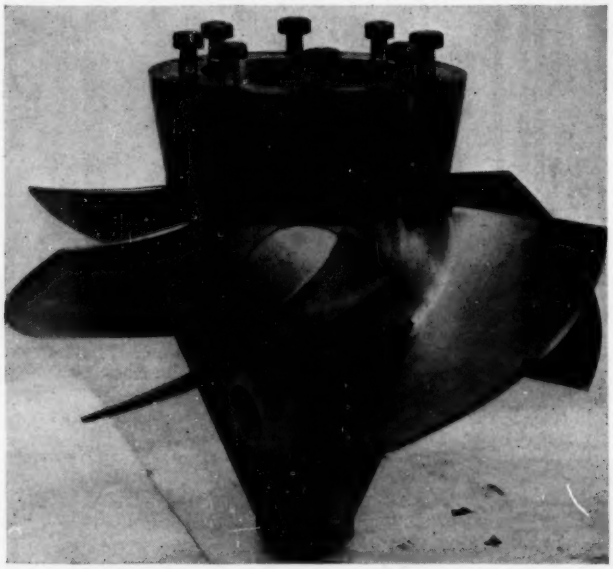


FIG. 3 MODEL OF ADJUSTABLE-BLADE TURBINE IN WHICH THE POSITION OF THE BLADES IS CONTROLLED BY THE FLOW OF THE WATER THROUGH THE TURBINE
(Photograph by courtesy of Newport News Shipbuilding & Dry Dock Company.)

blades adjusted entirely by the flow of the water through the turbine. A model of this last type is shown in Fig. 3.

Although little progress has been made during the year in pumped storage plants, some effort has been devoted to investigating their possibilities and to experimenting with the use of hydraulic prime movers as pumps, thus deriving the benefits of pumped storage without the necessary cost of providing separate pumps, reservoirs, etc. (See F. A. Allner's paper, A.I.E.E. Convention, Baltimore, October 12, 1932.)

There is increasing appreciation that the incremental cost per kilowatt of hydro development compares very favorably with that of steam; and so with the growth of power systems, and increasing sharpness of the peaks that must be carried as a part of the duty of a public utility to its customers, we may look to hydro as a means of securing a limited amount of peak capacity.

In order that hydro may serve to supplement steam to the best advantage, it should be able to perform the important function of governing. Large interconnected systems in the East are looking to the low-head run-of-river plants, especially those having comparatively short water passages, to control frequency. Considerable improvement has taken place in design of the governing systems, especially in the arrangement of the component parts. One plan is to connect the governors of adjacent units in pairs in order to combine the advantages of the central pumping system with those of the unit system. Fig. 4 shows a

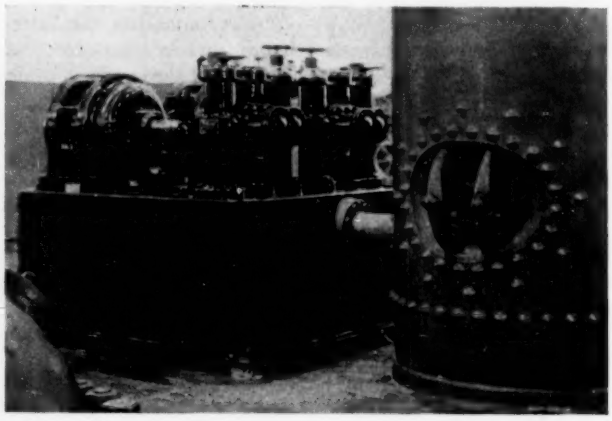


FIG. 4 COMPACT ARRANGEMENT OF GOVERNOR PUMPS, SUMP TANK, AND ACCUMULATOR
(Photograph by courtesy of Woodward Governor Company.)

compact arrangement of pump, sump tank, and accumulator to reduce the amount of piping necessary.

The introduction of the propeller turbine in general and the Kaplan wheel in particular has opened up new possibilities in the development of low heads which would be impossible with wheels of lower specific speed. It has been pointed out in a paper presented before The American Society of Mechanical Engineers at its Annual Meeting, December 5, 1932, by Prof. W. Spannhake, that in order to take advantage of these possibilities it will be necessary to pay special attention to details of design.

Kaplan wheels are particularly adapted to carry swinging loads without serious loss in efficiency. In the future these wheels will possibly be operated with an effective speed drop much less than has hitherto been thought possible without a sacrifice in stability of the unit while operating in parallel with others. It is also possible through the governor to give the unit very desirable operating characteristics.

Considerable attention has been given to the operation of hydro units as synchronous condensers. Such practices increase the usefulness of a turbine when operating on a large system.

With the advent of the Kaplan unit it has been very important to find some method of securing a comparative measure of the amount of water used. For this purpose it is not necessary to know the absolute quantity of water as long as a comparative measure may be found to indicate on an arbitrary scale the economy secured. There are several methods for doing this, but recent experiments have shown that highly satisfactory results have been obtained by observing the pressure differences at various points in the wheel casing.

In the field of testing by actual water measurement a great deal of progressive work has been done with current meters, using a method which is an adaptation of that commonly used in Europe.

As a result of experience, improvement in technique has been made which allows the field and office work to be reduced.

Certain power companies have reported very gratifying results from long-time forecasting of water supply by cyclic analysis. These companies are located where conditions are probably as

favorable as in any part of the country, and although their success cannot be taken as proof that the method is generally applicable, it does show the possibility of solving the mystery of cyclic variations in meteorological phenomena and that success may possibly be attained in other localities when the confusing influences are correctly analyzed.

Gratifying results have been secured from model testing and the development of new runner designs based on experimentation. During the year a unit has been installed, breaking the world's record for high efficiency of a unit tested in place. This success was the result of the surprisingly high efficiency attained by the commercial-size model tested in the Holyoke flume.

Fig. 5 shows the shop assembly of one of the three units in this plant.

A laboratory designed especially for studying cavitation has been exceedingly busy deriving the characteristics of runners and studying the influences that are conducive to cavitation. At the same time there is being conducted at the Massachusetts Institute of Technology an extensive series of research tests to determine the fundamental principles that govern the phenomenon. Attention is being given especially to the pressure distribution in the zone of cavitation and the influence of cavitation on various metals and protective coatings. The formation of the cavitation bubbles and their collapse is being studied by means of spark photography.

Particular attention is being given to two important subjects: the economical operation of plants and simplification of power-plant design. Material is being collected from companies that are active in promoting not only efficiency of operation, but also saving in operating and maintenance expenses and in the better use of available water supply. Another group is gathering a large amount of information and suggestions regarding the means that have been taken in recent plants for securing more economical design, either by outright reduction in first cost or by savings to be effected as the result of better design. The material collected so far is evidence of the activities which plant designers have shown in their endeavor to secure a more

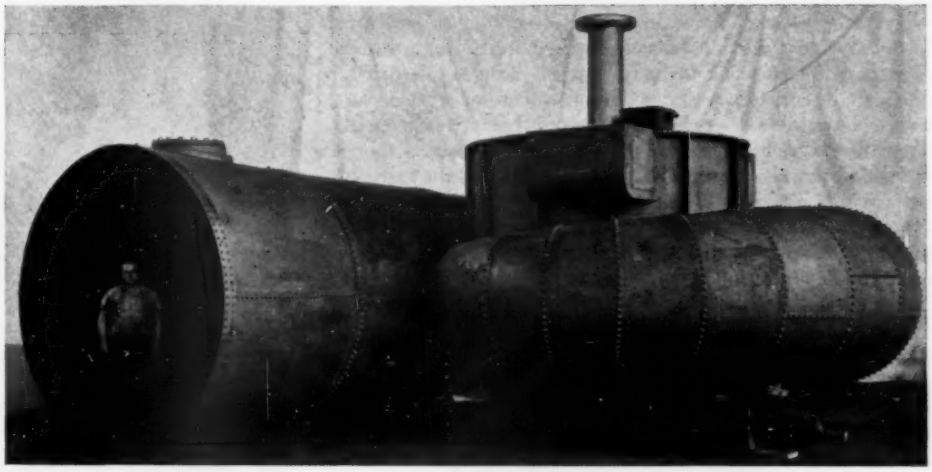


FIG. 5 SHOP ASSEMBLY OF ONE OF THE THREE UNITS IN THE EAGLE PASS PLANT; 4500 HP
(Photograph by courtesy of James Leffel & Company.)

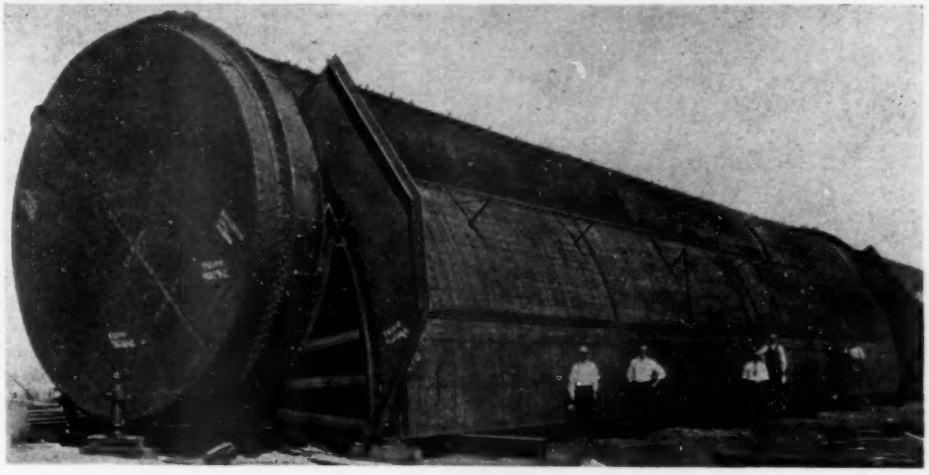


FIG. 6 ROLLER GATE, 26 × 100 FT, FOR MISSISSIPPI FLOOD CONTROL
(Photograph by courtesy of S. Morgan Smith Company.)

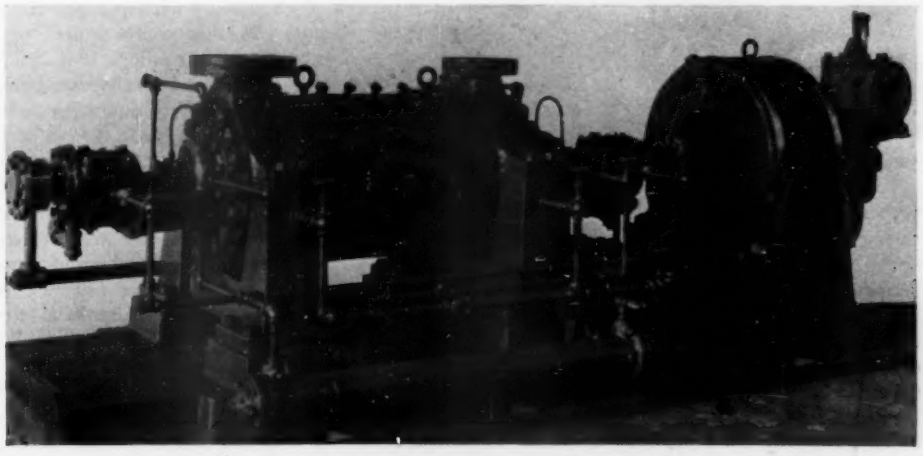


FIG. 7 EIGHT-STAGE PUMPS FOR HOT-OIL SERVICE
(Photograph by courtesy of Ingersoll-Rand Company.)

economical development of hydro resources. It is encouraging to note that in a plant recently built (Eagle Pass) it was claimed that attention to architectural details had not added appreciably to the cost.

Hydroelectric engineers have shown a keen interest in securing better and more durable concrete for hydraulic structures, and as a result we can look to better understanding of the factors which are essential for building durable structures.

Progress on the Hoover Dam was recorded early in November, when the Colorado River was diverted into the bypass tunnels in the canyon walls, this marking the completion of the first stage of the work, said to be a full year ahead of schedule.

Interest was also aroused by the announcement (see *Power*, August, 1932, p. 101) that the contract had been let for 55,000 tons of steel-plate work for four main headers, 30 ft in inside

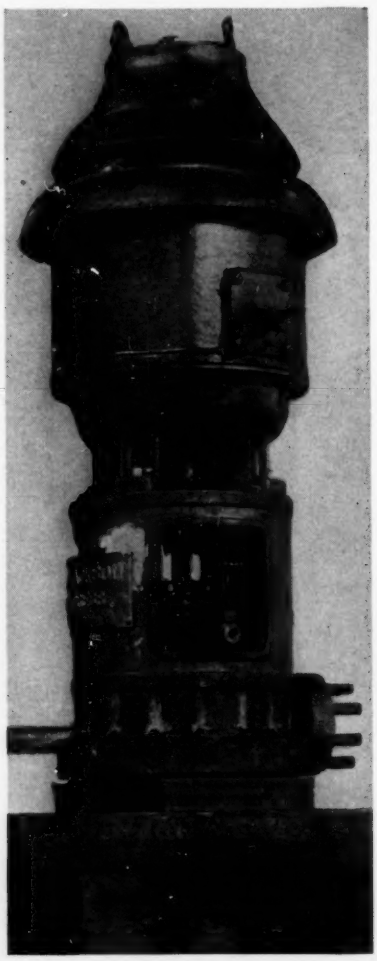


FIG. 8 ONE OF TWO HYDROPRESS PUMPS INSTALLED AT THE GRAND CANYON
(Photograph by courtesy of Byron Jackson Company.)

diameter, each one of which will feed four 13-ft penstock outlets, which will carry the water to the 16 turbines. The plate is to be welded to conform to the current rules for the fusion process of welding, Class 1, A.S.M.E. Boiler Construction Code for Unified Pressure Vessels.

A most important contract was received by the S. Morgan Smith Company from the United States War Department for eleven roller gates to be installed on the Mississippi River for flood control. These gates will have a damming width of 100

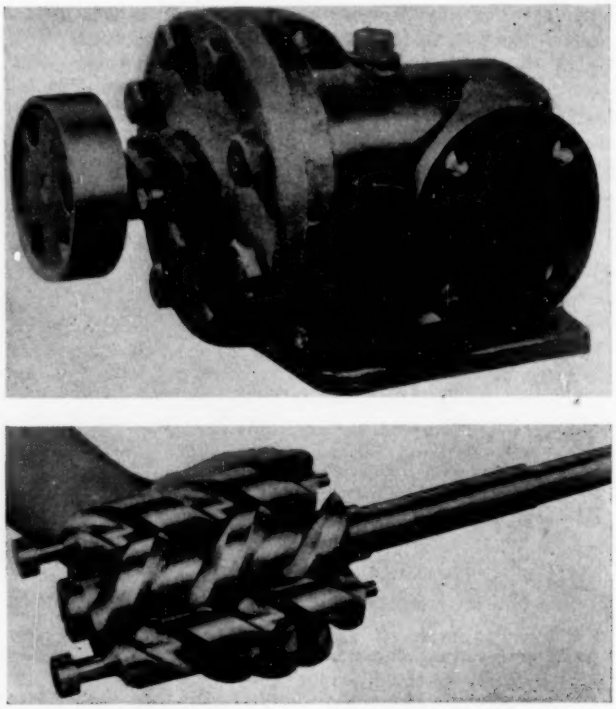


FIG. 9 ASSEMBLED UNIT AND ROTORS OF A POSITIVE DISPLACEMENT PUMP
(Photograph by courtesy of DeLaval Steam Turbine Company.)

ft and the largest of them is 26 ft high. Fig. 6 shows one of these gates before installation.

PUMPS

As in the case of water-power development, there has been a limited activity in the development of pumps and pumping plants. There have been a number of installations with new features of design and some having special interest to the technical man.

Some noteworthy advances in the development of high-head centrifugal pumps have been made. Hot-oil charging pumps up to 4800-ft head in one case and up to 7200-ft head in two cases, with efficiencies reaching 70 per cent, have been reported. A multi-stage high-pressure hot-oil pump has been built in various sizes and stages which develop pressures up to 1500 lb when handling 100 to 800 gal per min at temperatures ranging up to 750 F. Such a pump is illustrated in Fig. 7.

Further encroachment on the reciprocating-pump field by centrifugal pumps is shown by units furnished for the Grand Canyon, where they are pumping 90 gal against a 3400-ft head from the bottom of the canyon to the top. One of these pumps is shown in Fig. 8.

In the last year practically every manufacturer of electric motors has completed an extended line of 3600-rpm motors, which include horsepowers up to and including 200. With these motors now available, it has made the introduction of high-speed single-stage pumps possible for heads in excess of 650 ft.

It has become generally recognized that adequate absolute pressure on the intake side of a centrifugal pump is necessary for satisfactory operation. Studies and experiments have been made of the problem both here and abroad and have served to establish the fact that attainable limits depend upon, and in fact are fixed by, the specific speed of the pump. It seems impractical at this time to predict from any mathematical analysis at what absolute suction heads cavitation begins. A group of manufacturers of centrifugal pumps has agreed to accept certain limita-

tions in specific speed which will have an adequate margin of safety, based on experience. Charts have been developed which show the upper limit of specific speeds for double-suction, single-stage centrifugal pumps at a normal water temperature of 80 F and operating under heads from 45 to 350 ft. For heads greater than 350 ft, considerably more experience will be needed, even though there are a number of installations in actual operation with heads as high as 1000 ft per stage.

A new rotary-type positive-displacement pump has been developed. The power rotor has a convex tooth contour, which fits the concave contour of the idling rotors, providing an almost fluid-tight closure. The idling rotors are driven by fluid pressure, making pilot gears and bearings unnecessary and requiring only one stuffing box. High speeds for direct connection to motors or turbines, pressures up to 4000 lb, and high efficiencies are advantages claimed for this pump, illustrated in Fig. 9.

bed loads. Work has already been started on these problems.

Experimental seamless steel and copper tubing in 4 in. and 6 in. diameters has been obtained for the pipe-bend project, and various types of joints are being studied with a view to causing the least possible disturbance of flow due to the joints.

Other research problems proposed for study in the National Hydraulic Laboratory are: Coefficients for orifices and weirs of all types; the hydraulic jump; flow around bends in pipes and channels; laws of movement of detritus; loss in pipe lines and fittings; wave phenomena; cavitation; scale effects, etc. Model tests have been proposed on flow in chutes and transition sections; erosion below dams; studies of siphon spillways; entrainment of air in tunnels; backwater due to various structures; silting of canals, etc.

Tulane University has established a hydraulic laboratory for the study of fundamental hydraulics.

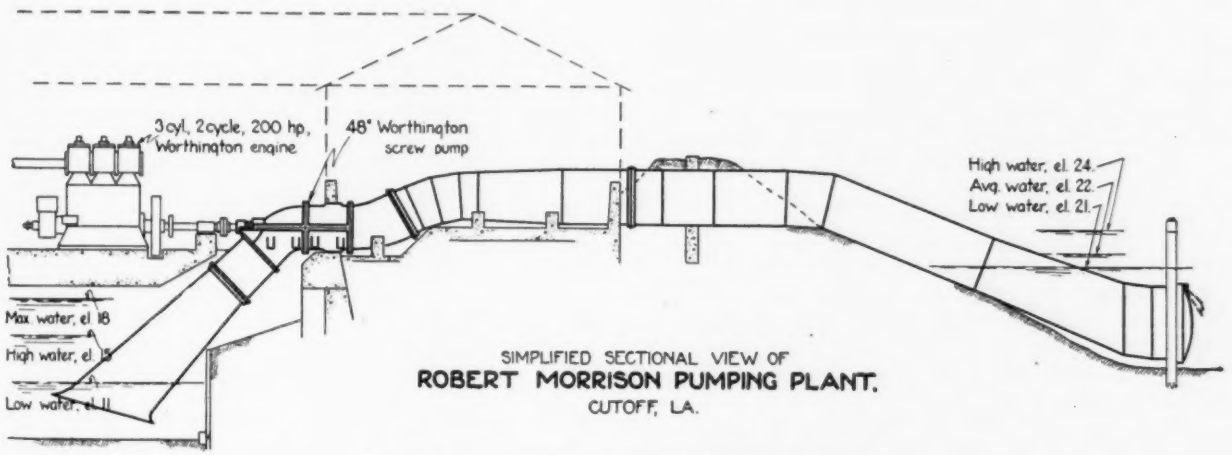


FIG. 10 INSTALLATION OF LOW-HEAD PROPELLER PUMP
(Drawing by courtesy of Worthington Pump & Machinery Corporation.)

In the low-head field (and by that is meant heads below 45 ft) a number of installations of the propeller-type pump have been made. Their application has not become so general, however, as to cover all the possible variations in pumping conditions for which they are said to be suitable. Experimental research is proceeding, but as an example, practical operating data are not yet available to establish the relationship between absolute suction pressure and specific speed. A 48-in. propeller pump under test delivered 52,400 gpm against a dynamic head of 9.8 ft with an efficiency of 78 per cent. Fig. 10 shows the installation.

HYDRAULIC LABORATORIES AND RESEARCH

The outstanding item of interest in this field is the announcement of the completion of the National Hydraulic Laboratory at the Bureau of Standards in Washington, D. C., and the initiation of the first experimental projects. The building of this laboratory was completed in March of this year, and the installation of equipment began immediately. The laboratory has been described in *Mechanical Engineering* and in the technical press.

About twenty research problems have been submitted to the laboratory by several bureaus of the Federal Government for study at the earliest possible date. These include an investigation of the loss of head at bends in large pipe lines, the study of means for preventing the silting of an irrigation reservoir, methods and appliances for measuring the depths of swift and deep rivers, the effect of turbulence on current meter ratings, study of deep-well current meters, the velocities necessary to transport sand particles of various sizes in streams, and a study of the

The new University of Tennessee Hydraulic Laboratory features a complete hydroelectric unit assembly.

Construction work has been started on a new hydraulic laboratory for the Newport News Shipbuilding and Dry Dock Company. It is expected that this laboratory will be ready for operation in March, 1933. All modern facilities for testing turbine and pump models and component parts of hydraulic-power installations are provided for. Provisions are also being made in this laboratory for the study of cavitation and pitting.

The laboratories of the various hydraulic machinery and equipment companies have been carrying on experimentation for the improvement of their specific lines, and considerable progress as a whole can be indicated. One line of laboratory experiment is on the development of reversible pump-turbine units which may operate efficiently either as a pump or as a turbine. One model reported suited to high heads developed 85 per cent as a pump and 89 per cent as a turbine. Another line of experimentation is in the development of propeller-type pumps. An axial-flow propeller pump with centrifugal discharge has shown good efficiency and high specific speeds. One company has been developing a type of adjustable-propeller runner for turbines in which the vanes are adjusted by the flow of water through the turbines.

Since research operations are expensive in money as well as in time, the possibilities of carrying on any worth-while studies are limited very materially by the reduced income from production. However, this survey of activity shows that during this period considerable development and experimental work is being carried on to place the hydraulic industries in a better position to take advantage of increased business activity in the future.

HYD-55-2

A Study of the Data on the Flow of Fluids in Pipes

By EMORY KEMLER,[1] PITTSBURGH, PA.

It was not until 1883, when Reynolds by dimensional analysis developed the equations for dimensional homogeneity and dynamic similarity, that any scientific basis for analyzing the experimental data on the flow of fluids was available. In view of the fact that only a small part of the experimental work previous to that of Stanton and Pannell has been reduced to Reynolds' dimensional basis, it was deemed desirable to undertake the present investigation in order to get a more complete knowledge of the flow problem, and in it an attempt was made to obtain all the data available on flow in pipes under pressure.

Some of the more important conclusions to be drawn from the data presented are:

1 The friction factor is independent of the fluid flowing for the same values of the Reynolds number *R*.

2 The friction factor for drawn brass pipe varies only ± 5 per cent from an average value and is substantially independent of the pipe size and fluid flowing for the same value of *R* within the range encountered in practice.

3 The friction factor for new steel pipe varies with the size, and varies about ± 10 per cent from an average value.

4 The data on used and artificially roughened pipes show that if the roughness is not large enough to cause contraction and enlargement losses, the friction factor never exceeds the value 0.054.

5 The mean hydraulic radius can be safely used as a criterion only in the turbulent region.

6 Special care should be taken in the selection or determination of the diameter.



I—INTRODUCTION

IT WAS not until 1883, when Reynolds[2] by dimensional analysis developed the equations for dimensional homogeneity and dynamic similarity, that any scientific basis for analyzing the experimental data on the flow of fluids was available. Little attention was given this work by engineers until the last few years. The early experimenters, starting with Couplet in 1732,[3] attempted to develop empirical formulas to fit their experimental results. The result was that, while formulas were developed that fitted the data, they were applicable only to a particular pipe size within a given velocity range, and only to one fluid. The work of Stanton and Pannell[4] in 1914 on the flow of fluids through smooth pipes was the first real advance in the application of the method of dimensional analysis to fluid-friction problems which arise in engineering practice. Their experiments on compressible and incompressible fluids showed that this method of analysis was practically as well as theoretically correct. Since that time there have appeared a number of researches on the flow of fluids which have demonstrated that this method of handling the problem is satisfactory.

In view of the fact that only a small part of the experimental work previous to that of Stanton and Pannell has been reduced to Reynolds' dimensional basis, it was deemed desirable to undertake the present investigation in order to get a more complete knowledge of the flow problem.

II—TYPES OF FLUID FLOW

Reynolds[5] found from his experiments on the flow of fluids that there are two types of fluid motion. These experiments were made by drawing clear water through a transparent tube and admitting a very small stream of colored fluid near the entrance to the tube. When the velocity was low the stream of colored fluid extended through the tube in a straight line without mixing with the water. As the velocity was increased it finally reached a value at which the colored stream would mix with the surrounding water and the tube would appear to be filled with a colored liquid.

The first of these types of flow is called streamline or viscous flow, and, as the motion of the fluid particles is definitely known, the problem of flow for this case can be treated mathematically. The second type is called turbulent or sinuous flow. In this type of flow, eddies are present, and, since the motion of the individual particles is not known, the problem of flow in this case must be studied experimentally. The principle of dimensional homogeneity offers the method of approach to the study of the turbulent-flow problem.

III—DIMENSIONAL HOMOGENEITY

The principle of dimensional homogeneity[6] states that in any complete physical equation all equal terms must have the same dimensions. It can be shown by this principle that, since only the diameter of the pipe, the velocity of flow, the density of the fluid, and the viscosity affect the loss of head per unit length, we can express the loss of head by

$$\Delta p = \rho \Delta h = \frac{f l v^2 \rho}{2gD} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

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[1] Engineering Division, Gulf Research Laboratory, and Instructor in Mechanical Engineering, University of Pittsburgh. Mr. Kemler received his B.S. in M.E. degree from the University of Kansas in 1927, M.S. degree from the University of Pittsburgh in 1931, and M.E. degree from the University of Kansas in 1932. From 1927 to 1929 he was with the Dodge Manufacturing Corporation. In 1929 he went to the University of Pittsburgh as instructor in Mechanical Engineering. Since 1930 he has been doing research work with the Gulf Research Laboratory, in addition to teaching at the University of Pittsburgh.

[2] Reynolds, "Scientific Papers," vol. II, p. 55.

[3] Couplet, "Memoirs of the Academy of Sciences," 1732.

[4] Stanton and Pannell, "Similarity of Motion in Relation to the Surface Friction of Fluids," Proc. Royal Soc., London, vol. A 214, pp. 199–224 (1914).

[5] Reynolds, "Scientific Papers," vol. II, p. 68.

[6] Buckingham, "Model Experiments and the Forms of Empirical Equations," Trans. A.S.M.E., 1915, pp. 263–296.

Contributed by the Hydraulics Division and presented at the Annual Meeting, New York, N. Y., Dec. 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

provided f, the friction factor, is a dimensionless quantity. Reynolds showed that for this to be true it must take the form

$$f = \phi\left(\frac{Dv\rho}{\mu}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

that is, f must be some function of $Dv\rho/\mu$. The form of this function must be determined experimentally for turbulent flow. Hereinafter the term $Dv\rho/\mu$ will be called Reynolds' number and will be designated by R.

List of Symbols

D = diameter in feet
v = velocity in feet per second
ρ = density in lb per cu ft
μ = absolute viscosity in ft-lb-sec units
ν = kinematic viscosity in ft-lb-sec units
l = length of pipe in feet
g = 32.2 = acceleration of gravity
f = friction factor
R = Reynolds' number
Δp = drop in pressure in lb per sq ft
Δh = loss of head in feet of liquid
W = pounds of fluid per second
V = volume
m = mean hydraulic radius.

IV—THE CRITICAL REYNOLDS NUMBER

The value of Reynolds' number at which the type of flow changes from streamline to turbulent flow or vice versa, is called the critical Reynolds number. This number is not well defined and may vary over a considerable range. In this unstable range of R the projected friction factor for streamline flow falls below the value for turbulent flow. Because of this fact, it is the minimum value of Reynolds' number at which turbulent flow can exist under normal conditions in which we are interested, since this value represents the maximum value of f that will be found in practice.

Schiller[7] concluded from his experimental work on the problem of turbulent flow that, for every Reynolds' number above 1160, a definite amount of disturbance is necessary to produce turbulence. The higher the Reynolds number, the less the disturbance necessary. Below the critical number R = 1160, viscous flow is stable against all disturbances. This value will be called the absolute critical Reynolds number for straight circular pipes. There no turbulent flow is possible; any eddies occurring will, after a sufficient distance, always disappear.

Hopf in the "Handbuch der Physik"[8] gives a review of the work done on determining the value of the critical Reynolds number and says that 2000 is a minimum value at which turbulence occurs under conditions similar to those met in practice. Streamline flow may continue at higher values, but turbulent flow cannot continue below this value except under the most favorable conditions. The various experimenters have found that under ordinary conditions the critical value varies from approximately 2000 to 3000. The region between these two values is an unstable one and the value of the friction factor depends upon how this region is approached. For instance, Ekman[9] was able to maintain streamline flow up to a Reynolds number of 25,500 by using smooth tubes and taking precautions to eliminate any disturbances at the entrance. At such values of R any slight disturbance would immediately change the type of flow; the phenomenon being somewhat similar to supersaturation or to undercooling.

It is interesting to note that in the case of curved or inclined pipes the critical Reynolds number is reduced below 1160, which has always been considered as the minimum value. Gibson and Grindley[10] obtained a critical number of 130 in experiments on the flow of air through a brass tube 0.32 cm in diameter coiled around a drum 36 cm in diameter. Jeffrey[11] also found that turbulent flow of water in an inclined rectangular section starts at a much smaller number than in a straight pipe. Carothers[12] obtained a critical Reynolds number of 200 for oil flowing in jointed pipes, and Davies and White[13] found that R = 1440 was the critical number for rectangular pipes. Schiller[7] quotes results of Ruches and Sorkau, who, respectively, found R = 400 and R = 430, as the critical number for glass capillary tubes. This shows that the critical number varies widely with conditions, and that any maximum, average, or minimum value that is set forth must be qualified. A more detailed discussion of the problem is given by Hopf in the "Handbuch der Physik"[8] and by Bateman[14] in the National Research Council's book on hydrodynamics.

Experimental data show that in straight pipes the transition from streamline to turbulent flow takes place about as shown in Fig. 1. Gibson[15] states that at low velocities the stabilizing effect

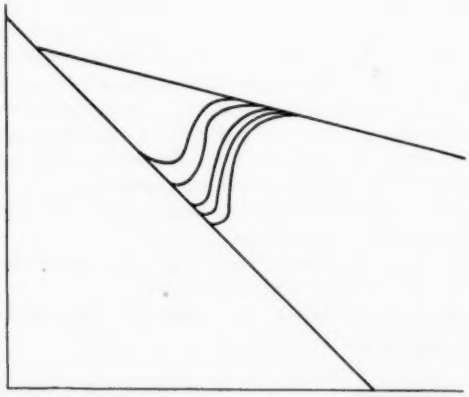


Fig. 1 Some Typical Curves in the Critical Region
(From Prandtl and Tietjens' "Hydro- und Aeromechanik," vol. 2.)

of viscosity prevents any tendency to deviate from streamline motion, such as may be produced by roughness which tends to set up eddies in the stream. It is probable that when the change is from the streamline to the turbulent region, the forces setting up eddies are not strong enough to send the entire system into turbulence, there being some damping of the eddies until some higher velocity is reached when the stabilizing effect of the viscosity has little to do with the flow.

However, if the change is from the turbulent to the streamline regions, the turbulence may persist until a value of 2000 or thereabouts is reached before the change to laminar flow takes place. In this case the damping effect of viscosity is not enough to stop any of the eddies. The suddenness with which this change takes place is probably dependent upon the roughness of the pipes, since

[7] Schiller, "Experimental Investigation of the Turbulence Problem," *Zeit. für Angew. Math. u. Mech.*, vol. 1 (1921), pp. 436–444.
[8] Hopf, "Handbuch der Physik," vol. VII.
[9] Prandtl and Tietjens, "Hydrodynamics and Aerodynamics."
[10] Gibson and Grindley, Proc. Roy. Soc., vol. A 83, p. 376 (1910).
[11] Jeffrey, "On the Flow of Water in an Inclined Rectangular Section," *Phil. Mag.*, vol. 49 (1925), sec. 6, pp. 793–807.
[12] Carothers, "Portland Experiments on the Flow of Oil in Tubes," Proc. Roy. Soc., vol. A 87, pp. 154–63 (1912).
[13] Davies and White, "A Review of Flow in Pipes," *Engineering*, July 19, 1929.
[14] Dryden, Murnahan, and Bateman, "Hydrodynamics," National Research Council, 1932.
[15] Gibson, "Hydraulics," D. Van Nostrand.

TABLE 1 SUMMARY OF ALL TESTS ON SMOOTH PIPES

(Water is the fluid used except as noted)

Experimenter	Range of pipe sizes (diam, in inches)	Source of data
BRASS		
Mills	0.54	"Flow of Water in Pipes," by Mills
Stanton & Pannell[1]	0.28–4.97	Phil. Trans. A, vol. 214, p. 199
Freeman	2–4	"Flow of Water in Pipes," by Mills
Saph & Schoder	0.107–2	Trans. A.S.C.E., vol. 11 (1903), p. 253, or "Flow of Water in Pipes," by Mills
Poche	1	*Power*, vol. 34 (1911), p. 134
Gibson[2]	....	"Modern Theory and Practice of Pumping," by Swindin, or *J. Ind. & Eng. Chem.*, vol. 14, no. 2, p. 112
Omkeck[3]	0.79–1.54	"Handbuch der Physik," vol. 7, p. 140
Jakob	0.393	Do.
Schiller	0.314–0.945	Do.
Jacob & Erk	2.75–3.9	Do.
Blasius	....	"Handbuch der Physik," vol. 7, p. 140
Eagle & Ferguson	1/8–1	Proc. Inst. M.E., vol. 2 (1930), p. 1001
GLASS		
Blasius (Bl)	0.2–0.4	"Handbuch der Physik"
Darcy (D)	1.954	"Flow of Water in Pipes," by Mills
Smith (S)	0.502–0.917	Do.
LEAD		
Bossut (Bo)	1	"Flow of Water in Pipes," by Mills
Couplet (C)	5 1/4	Do.
Darcy (D)	1/2–1 1/2	Do.
Blasius (Bl)	0.2–0.4	"Handbuch der Physik"
Iken (I)	1	"Flow of Water in Pipes," by Mills
Proves (Pr)	1 1/2	Do.
Kester (K)	....	"Modern Theory and Practice of Pumping," by Swindin
Reynolds (Re)	1/4	"Hydraulic Flow" reviewed by Barnes
TIN		
Bossut (Bo)	1.42–2.14	"Flow of Water in Pipes," by Mills
Dubreat (Du)	1.066	Do.

[1] Air and water. [2] Brine and water. [3] Air.

this could have a decided effect on the starting of eddies. It will also depend upon the viscous friction which is the cause of the damping effect. In order to get results that will agree with average conditions it is considered here that the streamline region exists to a value of $R = 2000$, at which value there is assumed to be a sudden change to turbulent flow.

The production of turbulence and instability of laminar flow has recently been discussed by Tollmien.[16] He shows that turbulence can only originate because of the instability of laminar flow, and that the critical Reynolds number depends largely upon the disturbances occurring.

V—STREAMLINE FLOW

As stated earlier, streamline flow can be treated mathematically since the flow satisfies the hydrodynamical equation of Navier-Stokes. In this case the pressure drop results from the shearing stresses necessary to move the particles relative to each other, or it may be considered as being caused by the viscosity of the fluid. The equation for pressure drop can be derived on this basis and is given by

$$\Delta p = \frac{64\,\mu l v}{2\,g D^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

Equation [1] will also apply to viscous flow provided f is a function of R. It has been found for streamline flow that $f = 64/R$. Substituting this value for f in Equation [1], the result is the same as that given by Equation [3]. The pressure drop for flow in straight pipes can then be calculated by Equation [3] provided R does not exceed 2000 and the pipe is circular in cross-section. Flow in non-circular sections will be discussed later.

[16] Tollmien, "The Present State of the Turbulence Problem." Paper before Pacific Coast Aero Meeting, June, 1932.

VI—ANALYSES OF TURBULENT-FLOW DATA

1 Method of Analyzing Data

In summarizing the data on turbulent flow for pipes, the original or published data were recalculated and analyzed on the basis of dimensional homogeneity. The data of each experimenter on a particular pipe size were recalculated and then plotted. An average curve for the particular pipe size was then selected. This average curve has been put on the summary drawing to serve as a permanent record and for the selection of a final average for a particular kind and size of pipe. Each curve on the summary drawings is marked with a letter and number. The letter refers to the experimenter (the letter used for each experimenter being given in the table of references), and the number to the inside diameter of the pipe in inches. Riveted steel, wood, and cement pipes are treated differently, as explained in the discussion of each.

In many cases data on the temperature of the water or gas were not given. A value of the temperature was then assumed in order to select a value of the viscosity. The assumption of the temperature or viscosity does not change the value of f since it is calculated from Equation [1]. The viscosity does, however, enter into R and defines the relative location of f. Therefore any error made in selecting the viscosity merely moves the friction curve parallel to the R-axis. Because of the general flatness of the friction curve in the turbulent region, a considerable change of μ does not change the position of the curve enough to alter the results to any appreciable extent.

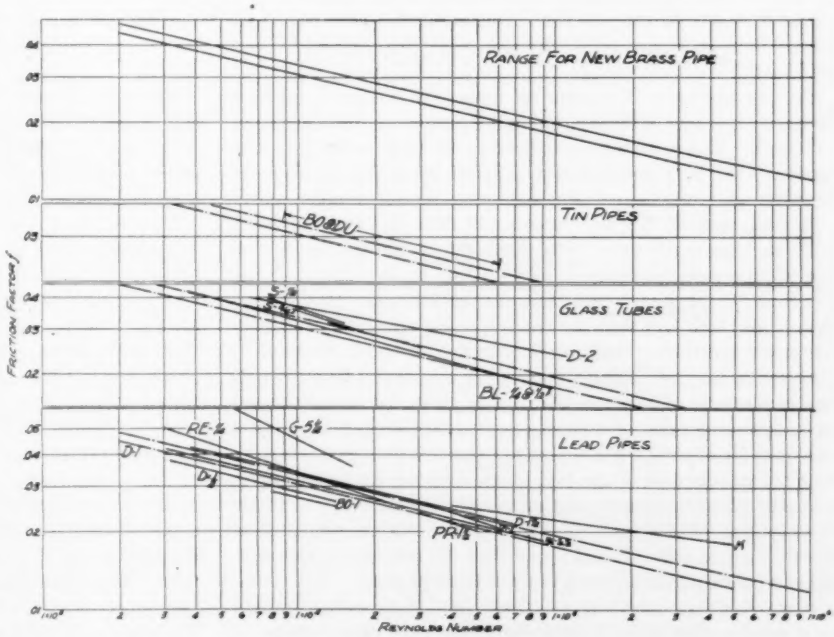


FIG. 2 FLOW IN SMOOTH PIPES

The fact that there is no slip at the boundary or surface of the pipe in either streamline or turbulent flow makes it seem reasonable that the friction factor at the absolute critical Reynolds number should be independent of the pipe size. On this basis at a Reynolds number of 1160 the friction factor should be independent of the pipe size regardless of its roughness provided the

diameter used is the one to make the friction curve coincide with Poisueille's law in the viscous region. This then gives one point for comparing the friction curve for various sizes of steel pipe as well as rough pipes, and its significance will be discussed in detail later.

The curves for flow in smooth pipes are taken as the basis upon which the averages for rough pipe were selected. Considering the large amount of data and the consistency of the friction factor, there is no reason why values of friction factor for rough pipe should ever fall below the values for smooth pipes except in the event of experimental errors. Therefore no data falling below the values for smooth pipe were given any consideration in selecting an average. In the analysis of the data, each type of pipe is considered separately and the various sizes of steel pipes are analyzed separately.

2 Smooth Pipes (Fig. 2 and Table 1)

The flow in smooth pipes offers a starting point for the study of the general turbulent-flow problem since they will have the lowest friction factor and will therefore determine the minimum value of friction factor obtainable. Smooth pipes are closely approximated by good commercial drawn brass pipe. Smooth lead, glass, and tin pipes also give results near those of brass pipe. Fortunately a considerable amount of experimental work has been done on this kind of pipe, so any conclusions that can be drawn will be well substantiated by experimental evidence. A few summaries have been made by other investigators of the work on smooth pipes, but none of these has taken all the data available for use in their analysis. In the following summary all the experimental data that have been published were reduced to the basis made available by Reynolds.

(*a*) *Brass Pipe.* The tests on brass pipe analyzed numbers between 1500 and 2000 and represent pipe sizes from 0.107 in. to 5 in. in diameter, using air, water, and brine as fluids. The summary of the sources of test data on brass pipe is given in Table 1.

In analyzing the immense amount of data on brass pipe each individual series of tests by a particular experimenter on one size of pipe were recalculated and plotted, using f and R as the coordinates. It was found that the experiments gave values that agreed well and that the friction factor was independent of the pipe diameter for the pipe sizes tested for equal values of R. The results for the different fluids were also found to be in perfect agreement. Therefore with brass pipes the friction factor for a particular value of R can be used regardless of the pipe size and fluid flowing. In the analysis of the test data it was impossible to reproduce all of the data in one drawing since so many of the tests gave identical results, and it became necessary to study the problem by determining the boundary within which all of the data would fall. It was found that the two curves shown in Fig. 2 would contain all but a few scattered points.

It was felt that even with relatively smooth pipes there would be a small variation in friction factor for each pipe because of slight variations in the pipe surface. By representing the friction factor as falling within a certain range, as was done in this case, the results would more nearly represent the actual conditions than by using a single average line. The lower curve represents a value of the friction factor about 10 per cent lower than the upper or maximum value. The range of friction factor for smooth pipes can be considered as varying by ± 5 per cent from an average curve. Considering the nature of turbulent flow and the variations in commercial pipe, this is rather consistent from an engineering standpoint.

There are no experimental data on flow at very high Reynolds' numbers for smooth pipe. In order to get some idea as to how the friction factor may behave at these high values of R, we must depend upon the present theories of turbulence. Schiller[7] states that even the technically smooth pipes become hydrodynamically rough at high Reynolds' numbers. The equation $\Delta = 8D/fR$ which is derived by Schiller[7] gives the approximate boundary-layer thickness (Δ = thickness of boundary layer in inches if D is taken in inches). When the boundary layer becomes the same thickness as the roughness projections, complete turbulent flow takes place. While this gives a plausible explanation as to why the friction factor should reach a minimum value, it does not explain why with complete turbulence we should find the friction factor varying from 0.54 to very small values depending upon the roughness. For the explanation of this we can revert to Prandtl's theory of turbulence. In his theory he introduces a distance, l, called the mixing length, which is a term analogous to the idea of mean free path used in the kinetic theory of gases. According to Prandtl, a particle travels a distance l perpendicular to the direction of mass flow before it loses any momentum. The momentum loss is then directly dependent on l. Von Kármán[17] has shown that this mixing length is directly proportional to the roughness projection. This would mean that with any given roughness projection the mixing length is fixed, hence the energy losses are fixed and the value of f, which depends upon the energy losses, is fixed and varies directly with the roughness projection.

The foregoing discussion is not to be taken as a rigorous proof since the explanations given are very much simplified, and the theory of turbulent flow has not been placed on a final mathematical basis. However, while based upon the present theories, the discussion is intended only to give a plausible explanation which will be helpful in interpreting some of the experimental data.

(*b*) *Glass Tubes.* For the remaining types of smooth pipes the average curve for the individual experimenter and pipe size is given on the graph. Each curve is labeled by a letter which refers to the experimenter and by a number which is the inside diameter of the pipe in inches.

The results of Blasius for glass tubes fall within the values for smooth pipes and indicate that the values for smooth pipe can be used for calculating pressure losses provided there is no appreciable variation in the diameter. The test values by Darcy are high, the reasons being that the tubes vary in diameter and that these variations in diameter cause an increase in energy losses because the fluid is speeded up and slowed down as it passes along the tube. The test results obtained by Smith seem to be high, and the reason may be because of variations in actual size from that used in the computations and in a failure to correct for entrance losses which are a considerable factor in short tubes with sharp entrances.

(*c*) *Lead Pipe.* If the lead pipe is drawn, the surfaces will be smooth and the values will correspond with those of smooth pipe. The results in general show clearly that the values for lead pipes correspond with those for the smooth or brass pipe. The test by Couplet was on a pipe made of rolled lead that had several bends. This resulted in high energy losses, and consequently the apparent friction factor is much higher. The tests by Kester were on pipes with couplings and would therefore be expected to show a higher apparent friction factor than a uniform pipe.

(*d*) *Tin Pipe.* The results for the tests on tin pipe fall on the upper line for smooth pipes. Since there are no data on the method by which the pipes were made or on their condition, the results are not entirely conclusive, but obviously since the method of manufacture and the surface character of the material are

[17] Von Kármán, "Mechanical Similitude and Turbulence," National Advisory Committee for Aeronautics, paper No. 611, March, 1931.

TABLE 2 SUMMARY OF TESTS ON NEW STEEL PIPE

(Water is fluid used unless otherwise noted)

Experimenter	Range of sizes (diam, in.)	Source of data	Remarks
Freeman (F)	0.36–8	"Flow of Water in Pipes," by Mills	Pipes had screw-joint couplings
Darcy (D)	0.48–1.55	Do.	Diameters varied 18 per cent on small sizes
Smith (S)	0.627	Do., or "Hydraulics," by Smith	Joints made by butting ends of pipe
Omkeck (O)	0.78	"Handbuch der Physik," vol. 7, p. 150, or "Forschb. der V.D.I.," vols. 158 & 159, 1914	No data available
Francis (Fr)	0.801–2	"Flow of Water in Pipes," by Mills	Apparently the same as Smith's tests
Rowland (R)	1	"Flow of Water," by Schmeer	No joints
Poche (P)	1	*Power*, vol. 34 (1911), p. 34	Couplings used
Fritzsche (Fri)	1–1.54	"Handbuch der Physik," vol. 7, p. 150	No data available
Gould & Levy (G & L)	1.38	Univ. of Ill. Eng. Expt. Bul. No. 182	No couplings. Brine used
Brix (B)	3¼–6½	Phil. Trans., vol. 214	No data given on pipe. Air used
Corp & Ruble (C & R)	½–12	Univ. of Wis., Bul. No. 1, vol. IX, p. 39	Flanged joints
Brabbée (Br)	½–6	Gresecke in *Jl. A.S.H. & V.E.*, vol. 21, p. 478	Temp. varied widely, but values not given
Kite & Kennedy (K & K)	1, 2, 4	*Jl. Ind. & Eng. Chem.*, vol. 14, no. 2, p. 112	Data on individual sizes not given. Oil used
Bayles (Ba)	8.94	Scobey, "Flow of Water in Riveted Steel & Analogous Pipes"	Standard coupling-type joints
Glazebrook, Higgins & Pannell (GH & P)	3 & 4	*Jl. Inst. Pet. Tech.*, vol. 2 (1914–1915), p. 54	No couplings, 5 ft. gage length and accurate manometer used. Oil used
Lander (L)	0.422–1.3	Proc. Roy. Soc., vol. 92-A (1916), p. 337	Joints made by butting ends of pipe. Steam and water used
Greeve (Gr)	1½	Purdue Univ. Bul. No. 2, "Flow of Water through 1½-In. Pipe"	One coupling in 40-ft length
Corp & Hartwell (C & H)	1–4	Univ. of Wis. Bul. No. 66, p. 83	Probably no joints
Davis (Da)	2	Univ. of Wis. Bul. No. 403	Flanged joints used
Standard Oil Co. of Calif. (SO)	....	"Flow Book"	Data from oil-pipe-line tests. Oil used
Heltzel (H)	8–12	*Oil & Gas Jl.*	Data from pipe-line tests. Oil used
Feely & Riggle (F & R)	0.27	Univ. of Pittsburgh unpublished thesis	No couplings. Air and water used
Berwald & Johnson (B & J)	1–2	U. S. Bureau of Mines R.I. 3153	Air and natural gas used
Beale & Docksey (B & D)	0.3	*Jl. Inst. Pet. Tech.*, Aug., 1932	.....
Anglo Persian Oil Co. (APOC)	2–12	*Jl. Inst. Pet. Tech.*, Aug., 1932	Natural gas used
Fair, Whipple & Hsiao (F, W & H)	1½	*Jl. N.E. Water Wks. Assn.*, vol. 44 (1930), pp. 499–513	.....

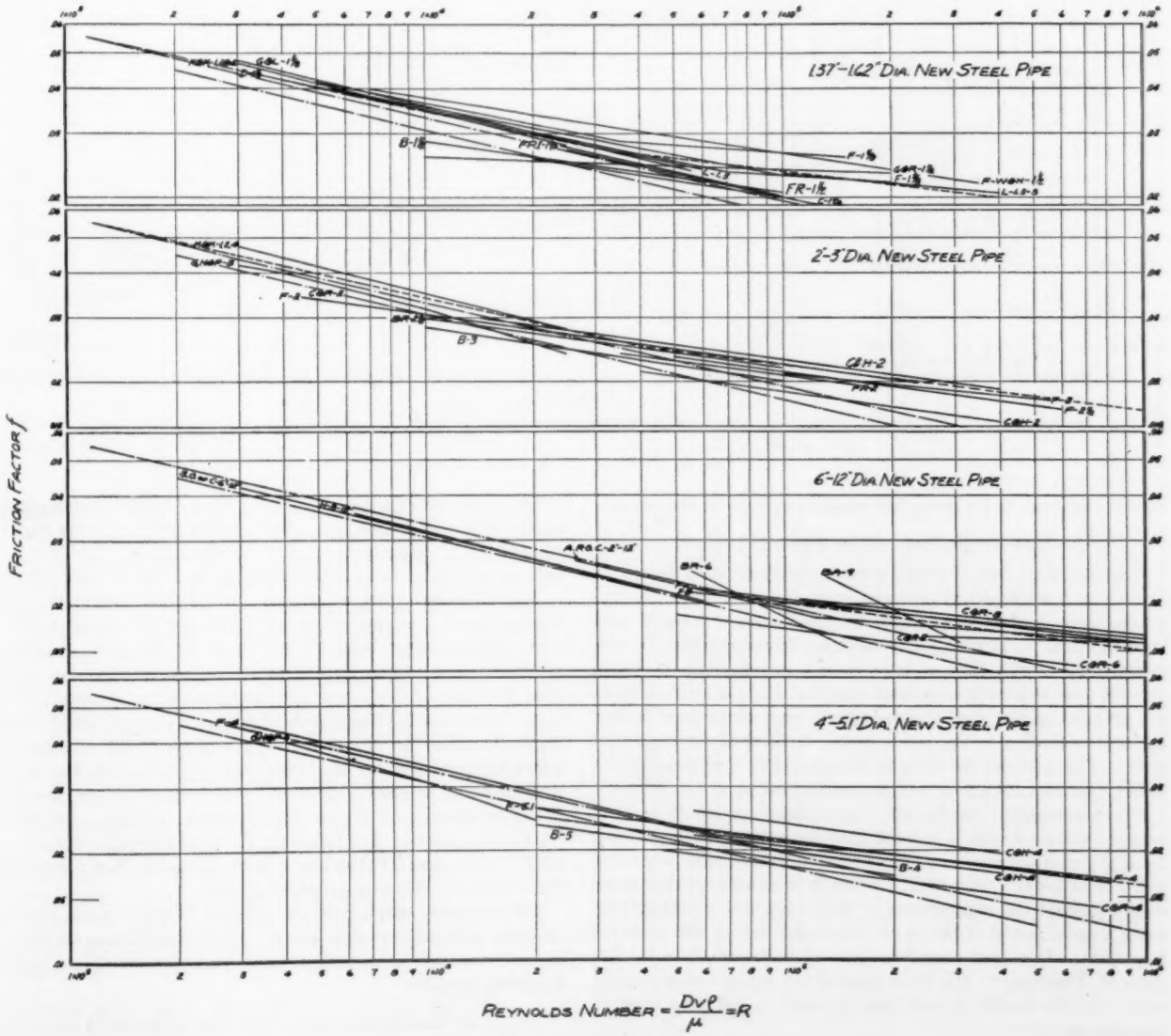


FIG. 3 FLOW IN NEW STEEL PIPES

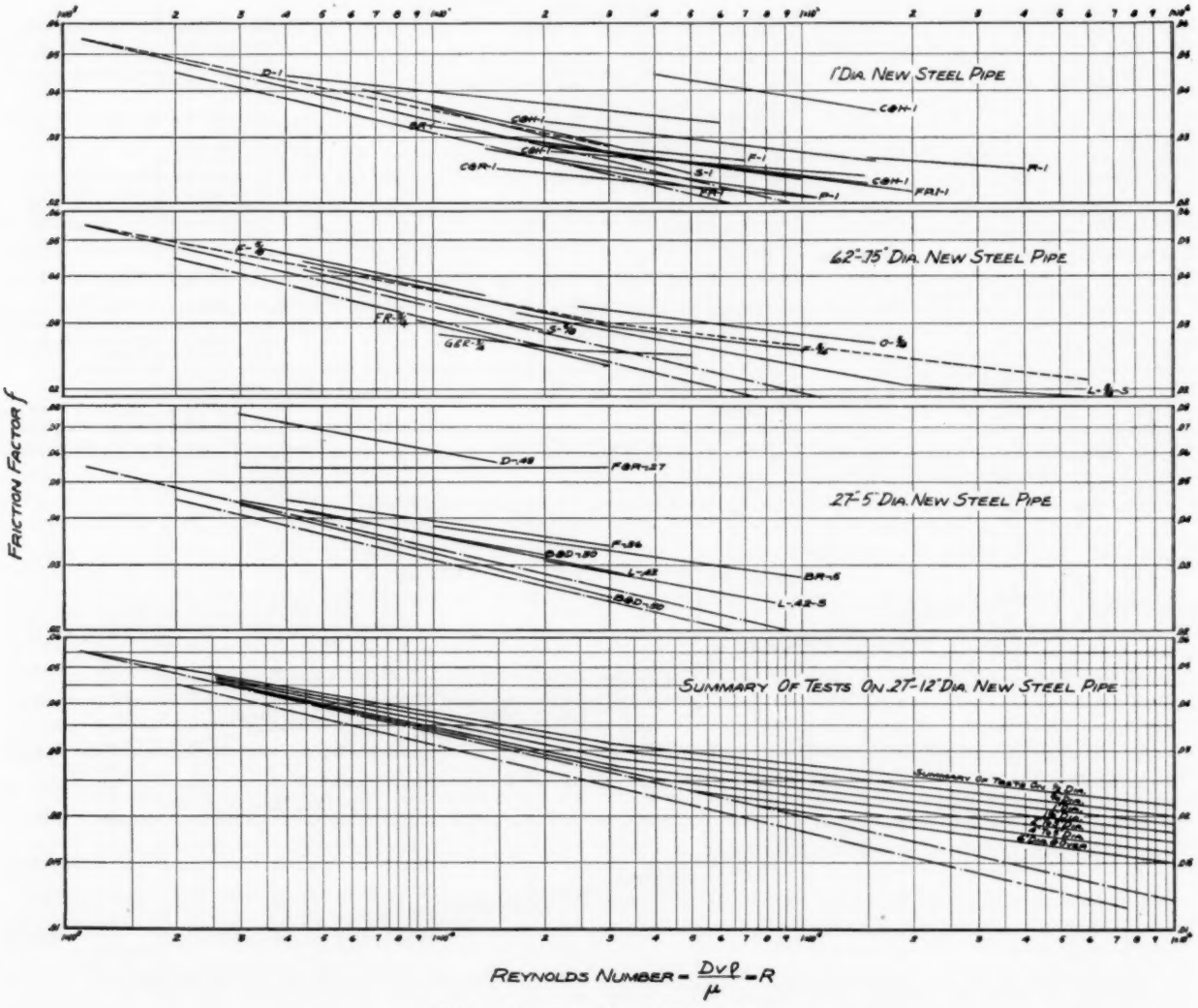


Fig. 4 Flow in New Steel Pipes

almost the same as for brass, the results ought to be the same.

3 New Steel Pipes (Figs. 3 and 4, Table 2)

The turbulent flow of fluids in steel pipes has never been subjected to a scientifically planned study because most investigators were unable to correlate the data for the various pipe sizes or their data with those of other investigators. In this analysis all of the data of independent investigators of which records were available have been compiled and studied. Curves are grouped on the basis of actual inside pipe diameter in inches, and not according to the conventional method of denoting pipe sizes. This permits the data to be applied to heavy and extra heavy pipe and tubing as well as to standard pipe.

The summarized results are surprisingly consistent in that extreme values do not in general vary more than ± 10 per cent from the average curve at higher values of Reynolds' numbers, and less at lower values of R. When it is considered that these data represent the experiments of 26 independent investigations made over a period of 80 years, the results are all the more remarkable. The most complete series of tests are those by Dr. John R. Freeman.[18] His tests covered the entire range of pipe sizes, and the results in each case are seen to fall very close to the average curve.

[18] Mills, "Flow of Water in Pipes."

The oldest set of tests are those by Darcy, which were made in 1852. The $^1/_2$-in. and 1-in. sizes show a friction factor higher than those in other tests on corresponding sizes. The description of the pipes states that the diameter varied as much as 18 per cent for these sizes, which accounts for their apparently high friction factor. His results on the $1^1/_2$-in. size are consistent with other experimental results. The tests by Lander on steam (L-.42-S) and on water (L-.42) form a continuous curve and show that the flow of dry steam practically follows the dimensional law. The tests by Feely and Riggle on $^1/_8$-in. standard black pipe also show the curve for water and air to be continuous. These tests were made on a pipe ordered from stock and show that this size of pipe is relatively very rough. A test using a heavy lubricating oil showed that the measured pipe size made the friction curve for low Reynolds' numbers follow the curve $f = 64/R$, which gives a better check on the diameter than measuring the volume by filling the pipe with water.

The summary curve gives the selected average values for the different sizes of new steel pipes. *The extreme values do not in general vary more than ±10 per cent from the average at higher Reynolds' numbers.*

4 Galvanized Iron Pipe (Fig. 5, Table 3)

Galvanized iron pipe, because of the method of manufacture

TABLE 3 SUMMARY OF TESTS ON ROUGH PIPES

(Water is fluid used unless otherwise noted)

Experimenter	Range of sizes (diam, in.)	Source of data	Remarks
GALVANIZED IRON PIPE			
Saph & Schoder (S & S)	0.350–1	Trans. A.S.C.E., vol. 11 (1903), or "Flow of Water in Pipes," by Mills	No couplings
Iken (I)	1	"Flow of Water in Pipes," by Mills	Pressures read with metallic gage
Poche (P)	1	*Power*, vol. 34 (1911), p. 34	.....
Gibson (Gi)	2	"Modern Theory and Practice of Pumping," by Swindin	Pipe had joints
Petit (Pe)	10–13	"Handbuch der Physik," vol. 7, p. 150	Newly galvanized sheet iron
Gibson (Gi)	2 & 4	"Hydraulics and Its Applications," by Gibson	No date on pipe
Emswiler (E)	6 & 12	Univ. of Mich. Expt. Station Bul.	Probably no joints . Used air
Feely & Riggle (F & R)	0.62	Univ. of Pittsburgh unpublished thesis	No joints. Air and water used
Beale & Docksey (B & D)	0.30	*Jl. Inst. Pet. Tech.*, Aug., 1932	.....
USED WROUGHT-IRON PIPE			
Freeman (F)	2–3–4.042	"Flow of Water in Pipes," by Mills	Scale and pits on 2- & 3-in. pipe; 1/64-1/32-in. quite rough; 4.042-in. pipe very rough. Pipes very rusty
Corp & Hartwell (C & H)	6–8	"Loss of Head in Pipe Bends," Univ. of Wis. Bul. No. 6	Pipe was clean
Corp & Ruble (C & R)	6	"Tests on Wrought Iron Pipe," Univ. of Wis. Bul. No. 1, vol. IX	This pipe unused for six months
Mills (M)	3	"Flow of Water in Pipes," by Mills	Scale pits about 1/32 in. Pipe was rusty
Schoder (Sc)	6	Do.	Pipe had been used in a heating main for several years
USED TAR-COATED WROUGHT-IRON PIPE			
Frederick & Wilson (F & W)	3.628 5.72 8.00	"Flow of Water in Riveted Steel Pipe," by Scobey	.....
Scobey (Sc)	7.69	"Flow of Water in Riveted Steel Pipe," by Scobey	.....
NEW CAST-IRON PIPE			
Darcy (D)	3.225 5.395 7.402	"Flow of Water in Pipes," by Mills	.....
OLD TAR-COATED CAST-IRON PIPE			
Bonn Waterworks(Bonn)	12.3	"Hydraulics," by Unwin	.....
Ehmann (Eh)	2, 8, 10	"Flow of Water in Pipes," by Mills	There seemed to be no incrustations in the pipe; 2-in. pipe 2 years in use, 8-in. 4 years, and 10-in. 1, 4, and 6 years
Iben (I)	6, 12	Do.	6-in. pipe 2 months in use, 12-in. 2 and 14 years
Mills (M)	24	Do.	Pipe had been cleaned before test; unused for 2 years
Safford (Sa)	12	Do.	Pipe 12 and 37 years in use
USED CAST-IRON PIPE			
Darcy (D)	1.41–11.69	"Flow of Water in Pipes," by Mills	Pipes had a thin calcareous deposit
Iben (I)	4–20	Do.	Pipes in use from 13 to 25 years
Safford (Sa)	12	Do.	Pipe heavily tuberculated
Teatson (T)	20	Do.	Pipe in use 4 years
NEW TAR-COATED CAST-IRON PIPE			
Freeman (F)	3.894 7.978	"Flow of Water in Pipes," by Mills	3.894-in. much rougher than 8-in.
Iben (I)	4, 12	Do.	.....
Mills (M)	11.94	Do.	Special care taken to get pipe of uniform diameter
Safford (Sa)	24	Do.	Test made on a length of 5195 ft
SPIRAL RIVETED PIPE			
Greeve & Martin (G & M)	4, 6, 8, 10	"Flow of Water in Spiral Riveted Pipe," Purdue Univ. Bul. No. 8	One screw coupling in a 40-ft gage length
Schoder & Gehring (S & G)	4, 6	"Flow of Water in Riveted Steel and Analogous Pipes," by Scobey	.....

and the variation in roughness, is to be expected to give varied results. The general range is considerably above that for new wrought-iron pipe. The average of the values for 1-in. pipe on which data are available gives values about 50 per cent higher than for wrought-iron pipe. For smaller sizes there are not enough data available from which to make any worthwhile estimate of the increase of friction factor, but it is probable that the figure given would be too small for sizes under 1/2 in. in diameter.

The tests by Feely and Riggle[19] on 1/8-in. standard galvanized pipe bought from a supply house are shown in Fig. 5. The curve as run showed a friction factor over 6 times that of a rough pipe when the measured pipe size was used in making the calculations. In order to check the pipe

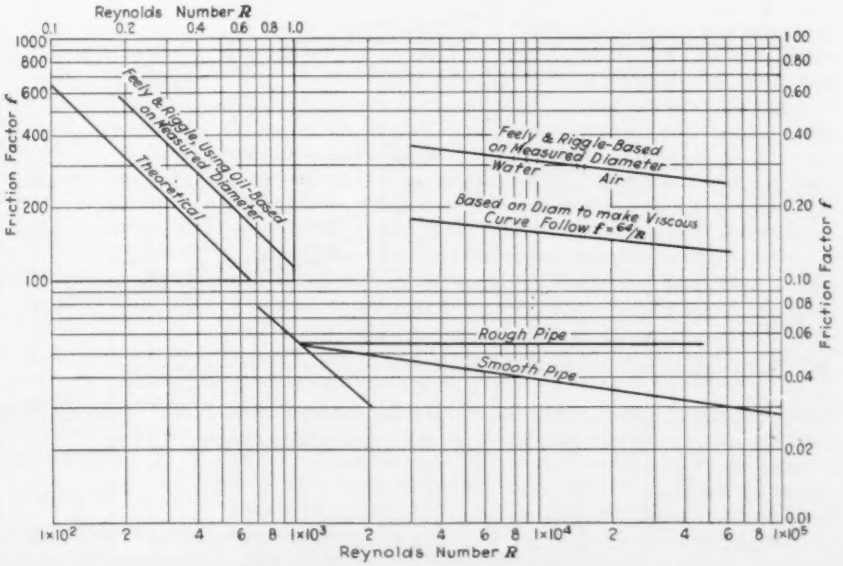


FIG. 5 TESTS BY FEELY AND RIGGLE

[19] Feely and Riggle, "Flow of Fluids in Small Pipes," unpublished thesis, University of Pittsburgh.

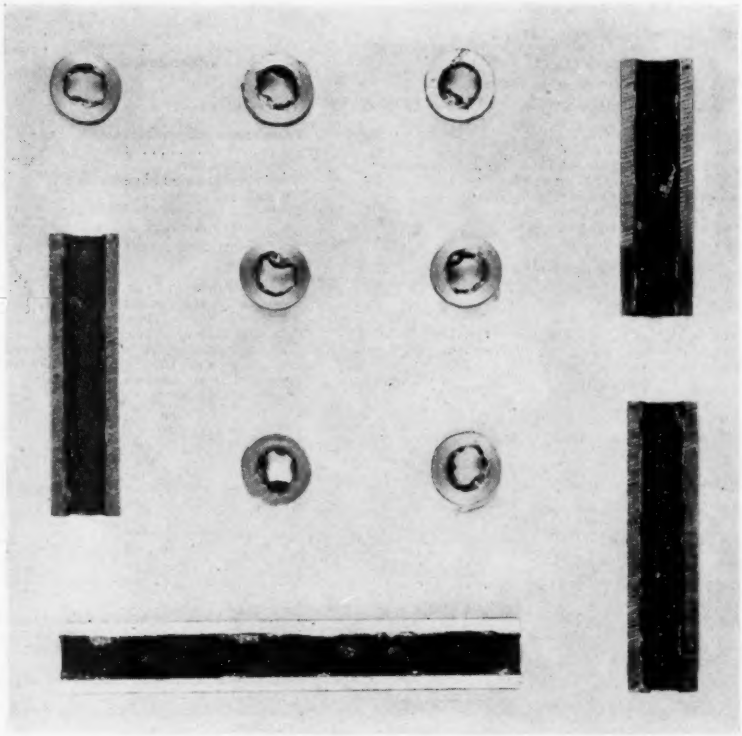


FIG. 6 LONGITUDINAL AND TRANSVERSE SECTIONS OF GALVANIZED PIPE

size a series of tests using a lubricating oil were made at Reynolds' numbers between 0.2 and 1, giving a friction factor between 600 and 100, which does not check with the theoretical curve. A correction of the diameter to make the results coincide with the theoretical curve brings about a reduction of the friction factor to one-half of its value when based on the measured value of the diameter.

This correction still does not show why the friction factor should be so much higher than that for a rough pipe as defined later. In order to find some plausible explanation the author examined the pipe by cutting a large number of sections from it. Some typical sections were machined and are shown in Fig. 6. The projections in the pipe were metallic and resulted from galvanizing. The sections were milled from the pipe, and as a result some of the smaller deposits on the edges were knocked off in machining. The resulting section is therefore not as irregular as the actual pipe, but shows the order of magnitude of the roughness. One of the longitudinal specimens will be seen to show half of the section almost filled with deposits. A pipe with a roughness relatively as large as this has many other factors introduced which affect the flow resistance besides the wall. In a set-up with a change of section it is recognized that there may be quite high losses due to sudden contractions or enlargements, and in this pipe the same type of

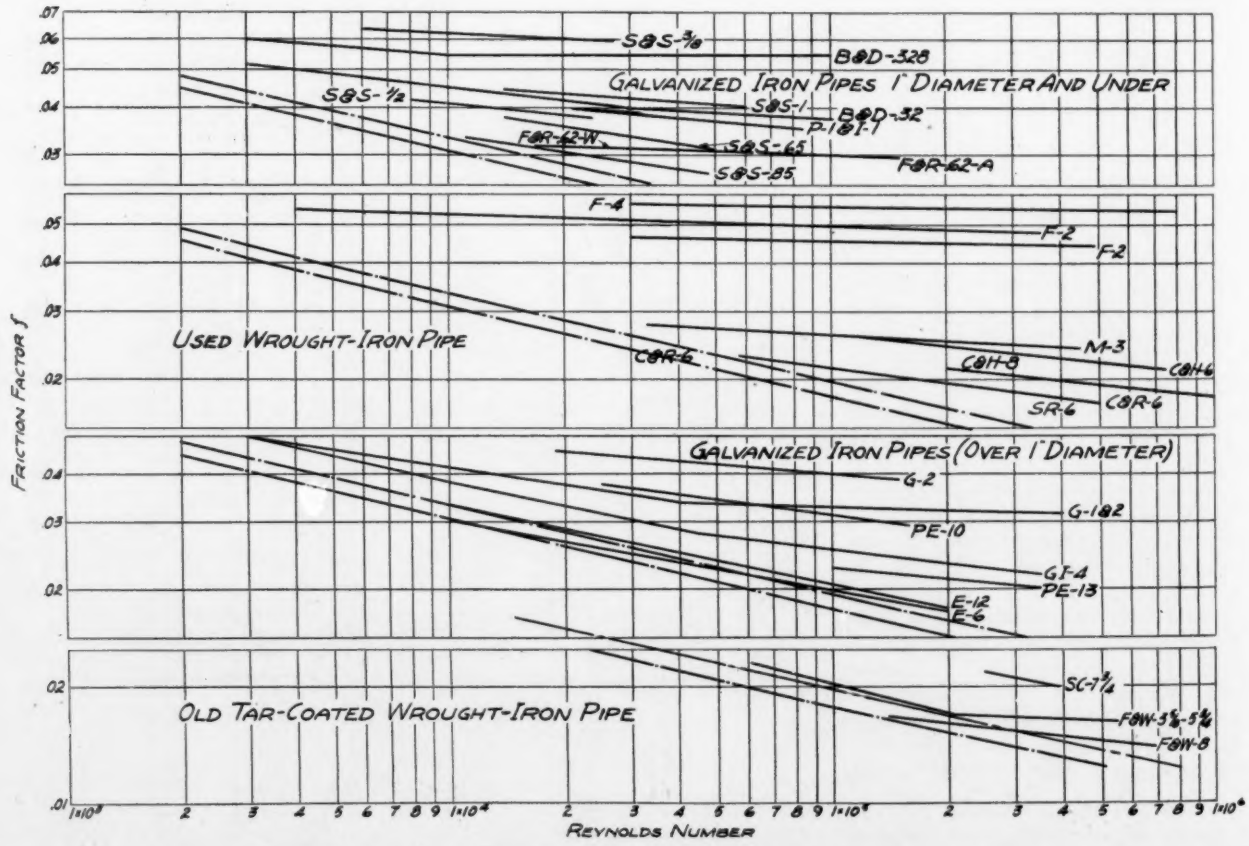


FIG. 7 FLOW IN GALVANIZED, USED, AND OLD TAR-COATED WROUGHT-IRON PIPES

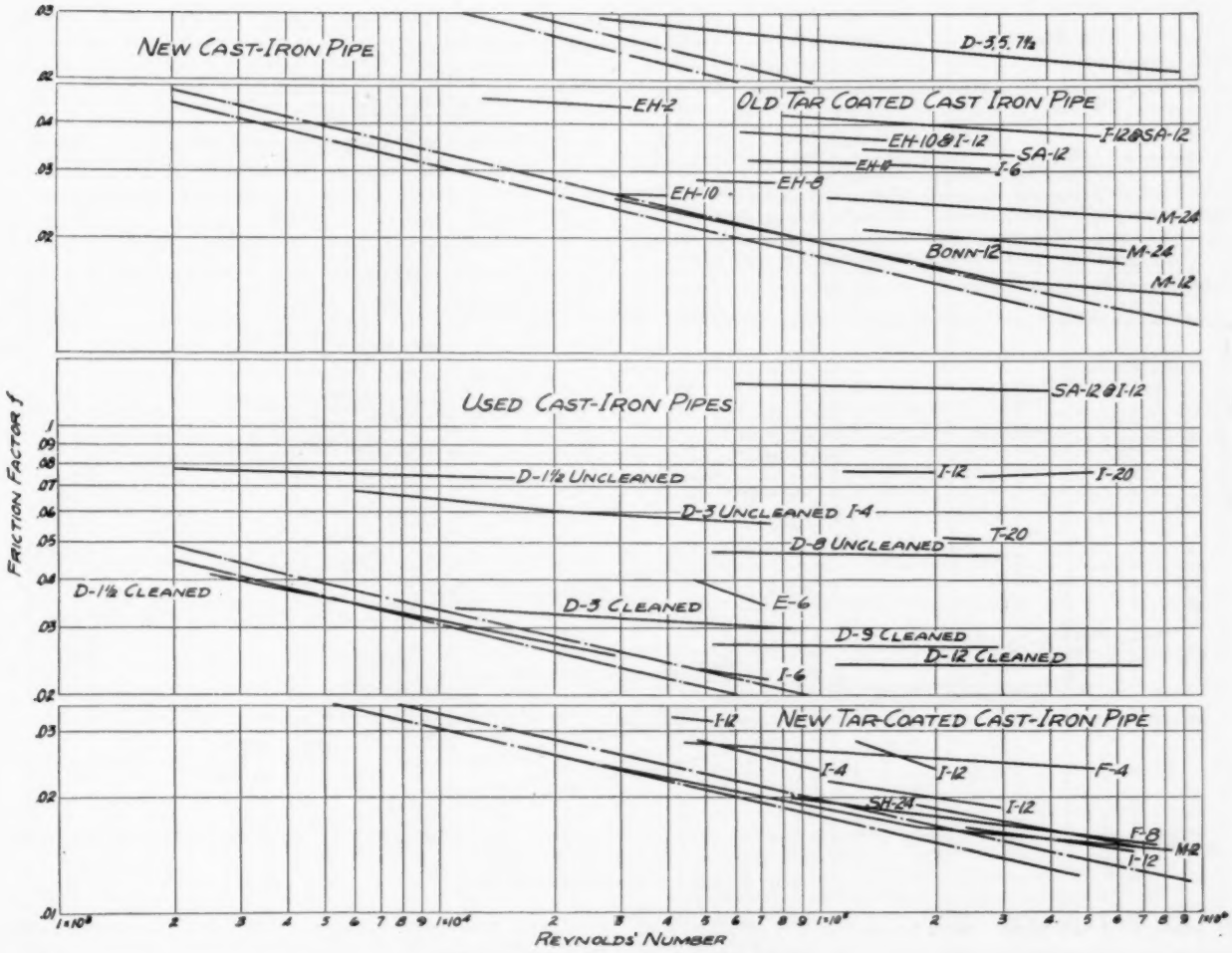


FIG. 8 FLOW IN CAST-IRON PIPES

losses would be present. There would also be additional losses caused by the fact that the roughness projections occur at different positions around the pipe, giving an effect such as occurs at elbows, sharp curves, valves, etc.

This pipe illustrates the condition that can exist in small galvanized pipes and also in small used corroded pipes. For larger sizes the condition will not be so exaggerated.

For the sizes over 1 in. in diameter, the data are meager and inconsistent. The tests by Gibson show the values for 2-in. pipe to be about 50 per cent higher than the average for wrought iron. The tests by Emswiler indicated that for large diameters the effect of galvanizing decreases quite markedly, as would be expected.

About all that can be concluded from the tests on galvanized iron pipe is that the friction factor should be assumed to be at least 50 per cent greater than for ordinary new steel pipe in sizes up to 2 or 3 in. in diameter. In larger sizes the effect is apparently considerably reduced, and for very small sizes is greatly increased. Because of variations from pipe to pipe a large number of tests would have to be made before any reliable values could be decided upon for the range of friction factor for galvanized pipe.

5 USED WROUGHT-IRON PIPE (Fig. 7, Table 3)

As would be expected, the friction factor for used pipe varies a great deal from pipe to pipe. Several very interesting general conclusions can be drawn from the data collected on this kind of pipe, even though these are insufficient to give quantitative results.

The results of the tests by Freeman are very useful in setting up a basis for studying other kinds of very rough pipes. The pipes tested by Freeman are very accurately described[18] and are said to be very rough, but he states that there was no change of diameter and no accumulation of material on the walls of the pipe. He describes these pipes as having some rust scale and pits from $^1/_{64}$ in. to $^1/_{32}$ in. deep. The first thing that will be noticed about his results is that the friction factor is practically independent of the velocity, viscosity, and density. That is, the pressure drop varies as the velocity squared and not as a power between 1 and 2. The second is that if these lines are projected to low values of Reynolds' numbers they will average a point about where the curves for all pipes are started.

From these facts it would be concluded, first, that as the roughness of the pipe increases, the curve for f would approach the horizontal line; second, that if the correct diameter is used, the curve should always pass through the critical point when this pipe becomes very rough.

On this basis we could set up a hypothetical completely rough pipe in which the friction factor was constant in the turbulent region, and the value of the friction factor would be the same as for all pipes at the critical velocity. Here we just limit the roughness to such an order of magnitude as not to introduce contraction effects in the pipe.

The results on the larger pipes show only an average increase of 20 per cent over that for new pipe. The curve by Schoder on a pipe used in a heating main shows values slightly less than for

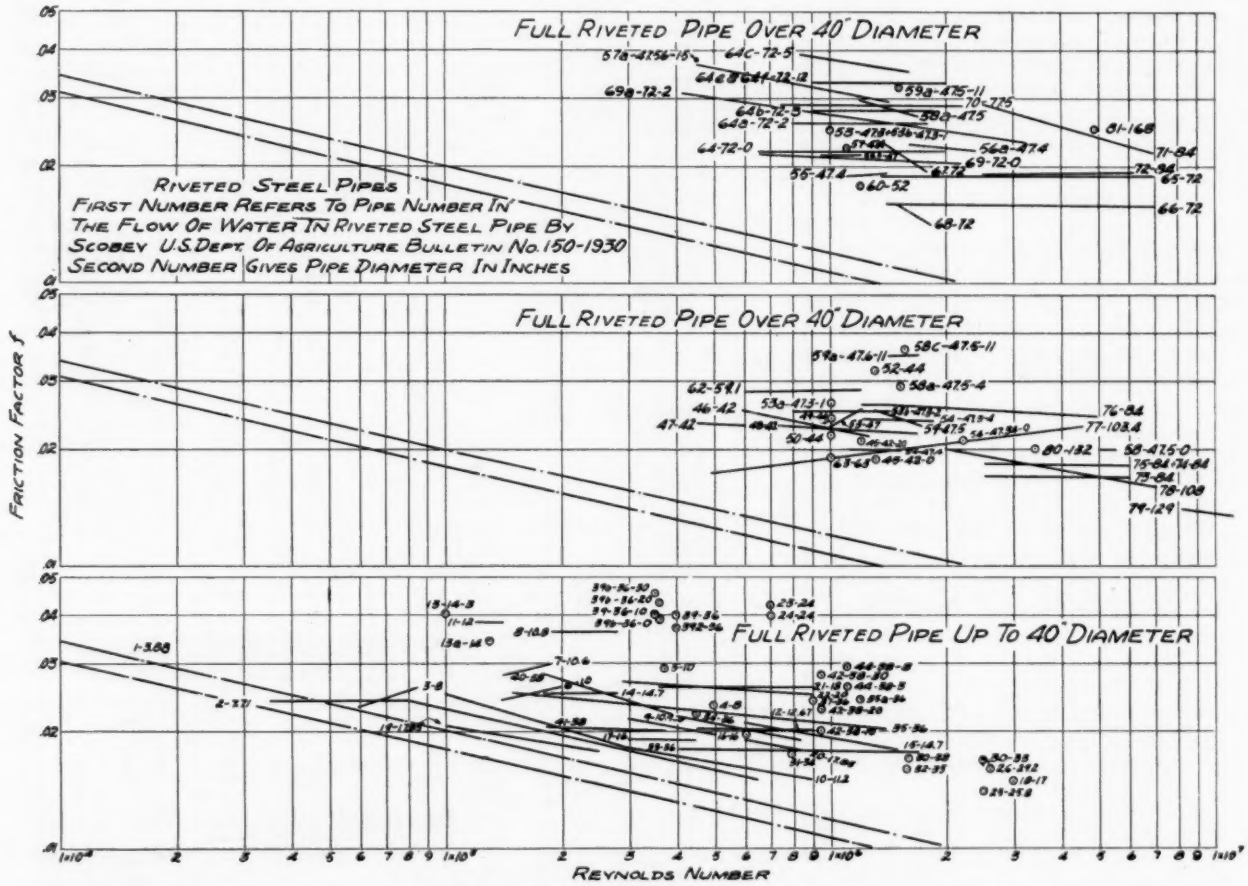


FIG. 9 FLOW IN RIVETED STEEL PIPE

the average for new pipes. Such results could be obtained with pipes with scale present when they were new, and with use, if the scale were removed, f would be reduced. From the tests on large pipes it might be concluded that the effect of use would be inconsiderable unless there was excessive corrosion or accumulation of material in the pipe.

Any accumulation of dirt, sand, organic matter such as paraffin, or other substances in a pipe, will reduce the effective diameter and give an apparent friction factor very much higher than the value on the basis of the hypothetical completely rough pipe discussed above. Also with the accumulation of material, there will probably be variations in diameter which will cause additional energy losses because the fluid will be alternately accelerated and decelerated. This will have little effect except in very small pipes.

6 OLD TAR-COATED WROUGHT-IRON PIPES (Fig. 7, Table 3)

There are very few data on which to base any very general conclusions, but those available indicate values of f much lower than for old uncoated wrought-iron pipes. This is probably due to the fact that the coating checks corrosion or deposition of calcareous material.

7 NEW CAST-IRON PIPE (Fig. 8, Table 3)

The results of these tests differ greatly from what would ordinarily be expected. However, Darcy's notes state that the pipe had a remarkable degree of polish. The values correspond closely for those of new $^3/_4$-in. steel pipe. The data are very meager, but indicate that with well-polished cast-iron pipe the friction losses would not be over 25 to 50 per cent greater than for new steel pipe of the same diameter. The rougher pipes would show values corresponding more nearly with those for the perfectly rough pipe. There seemed to be many values recommended for new cast-iron pipe, but apparently Darcy's experiments were the only ones on which the experimental data could be found.

8 USED CAST-IRON PIPE (Fig. 8, Table 3)

This set of experiments shows the effects of thin deposits on the inside of pipes. The tests by Darcy on cleaned and uncleaned pipes show the large change of friction factor with changes in surface and diameter. The curves for these very rough pipes are quite flat, indicating that the variation with viscosity and velocity is small. The tests by Safford and Iken are interesting because they are on very old pipes and show that there was some change in effective diameter. It is seen that the cleaned pipes give a friction factor such that if the curve were projected to low values of Reynolds' numbers, the point would not lie above the critical point. These data corroborate the theory already given.

9 NEW TAR-COATED CAST-IRON PIPES (Fig. 8, Table 3)

These tests show values that are about 25 per cent higher than those for wrought-iron pipes of the same size (4 in.). For larger sizes, the values seem to average somewhat lower than the values for 6-in. steel pipe.

10 OLD TAR-COATED CAST-IRON PIPES (Fig. 8, Table 3)

These tests show a wide variation in friction factor and a much higher value therefore than for new tar-coated pipes. The effect of tar coating is probably to reduce corrosion and the accumulation of calcareous materials. This is shown by the fact

that the curves in general do not go much above a horizontal line through the critical point. Several of these are curves for pipes in which there was a slight incrustation.

11 Riveted Steel Pipe (Fig. 9)

The data used in Fig. 9 were taken from U. S. Bureau of Agriculture Bulletin No. 150 (1930), by Fred Scobey. The curves show the results of field tests on pipe lines used for irrigation and power purposes. No discussion of the various pipes will be attempted, and those desiring information concerning them are referred to the original source. Mr. Scobey states that sheet-metal pipes with flat rivets as used for irrigation or low-pressure installations have a capacity appreciably above that of plate-type pipes used in high-pressure installations. The greater capacity means

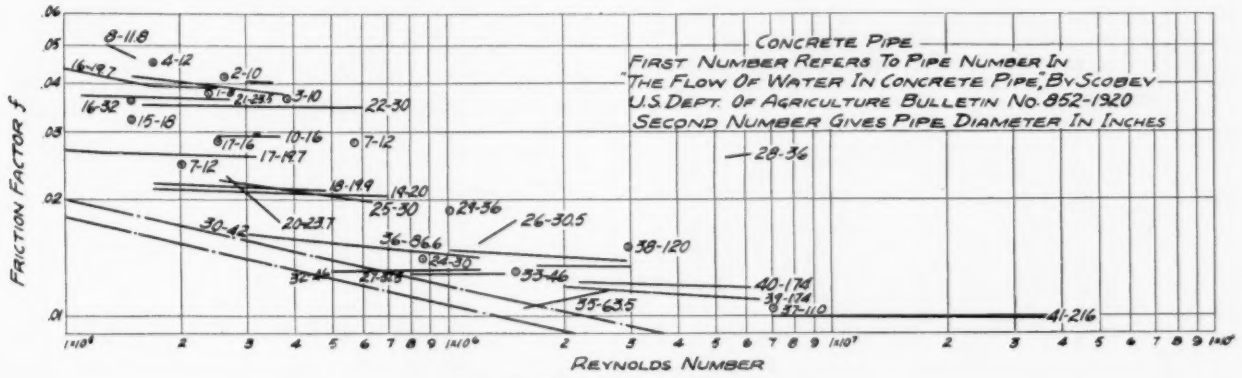


Fig. 10 Flow in Concrete Pipe

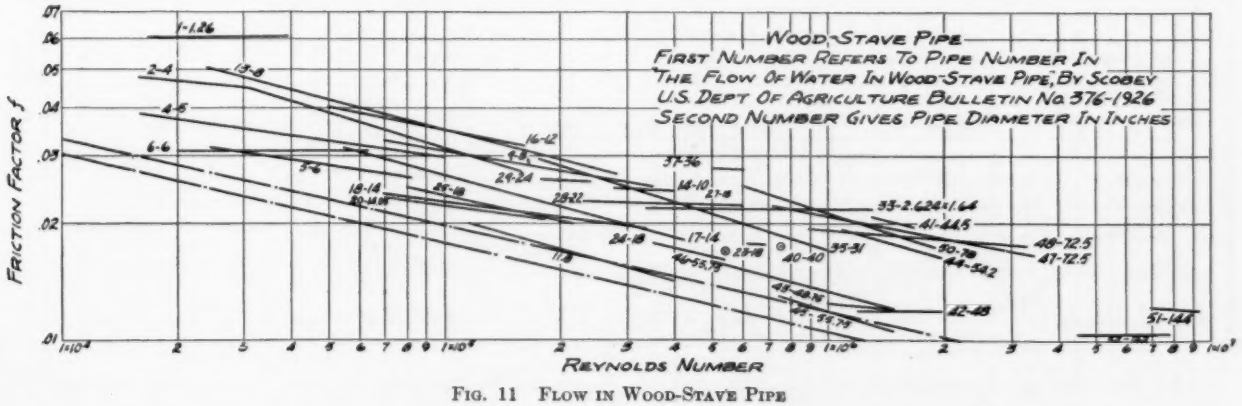


Fig. 11 Flow in Wood-Stave Pipe

that these pipes have a lower friction factor. This conclusion agrees with that arrived at by Mr. Pigott[20] from a consideration of roughness of pipes, in which he pointed out that the roughness of a light sheet-steel duct or pipe for low pressures such as flues or low-pressure irrigation work would be much less than for heavy pipe used for penstocks and high-pressure installations.

Another important point is that the friction factor for this type of pipe does not exceed 0.045, even for pipes in use as long as 30 years or more. Considering the relative roughness of this type of pipe, it is seen to support the previous theory that unless there is a change in diameter the friction factor will not exceed the value at the critical velocity.

12 Concrete Pipes (Fig. 10)

The data for these curves have been taken from U. S. Department of Agriculture Bulletin No. 852 (1920). They are of interest because they represent a type of pipe usually considered rough and because they include values at particularly high Reynolds' numbers. Again the data on small sizes show values below the value at the critical number. The lines are predominantly horizontal. It will be noticed that with one exception the values at high Reynolds' numbers are relatively low, this one exception being on a relatively small pipe. Several of the curves have slopes that are questionable, but this is probably because the range of velocities is not sufficient to permit an accurate curve to be drawn. Here again the data were plotted and a curve that was considered to be nearer an average was selected. In general the experimental points showed quite a spread, and in all cases an attempt was made to select a mean curve.

13 Wood-Stave Pipe (Fig. 11)

These data were taken from U. S. Department of Agriculture Bulletin No. 376 (1927). The curves again fall within the limit set for rough pipes except for the 1.26-in. size, which is not far out. Hopf[21] has concluded from his summary that the curves for wood-stave pipe would be parallel to the brass line. The data given here, however, show almost an equal tendency for the lines to be horizontal at the higher Reynolds' numbers.

14 Artificially Roughened Pipe

A great deal of work has been done by German investigators on artificially roughened pipe. Fromm,[22] Fritsch,[23] Schiller,[24]

[20] Pigott, "The Flow of Fluids in Closed Conduits." Paper before National Process Meeting, Buffalo, June, 1932.

[21] Hopf, "Handbuch der Physik," vol. 7, p. 149.

[22] Fromm, "Flow in Rough Pipes," *Zeit. für Angew. Math. u. Mech.*, vol. 3 (1923), pp. 339–358.

[23] Fritsch, "Turbulent Flow in Pipes," *Zeit. für Angew. Math. u. Mech.*, vol. 8 (1928), pp. 199–216.

[24] Schiller, "Flow in Rough Pipes," *Zeit. für Angew. Math. u. Mech.*, vol. 3 (1923), pp. 2–13.

and Nikuradse[25] have made rather extended studies of the problem, and a very brief outline of their researches will be given here.

Fromm's results are shown in Figs. 12 and 13. He experimented with four degrees of roughness. In his work he used a rectangular section having smooth sides and a rough top and bottom. This set-up permitted an investigation of the effect of depth of section on behavior under a given roughness. The first set of tests were made using a wire net surface having a thickness of 0.115 mm. The curves seem to behave rather peculiarly at low Reynolds' numbers, but at values much above the critical become nearly parallel and horizontal, and a variation with the depth is found for the shallow sections. Even the shallow section shows a value of f not much above that for the ideal rough pipe. A correction of diameter for the surface thickness would bring this value to the line for rough pipe.

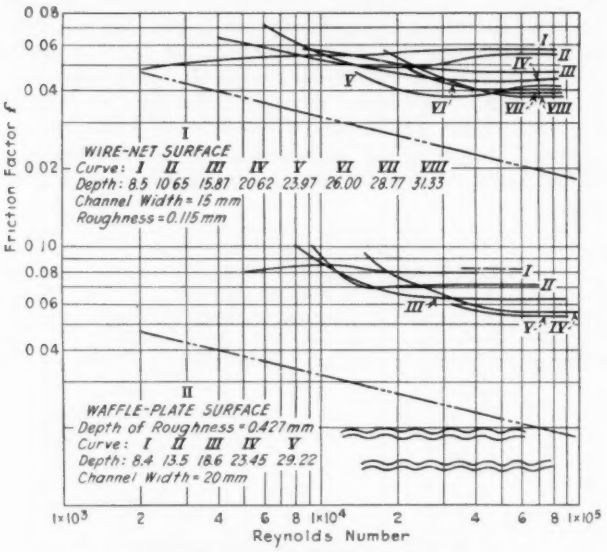


Fig. 12 Flow in Artificially Roughened Pipe

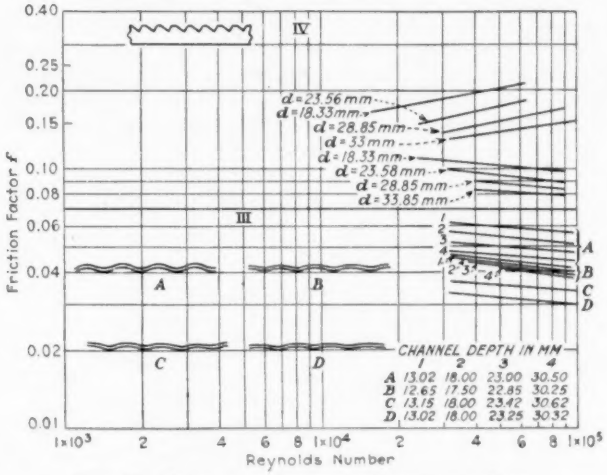


Fig. 13 Flow in Artificially Roughened Pipe

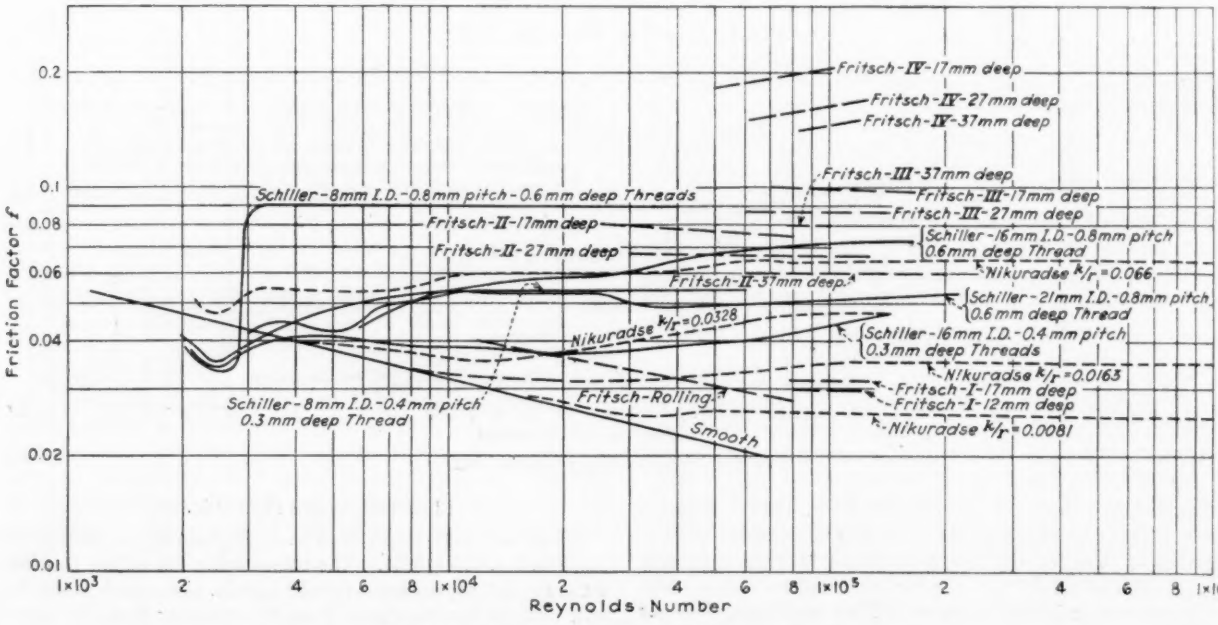


Fig. 14 Flow in Artificially Roughened Pipe

The second series of tests using a waffle-like indented zinc-plate surface have a roughness of 0.427 mm, which means that the difference between the minimum and maximum depth is 10 per cent of the effective depth in the case of the narrow channel. Since it is not certain how this depth was obtained, it cannot be known whether the effective depth is an average or not. If it is an average value, the high friction factor can be explained by using the minimum depth, which would be (8.4 — 0.427) mm. This would reduce the value of f from 0.083 to about 0.058. Since the projections amount to 10 per cent of the depth, contraction and expansion losses might account for the entire loss in case the value given for the depth was the minimum depth.

The sections with a deep channel show an f about equal to the value for a rough pipe. This series of tests also behave in an unusual manner near the critical region.

The third series of tests were made on another corrugated-plate surface which was successively rolled down. The deeper corruga-

[25] Nikuradse, "Flow in Rough Pipes," *Zeit. für Angew. Math. u. Mech.*, vol. 11 (1931), pp. 409–411.

tions show a variation with channel depth, but the smoothed section shows no variation with channel depth. In all cases the values are seen to be below the rough-pipe value.

The fourth series of tests were made using a sawtooth grooved plate with roughnesses 1.5 mm deep. As would be expected, the results show very high friction factors and also a decided variation depending upon the direction in which the flow took place. When flow was against the grooves, the pressure drop varied to a power greater than two, so that some other factor besides wall effect must enter. With roughnesses of this order of magnitude effects such as occurred in the previously described galvanized pipe take place and the results are no longer comparable with those for an ordinary rough pipe.

The results of the tests by Fritsch, Nikuradse, and Schiller are roughnesses of an order of magnitude ordinarily found in corroded pipe, the friction factor would not exceed the value set up for the ideal rough pipe.

15 Miscellaneous (Fig. 15)

The results on spiral riveted pipe show friction factors much higher than on new wrought-iron pipe. This would be expected from the nature of the surface of the pipe. The friction factor is higher for some cases where the flow is against the seam.

The data on natural-gas lines are interesting because they represent actual field tests. The data by Guman have been taken from the *Gas Service Journal*, May, 1930. The numbers on the other points correspond to the test numbers given in the Bureau of Mines' publication.

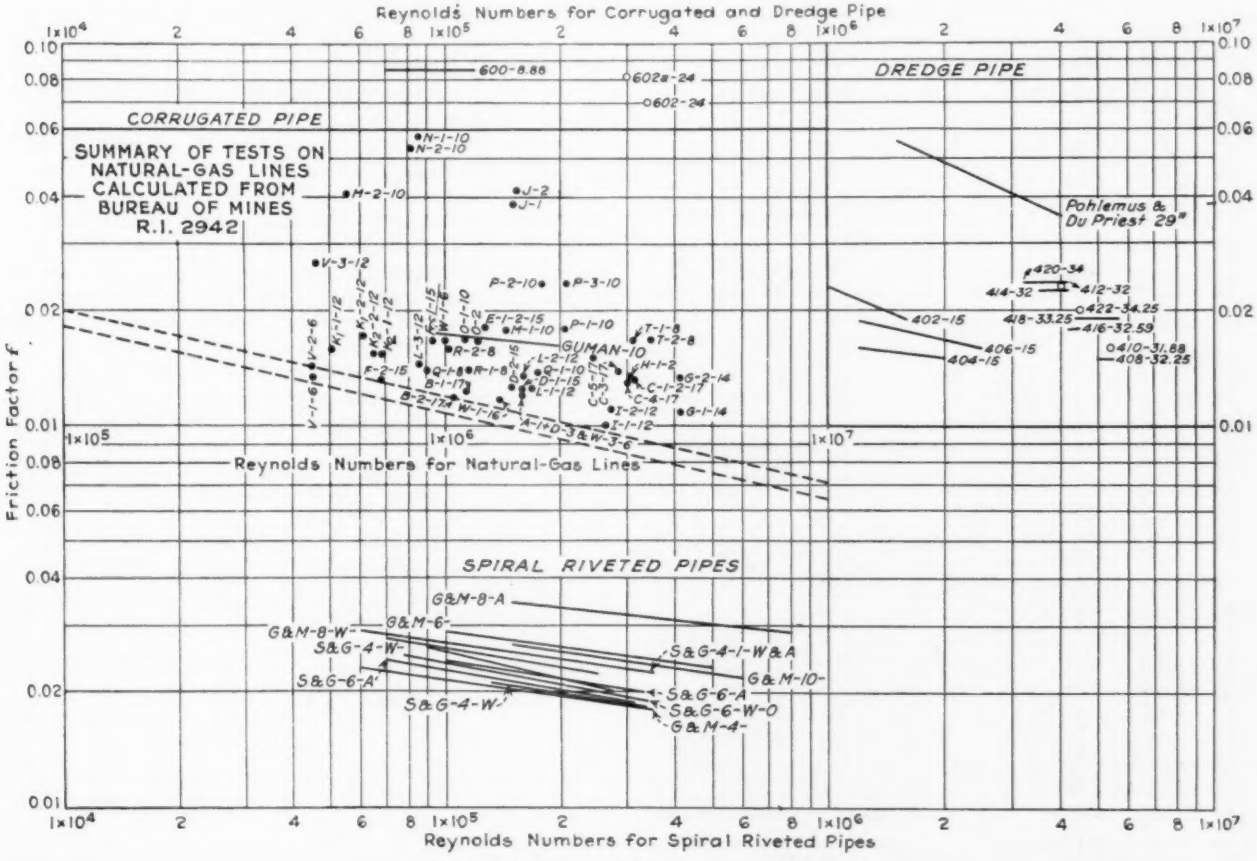


Fig. 15 Flow in Miscellaneous Pipes

shown in Fig. 14. Here again the method of determining the depth or diameter is not definitely known, so no attempt will be made to correct the data. The results obtained by Fritsch are very similar to those by Fromm. The curves marked I are for a roughness that might be found in moderately rough pipes.

The curves for II are similar to that for very rough pipes. Curves marked III and IV are similar to the sawtooth curves of Fromm. Nikuradse's results do not give values above those for rough pipe except for the very roughest pipe, in which case it seems that the original diameter of the pipe was used instead of the working diameter. The use of glue and sand on the inside of the pipes makes the diameter actually smaller than that of the original pipe. A correction for this would bring the friction factor back to the rough-pipe range. In his tests the values of k/r refer to the ratio of the size of sand particles to the pipe radius.

The results on artificially roughened pipes show that with

Since no data are available on the diameters used in the tests on corrugated pipe, little can be said about them.

The data on dredge pipe are seen to give results comparable with those for ordinary pipe, even though there are usually a large number of bends, joints, etc. present in the line.

VII—COMPARISON WITH FORMULAS THAT HAVE BEEN USED

A comparison of the various formulas used in the calculations for fluid flow would be a long study in itself. In order to get a general idea as to the number of and variations in the formulas used the various recommendations made by those who proposed to make f a constant are given. Most of these were taken from the data compiled by Eason[25] and are given in Table 4. It will be seen from an examination of the data that the values vary

[25] Eason, "Flow and Measurement of Air and Gases."

TABLE 4 SUMMARY OF FRICTION FACTORS USED BY VARIOUS AUTHORITIES WHO HAVE PROPOSED MAKING f A CONSTANT

No.	Value of f	Authority	Notes
		STEAM	
1	0.0206	Eberle	Author says f may vary ±5 per cent
2	0.0281	Gutermuth	.....
3	0.0323	Geipel	.....
4	0.0332	Geipel	.....
5	0.0332	Booth	.....
6	0.020	Konig	Wrought-iron pipe
7	0.040	Konig	Cast-iron pipe
8	0.0201	Conrad	8-in. pipe
9	0.029	Daubisson	.....
10	0.017	Stocklapper	.....
11	0.0216	Ledoux	For 47- to 100-mm pipes
12	0.0160	Anocher	For 50- to 100-mm pipes at pressure of 0.3 to 10 kg/cm²
13	0.021	Stodola	.....
14	0.0176	Odell	.....
15	0.0284	Rey	For a 50-mm pipe with superheated steam
16	0.0296	Verner	For steam-heating systems
		AIR	
17	0.012	Kempe	.....
18	0.016	Church	For a 1-in. pipe
19	0.020	Church	.....
20	0.0166	Norwalk Iron Co.	.....
21	0.0255	Batcheller	Results are for 6½-in. air mains
22	0.018	Ledoux	.....
23	0.0184	Kent	.....
24	0.0220	Johnson	.....
25	0.0220	Laidlaw-Dunn	.....
26	0.0220	Rix	For air at 100 lb per sq in. pressure
27	0.0228	Cox	.....
28	0.0110	Giraid	.....
29	0.0132	Brabbée	Same as (19) except air saturated with oil
30	0.0176	Brabbée	For pipes 1⅔ to 2½ ft in diam; vel. varying bet. 13 and 55 fps
31	0.0308	Brabbée	For pipes 1 ft in diam
32	0.0148	Harding	For sheet-metal ducts 12 to 48 in. in diam with vel 25 to 42 fps
33	0.0174	Harding	Same as (21) with 25 per cent allowance for bad alignment
34	0.0176	Treat	For smooth pipes
35	0.0240	Treat	Round galvanized pipes
36	0.023	Eason	2¼-in. brass tube
37	0.024	Innes-Delong	For ventilating ducts
38	0.024	Buffalo-Forge Co.	.....
39	0.0252	Sturtevant	.....
40	0.0253	Sturtevant	For circular galvanized ducts
41	0.0253	Kinealy	For air at atmospheric pressure
42	0.0253	Thorkelson	For square galvanized iron ducts
43	0.0257	Taylor	For pipes over 6 in. in diam.
44	0.0258	Kempl	.....
45	0.0257	Martin	.....
46	0.0280	Hawkesley	.....
47	0.0307	Stanton	Pipes roughened by being cut with double screw threads
48	0.0608	Atkinson	For mine shafts
49	0.0428	Wabner	For mine shafts
50	0.0640	Wabner	For mine shafts
51	0.0252	Grierson	Flow in galvanized iron ducts
52	0.0460	Grierson	For brick and plaster ducts
53	0.026	Bisset	Air in ducts, 2½ to 30 in. in diam
54	0.0072	Taylor	For 1½-in. new clean pipes
55	0.0080	Taylor	For ¾-in. new clean pipes
56	0.0076	Taylor	For ⅝-in. new clean pipes
57	0.01168	Taylor	Same as (54) except pipe was dirty
58	0.0164	Taylor	Same as (55) except pipe was dirty
59	0.0174	Taylor	Same as (56) except pipe was dirty
60	0.0308	McElroy	Canvas pipes 8 to 16 in. in diam.
61	0.0252	McElroy	Average quality galvanized iron pipes 8 to 16 in. in diam
62	0.0186	McElroy	Very good galvanized iron pipes 8 to 16 in. in diam
63	0.0203	Polkinghome	Tests on a mine shaft in Africa
64	0.008	Cullen	Tests on mine shaft
65	0.0104	Cullen	Tests on mine shaft
		GAS	
66	0.0166	Robinson	.....
67	0.0172	Hawkesley	.....
68	0.0181	Oliphant	For a 23-in. diam pipe
69	0.0215	Oliphant	For a 32-in. diam pipe
70	0.0273	Lowe	For 24-in. pipe
71	0.0296	Lowe	For 22-in. pipe
72	0.0228	Hurst	.....
73	0.0768	Hurst	For branch pipes
74	0.0468	Hurst	For main pipes
75	0.0256	Hurst	For large smooth pipes
76	0.0243	Pole	For large smooth pipes
77	0.0294	Pole	.....
78	0.0256	Chapman Valve Co.	.....
79	0.0256	Geipel	.....
80	0.0256	Kilgour	.....
81	0.0258	Schilling	.....
82	0.0260	Beardmore	.....
83	0.0262	Thomas	.....
84	0.0265	Cripps	.....
85	0.0616	Cripps	For small pipes
86	0.0217	Masse	.....
87	0.0326	Masse	.....
88	0.0274	Monnier	.....
89	0.0368	Rix	At 100 lb per sq in.
90	0.0468	Molesionth	.....
91	0.0270	Grebel	Tests on a 3-in. main

widely. Any constant value of f would apply for a limited range of Reynolds' numbers, but at a low enough value of R it would be seen to give a value less than that for smooth pipes.

VIII—FLOW OF COMPRESSIBLE FLUIDS

1 Flow in Horizontal Pipes

The general energy equation ordinarily used for the flow of compressible fluids in horizontal pipes has the form

$$\frac{dp}{\rho} + \frac{vdv}{g} + \frac{fv^2dx}{2\,gD} = 0$$

where ρ = density, p = pressure, v = velocity, V = volume, W = weight of fluid per second, and A = area of pipe. This equation can be derived by using Bernoulli's theorem or from the energy relation as given by Goodenough.[27] This equation is not difficult to solve for isothermal flow, but is very cumbersome to use. Generally the term vdv/g, which is the one taking into account the change of kinetic energy, is omitted because it is very small in most problems.

When f is expressed as a function of the Reynolds number, the equation is justified in its simpler form. In finding the value of f from any set of experiments it will be remembered that the experimental values were substituted in the equation $\Delta h = flv^2/2gD$ and the equation was solved for f. This value of f was then plotted against the corresponding value of R. In the discussion of the flow in smooth pipes it was pointed out that the experimental results gave the same friction factor for both compressible and incompressible fluids. Also in the case for several sizes of rough pipe this same result was noted. The equation for flow of compressible fluids will be derived on the basis of the experimental data, which show that practically the drop in pressure head is given by

$$dh = \frac{fv^2dx}{2gD} \text{ for a length } dx.$$

$$dp = dh\,\rho = \rho\frac{fv^2dx}{2gD} \quad [1]$$

For isothermal flow we have from Boyle's law

$$p_1V_1 = pV = p_2V_2 \quad [2]$$

The equation of continuity gives

$$Av = V \quad [3]$$

$$v = \frac{p_1V_1}{pA} \quad [4]$$

$$\rho = \frac{W}{V} = \frac{W}{Av} \quad [5]$$

$$f = \phi\left(\frac{Dv\rho}{\mu}\right) = \phi\left(\frac{Dv}{\mu}\frac{W}{Av}\right) = \phi\left(\frac{DW}{A\mu}\right) \quad [6]$$

Equation [6] shows that the friction factor depends on the size of pipe, viscosity of fluid, and the weight of fluid flowing. Then for any given set of conditions, f is constant. The friction factor depends only on the initial or final conditions of the gas and can be considered constant in Equation [1].

$$v = \frac{W}{A\rho} \quad [7]$$

so that Equation [1] becomes

$$dp = \frac{fWv\,dx}{2AgD} \quad [8]$$

[27] Goodenough, "Principles of Thermodynamics," Henry Holt & Co.

Substituting for v in terms of p,

$$dp = \frac{fWp_1V_1dx}{2AgDpA}$$

$$pdp = \frac{fWp_1V_1}{2gDA^2}dx, \text{ or } \int_{p_2}^{p_1} pdp = \int_0^l \frac{fWp_1V_1dx}{2gDA^2}$$

$$p_1^2 - p_2^2 = \frac{fWp_1V_1l}{gDA^2} \quad \ldots\ldots\ldots\ldots\ldots [9]$$

$$p_1V_1 = p_1v_1A = p_2v_2A \quad \ldots\ldots\ldots\ldots\ldots [10]$$

$$p_1^2 - p_2^2 = \frac{fWp_2v_2l}{gDA} = \frac{f\rho Av_2^2p_2l}{gDA}$$

$$p_1 = \sqrt{p_2^2 + \frac{fWp_2v_2l}{gDA}}$$

$$p_2 = \sqrt{p_1^2 - \frac{fWp_1v_1l}{gDA}}$$

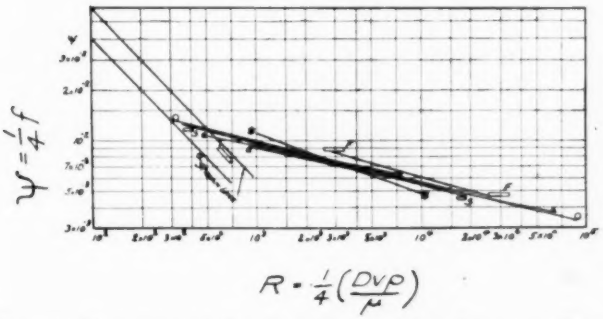


FIG. 16 VARIATION IN FRICTION FACTOR WITH SHAPE OF PIPE SECTION

2 FLOW IN VERTICAL PIPES

(*a*) *Flow Up the Pipe.* In this case there is in addition a loss in head due to the vertical displacement and the equation becomes

$$dh = \frac{fv^2dx}{2gD} + dx$$

$$dp = \rho dh = \frac{f\rho v^2dx}{2gD} + \rho dx$$

$$= \left[\frac{f\rho v^2}{2gD} + \rho\right]dx$$

$$= \left[\frac{fWv}{2AgD} + \frac{W}{Av}\right]dx = \left[\frac{fWp_1V_1}{2A^2gDp} + \frac{Wp}{p_1V_1}\right]dx$$

Let

$$\frac{fWp_1V_1}{2A^2gD} = C_1 \text{ and } \frac{W}{p_1V_1} = C_2$$

then

$$dp = \left[\frac{C_1}{p} + C_2p\right]dx = \left[\frac{C_1 + C_2p^2}{p}\right]dx$$

$$\int_{p_2}^{p_1} \frac{pdp}{C_1 + C_2p^2} = \int_0^l dx$$

$$\frac{l}{2C_2}\log\frac{\left(p_1^2 + \frac{C_1}{C_2}\right)}{\left(p_2^2 + \frac{C_1}{C_2}\right)} = l$$

or

$$\frac{\left(p_1^2 + \frac{C_1}{C_2}\right)}{\left(p_2^2 + \frac{C_1}{C_2}\right)} = e^{2C_2l}$$

(*b*) *Flow Down the Pipe.* In this type of flow the loss of head is decreased, for at every point the weight of fluid above the point increases the pressure. The equation would then be

$$dp = \frac{f\rho v^2dx}{2gD} - \rho dx$$

This change means that the sign of C_2 in the preceding equations would be changed; otherwise there is no difference. We can then write the solution directly as

$$\frac{\left(p_1^2 - \frac{C_1}{C_2}\right)}{\left(p_2^2 - \frac{C_1}{C_2}\right)} = e^{-2C_2l}$$

IX—THE MEAN HYDRAULIC RADIUS AS A CRITERION FOR FLOW

The equations and curves for making flow calculations have been given for circular pipes. Often it is necessary to make calculations for non-circular shapes, annular shapes, and cases with a free surface. In order to handle these problems a term called the mean hydraulic radius has been devised. This has been selected on the supposition that the resistance for pipes of the same area is proportional to the fluid in contact with the surface of the pipe. If we represent this mean hydraulic radius, or profile radius as it is sometimes called, by r, we have

$$r = \frac{\text{area}}{\text{wetted perimeter}}$$

Schiller[28] has done a large amount of experimental work on this subject. His results are shown in Fig. 16, which is taken from the "Handbuch der Physik." It is seen that for ordinary shapes the values of the friction factor in the turbulent region are very close to those for a circular shape. In the streamline or viscous region the variation is as much as 50 per cent more than for circular shapes. Caldwell has made experiments with different shapes in this region and found the discrepancy to be large in some cases. The case of streamline flow can be expressed by the differential equation[29]

$$\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} = \frac{\Delta p}{\mu}$$

for flow in a uniform section; x and y represent the coordinates of the section, Δp the drop in pressure per unit length, and μ the viscosity of the fluid.

The mathematical solution of this equation is possible in only a few special cases, which are the sections for which the torsion problem has been solved. The soap-film analogy proposed by Prandtl, and which is used in solving complicated torsion problems, can be applied to this problem. The differential equation of the soap film is given by[30]

$$\frac{\partial^2 z}{\partial x^2} + \frac{\partial^2 z}{\partial y^2} = \frac{\Delta p}{2T}$$

[28] Hopf, "Handbuch der Physik," vol. 7, p. 143.
[29] Lamb, "Hydrodynamics," Cambridge University Press.
[30] "Mechanical Properties of Fluids," D. Van Nostrand Co., p. 230.

where z is the height of the soap film above the section, x and y the coordinates of the section, T the tension in the film, and Δp the pressure difference on the two sides of the film. If then we let

$$\frac{\Delta p}{\mu} = \frac{\Delta p}{2T}$$

the velocity at any point in the section will correspond to the ordinate of the soap film and the volume of the soap bubble will correspond to the volume of flow. Methods for measuring these quantities have been devised for use in solving the torsion problem. The method described also permits a mental picture of the approximate velocity distribution over such a section if quantitative results are not desired. Such types of sections as thin annular spaces correspond to a long rectangular section, etc. By use of the values determined for the torsion of rectangular sections it is possible to calculate flow in annular spaces.

For streamline flow it is only safe to base the flow on the mean hydraulic radius if the section is circular or very nearly so. The following solution for the flow in a concentric annular section is given in Lamb's "Hydrodynamics."[31] If r is the large radius of the annulus and a the smaller radius, the quantity discharged is given by

$$Q = \frac{\pi}{8\mu}\frac{\Delta p}{l}\left[r^4 - a^4 - \frac{(r^2 - a^2)^2}{\log r/a}\right]$$

For a circular pipe

$$Q = \frac{\pi r^4}{8\mu}\frac{\Delta p}{l}$$

where Δp represents the drop in pressure.

If we take $a = 0.1\ r$ the mean hydraulic radius will be practically the same for both cases but the flow for the annular space will be

$$Q = \frac{\pi}{8\mu}\frac{\Delta p}{l} r^4 \left[1 - \frac{1}{\log_e 10}\right] \text{approximately}$$

$$= \frac{\pi}{8\mu} \times \frac{\Delta p}{l} r^4 \left(1 - \frac{1}{2.3}\right) = 0.53\,\frac{\pi r^4}{8\mu}\frac{\Delta p}{l}$$

For a circular pipe

$$Q = \frac{\pi r^4}{8\mu} \times \frac{\Delta p}{l}$$

So for the same pressure drop the flow in the annular space in which the inner radius is one-tenth of the outer radius is only 53 per cent of that for a circular pipe of the same outside diameter.

A bibliography of this subject is given by Bateman[14] in his recent book, together with a brief discussion of the principal works on the subject.

CONCLUSIONS

Some of the more important conclusions to be drawn from the foregoing data are:

1 The friction factor is independent of the fluid flowing for the same values of R.

2 The friction factor for drawn brass pipe varies only ±5 per cent from an average value and is substantially independent of the pipe size and fluid flowing for the same value of R within the range encountered in practice.

3 The friction factor for new steel pipe varies with the size and varies about ±10 per cent from an average value. The reason for the variation with size is because the roughness increases with decrease in diameter.

4 The data on used and artificially roughened pipes show that if the roughness is not large enough to cause contraction and enlargement losses, the friction factor never exceeds the value 0.054.

5 The mean hydraulic radius can be safely used as a criterion only in the turbulent region.

6 Special care should be taken in the selection or determination of the diameter.

Of these conclusions, the most important one is that there is an upper limit for the friction factor. While this upper limit has not been proved conclusively, the data show that unless the roughness becomes of such an order of magnitude as to cause effects such as occur when there are contractions and enlargements, the friction factor will not become greater than 0.054. The limits of roughness are yet to be determined and will probably have to be ascertained by taking velocity traverses along as well as across the pipe.

The upper and lower limits of the friction factor having been determined, the next problem is the selection of values for various types and sizes of pipes. This problem has been discussed by R. J. S. Pigott in a recent paper.[20] His curves for the various types and sizes of pipe are the best and most extensive that have been proposed for design work and should be considered in connection with the present paper.

The effect of diameter on flow is worthy of further mention. It has been seen that $v = V/A = 4V/\pi d^2$, so that

$$\Delta h = \frac{8flV^2}{g\pi^2 D^5} \text{ or } f = \frac{\Delta h g \pi^2 D^5}{8lV^2}$$

From this it can be seen that a small change in D would produce a very large change in f. Since all quantities in the equation for Δh are fixed except D, the value of f will vary inversely as D^5. If, for example, in the design of a pipe line the value of f is selected and a value of $D = 1$ is used while the actual value of $D = 0.9$, the calculated Δh would be $8fV^2/g\pi^2$, whereas the actual Δh would be $1.7 \times 8fV^2/g\pi^2$. The pressure drop would actually be 70 per cent greater than the value based on $D = 1$. Experimental results would show f 70 per cent higher than was actually the case. For design purposes where incrustations are expected it is much safer to estimate the ultimate diameter and use $f = 0.054$ than to estimate the ultimate value of f.

In the present study an attempt was made to obtain all the data available on flow in pipes under pressure, but in a number of cases references were not available and some data have no doubt been overlooked. Many engineers have experimental data which have not been published and which would be a valuable addition to this study. Because of the expense of publishing the individual curves with comments, and the need for economizing at this time, the author would be glad to receive and compile them in an appendix, together with a table giving the name of the experimenter, the fluid used, and brief comments.

ACKNOWLEDGMENTS

The author wishes to express his appreciation to Mr. R. J. S. Pigott, who proposed this research problem, and whose personal interest and help have made this paper possible. To him credit is due for seeing the practical possibilities of the work. The author also acknowledges the assistance of Mr. H. G. Botset, who has made many of the German translations, and his indebtedness to Prof. Linn Helander and Dr. Morris Muskat for their many helpful suggestions.

[31] Lamb, "Hydrodynamics," p. 555.

Discussion

J. M. Spitzglass.[32] The author's analysis shows three grades of fluid conduits: the smooth pipe, the rough pipe, and the irregular pipe. The smooth pipe having little disturbance to cause and to accelerate the condition of turbulence retains the condition of streamline flow to a larger value of the Reynolds number and a correspondingly lower value of the friction coefficient. The flow in the rough pipe "bursts" into the turbulence much sooner, and the "ideal" rough pipe, as the author has shown, makes an abrupt change from one mode of motion to the other. In other words, the change is so complete that from this point on the friction drop of pressures becomes proportional to the square of the velocity, and therefore the coefficient f remains constant. In the smooth pipe, however, not only does turbulence start at a lower value of f, but the turbulence when started is not as yet complete. The value of f, as shown, continues to drop, indicating that the complete proportionality of V^2 is approached only when the Reynolds number becomes very large.

Let us assume a given flow in a pipe of intermediate roughness where the friction drop is just reaching the point of porportionality to V^2. If from this point on the friction coefficient retains the same value, it would indicate that complete turbulence has been reached for the given condition. On the other hand, since the value of f at that point is necessarily less than the value of f in the "ideal" rough pipe, we must admit that the accidental turbulence in the smoother pipe is less effective than the corresponding turbulence in the rough pipe. This being the case, it is just as natural to assume that a further increase in the turbulence factor may increase the effectiveness of the turbulence, with the result that from this point on the friction coefficient should increase instead of decreasing with the increase of the turbulence factor.

Tests made some 20 years ago for the Peoples Gas Light & Coke Company of Chicago on the flow of illuminating gas in service pipes, the results of which were published by the Illinois Gas Association in 1912, have led to the belief that in the case of commercial gas pipes the coefficient of friction reaches a minimum value of 0.00367 in the 10-in. and 12-in. pipes. From this minimum the coefficient increases considerably as the size of the pipe decreases, and it also increases somewhat as the size of the pipe increases.

Of course, tests on commercial pipes, usually of unknown irregularity, are not conclusive to establish an assumed law of variation in the physical quantity. The unfortunate part of it is that with few exceptions the data on smaller sizes come from laboratory experiments, while the data on larger sizes come from tests on commerical pipe lines. The writer would like to ask the author whether his extensive analysis of the existing data would indicate a proof or a disproof of the assumption that under certain conditions the friction coefficient of smooth pipes may increase instead of decreasing with the increase of the turbulence resulting from the increase of the diameter of the pipe.

Theodore Baumeister, Jr.[33] The Reynolds number criterion has been accepted for some time as the most exact basis of comparing fluid dynamic circumstances. No one, however, until the present author, has attempted to analyze the bulk of older empiric and semi-empiric methods and to reduce these data to the common base of Reynolds' numbers.

The weakness of the Reynolds number method for the estimation of frictional resistance, aside from its cumbersome arithmetic and manipulation, rests in the designer's inability accurately to predict the relative friction coefficients when pipe other than some convenient laboratory standard is to be used. The author has contributed much to alleviate this condition by a careful reconsideration of available fact data. The statement at the close recites briefly the next problem of correlation for design purposes. It would be further contributive if the author would list the relative values of friction factor for the outstanding classes of commercial piping and duct work. Gibson gives such values, and the writer has used the following with some degree of success in the design of piping systems:

New, clean smooth, seamless drawn brass tubing	1.0
New galvanized-iron pipe	1.2
High-pressure steam lines	1.3
Sheet-metal ducts	1.3
New black wrought-iron pipe	1.4
New cast-iron pipe	1.5
Very smooth concrete pipe	1.8
Fiber ducts	5.0

In the light of the author's research these and similar tabulations must be open to criticism, and the author's comments on them should prove pertinent.

R. P. Genereaux.[34] Schiller is quoted as concluding 1160 to be the critical Reynolds number. As Schiller's Reynolds' number is $ru\rho/\mu$, where r is radius, the value 1160 should be multiplied by 2 to obtain $du\rho/\mu$, or 2320. The author states that "there are no experimental data on flow at very high Reynolds' numbers for smooth pipe." Since Mr. Kemler's correlation was made, Nikuradse has published his most recent data on the flow of water in smooth pipes (see *Forschung*, 3; Forschungsheft, 356, 1–36, 1932, Sept.–Oct.). With inside diameters of 10, 20, 30, 50, and 100 mm, Nikuradse obtained pressure-drop data for a range of Reynolds' number (in consistent units) from 3×10^3 to 3240×10^3, which upper value is somewhat higher than any for this case heretofore published. He compares his data with those of Stanton and Pannell and of Schiller and Hermann, and recommends a curve beyond the Reynolds numbers of these investigators. At a Reynolds number of 10^7, Lees' formula for Stanton and Pannell's data gives $f = 0.00937$, Schiller gives $f = 0.00854$, and Nikuradse gives $f = 0.00805$ by his formula $f = 0.0032 + (0.221/Re^{0.237})$.

Glen N. Cox.[35] It is regretted that the author did not present data on the behavior when streamline flow existed. To say that this type of flow can be handled mathematically is one thing, but to know that one can rely upon the mathematical analysis is quite another. For very viscous fluids, it may be quite difficult to obtain velocities great enough to produce turbulent flow.

The writer has had occasion to make some tests on the resistance of molasses in pipe. In this work, molasses that was not quite as thick as the black strap ordinarily encountered was pumped through a $1^1/_2$-in. pipe having a gage length of 42.16 ft. The pressures were measured with Bourdon gages and the discharge was weighed. The results of these tests did not approach the critical Reynolds number. With a temperature of 20 C, it would require a velocity of 65 ft per sec in the $1^1/_2$-in. pipe to obtain 2000 for Reynolds' number. This velocity would be accompanied by a pressure drop of 810 lb per sq in. for each 100 ft length of pipe.

This example clearly shows the impossibility of operating in the turbulent zone without heating the molasses considerably. There are also other very viscous fluids that must be pumped and which would also follow streamline flow. For this reason, the writer believes that some consideration should have been given data covering this type of flow.

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[34] Chemical Engineer, Experimental Station, E. I. du Pont de Nemours & Co., Wilmington, Del. Jun. A.S.M.E.

[35] Associate Professor of Mechanics and Hydraulics, Louisiana State University, Baton Rouge, La.

LYMAN M. VAN DER PYL.[36] The paper is valuable for those interested in the flow of fluids as it puts all previous work before an investigator. There are a few recent papers that were not included, and abstracts of them are given herewith.

Naumann's "Experimentelle Untersuchungen über die Entstehung der Turbulenten Rohrströmung," in *Forschung auf dem Gebiete des Ingenieurwesens*, 2, 85–98 (1931), deals with the method by which turbulent flow starts. Photographs show the turbulence in various forms as the Reynolds number increases.

In Wunsch and Zipperer's "Versuche zur Bestimmung des Fortleitungswiderstandes der Ferngasleitung Hamm-Hannover," *Gas- und Wasserfach*, 57, 428–33, 672 (1932), the line used was a new gas line 500 mm i.d., 95.927 km long. The inner surface of the pipe had a thin coating of bitumen. The test was made with air, with the following results:

Reynolds' number	Friction factor
146800	0.0168
147000	0.0168
247500	0.0152
244400	0.0159

The data fit very well on the curve of Stanton and Pennell, which was checked by Jakob and Erk. A shorter account of the same test is given by Zipperer in the *Z.V.D.I.*, 76, 916–917 (1932).

A third paper is Viesohn's "Untersuchen über Druckhöhenverluste in Rohrleitungen und Armaturen für die Hausleitungen der Wasserversorgung," *Gas- und Wasserfach*, 75, 679–685 (1932). His work was done in pipes 40 m long and 10 to 50 mm in diameter. The materials were lead, brass, copper, and galvanized iron, all new. The results are expressed as the coefficient λ of the formula $h = \lambda(Q^{1.802}/d^{4.854})$. The values of λ for the lead, brass, and copper pipes was found to be 0.00081 and for the galvanized-iron pipe 0.000977. Tests made with old pipes showed that brass, copper, and lead pipes were scarcely affected as compared with new pipes, while for old galvanized-iron pipe the value for λ was two to three times higher than for the new pipe. This formula used by Viesohn is not as convenient as the usual flow formula for plotting the relation of friction factor to the Reynolds number. The friction factor f from this formula is

$$f = \frac{k}{\left(\frac{vd}{\nu}\right)^{0.198} d^{0.052}}$$

so that it, like the Hazen-Williams formula, gives a slightly different curve for the f — (vd/ν) relationship for each pipe diameter.

ALTON C. CHICK.[37] So long as viscous forces are predominant and gravitational forces have negligible effect, it has been shown by many authors[38] that the friction coefficient f must be a function of the dimensionless ratio $vd\rho/\mu$ called Reynolds' number, or R, and that in smooth pipes this is the only criterion necessary for establishing similarity of flow.

The author has analyzed the data of many experimenters and has evaluated these on a common basis. It is a regrettable fact that many early experimenters did not include temperature observations, which would have opened to our use many more experiments.

The author has referred to experiments by Mr. John R. Freeman, and has used certain of Mr. Freeman's data. More concerning these experiments may be of interest. In 1892 Mr. Freeman organized and carried out what is probably the most complete series of experiments on the flow of water through commercial pipes ever yet made by a single experimenter. These tests covered a wide range of pipe materials and pipe sizes as well as many pipe fittings. Many times during the past 40 years Mr. Freeman undertook the publication of the results. Only two months prior to his death on October 6, 1932, the publication of these results was resumed, and the writer had recomputed much of the data on the scientific basis of Reynolds' numbers. Preliminary plottings show interesting results. The experiments on the smooth, seamless, drawn brass pipe cover a range of Reynolds' numbers from 175 to 880,000, and show a remarkable coincidence with the Stanton-Pannell curve in both the laminar and turbulent flow regions. The coefficient f appears to be entirely independent of the diameter of these smooth brass pipes. Preliminary plottings of data observed on the flow in new wrought-iron pipes indicate that the diameter does have an important effect on the coefficient f. The f-R plottings for the different pipe sizes form a family of curves that tend to converge in the vicinity of the upper range of the critical velocity and to diverge as the values of Reynolds' numbers increase. The larger the diameter of this new commercial wrought-iron pipe the nearer the f-R curve approaches that for smooth pipes; in other words, as the diameter of the pipe is increased so the relative roughness of ordinary commercial pipe is decreased.

In the case of pipes of the same geometrical form and with the same Reynolds number, but with different wall characteristics or different hydraulic radii, different values of the coefficient f are obtained. Therefore mechanical similarity of two flows is not assured through the equality of the Reynolds numbers.

If the relative roughness is known it becomes possible to determine in advance the pressure loss for a given flow.[39] However, many more experiments on the flow in pipes of known wall roughness are needed to determine a practical and useful relationship. It is hoped that the preparation of the data observed by Mr. Freeman will soon be completed and this information made available to others.

GEORGE I. RHODES.[40] The author has reduced to a common denominator much little-known but valuable information. Data coordinated by him were available in many forms, in metric and in English units, in the absolute and in the gravitational systems of units, and in some cases in mixtures of units and systems.

Equation [1] of the paper can be changed readily to the fundamental form

$$\frac{\text{Loss of head}}{\text{length}} = \text{Friction factor} \frac{\text{Velocity head}}{\text{Diameter}}$$

in which it is obvious that the friction factor has the same value whether loss of head, length, velocity head, and diameter are expressed in feet, centimeters, or any other measure of length, so long as they are all expressed in the same measure. Some investigators in reducing their results to a friction factor used an equation differing from the foregoing only in that radius is used instead of diameter, in which case the friction factor is one-half that of the author's curves; and other investigators prefer to use, in place of diameter, mean hydraulic depth, which has the distinct advantage of being universally applicable to a conduit of any cross-section, and the friction factor used by such investigators is one-quarter that of the author's data.

Reynolds' number is to most engineers a quantity as "ghostly"

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[37] Formerly principal assistant to the late John R. Freeman, consulting engineer, Providence, R. I. Jun. A.S.M.E.

[38] See "Hydraulic Laboratory Practice," A.S.M.E., 1929, pp. 775 to 827.

[39] *Zeit. f. angew. Math. u. Mech.*, Band 3, Heft 5, s. 339, K. Fromm.

[40] Vice-President, Ford, Bacon & Davis, Inc., New York, N. Y. Mem. A.S.M.E.

as entropy. They have little concrete conception of viscosity. They do not appreciate that viscosity differs from modulus of elasticity in shear only in that angular velocity of movement is the inverse measure of viscosity, whereas angular displacement is the inverse measure of shearing modulus of elasticity. Perhaps some conception of Reynolds' modulus can be obtained by the following approximations applicable to certain fluids at ordinary temperatures. In the case of water Reynolds' number is 105,000 with a flow of 1 cu ft per sec in a pipe 1 ft in diameter and increases or decreases in the ratio of flow to diameter. In the case of natural gas or air, Reynolds' number is about 8000 with a flow measured in free gas or air of 1 cu ft per sec in a pipe 1 ft in diameter, which increases and decreases with the ratio of flow to diameter. In the case of steam at 250 lb pressure, Reynolds' modulus is about 18,000, with a flow of 1 lb of steam per minute in a pipe 1 in. in diameter and increases and decreases with the ratio of flow to diameter.

The author uses in his equations a definition of density that is correct in the absolute English system of units, but which definition is that of specific weight in the English gravitational system of units used in most engineering equations. He uses the symbol for density as specific weight in Equation [1], which therefore must be an equation in the gravitational system, but in Equation [2] he uses it as density, which equation therefore must be an equation in the absolute system. So long as density and viscosity are expressed in the same system of units, however, Equation [2] is correct irrespective of the system used. This double use either of the symbol for density or of the systems of units has apparently led to no confusion of results. It is pointed out only as an illustration of the pitfalls one encounters in attempting a work such as is summarized in this paper.

The author's statement that "at a Reynolds number of 1160 the friction factor should be independent of the pipe size regardless of its roughness" is far from proved. Curvature and irregularity are quoted by him as producing turbulence at Reynolds' numbers far below 1160, and in such cases with increased velocities sufficient to result in a Reynolds number of 1160, the turbulence must have been complete, with a resultant friction factor higher than that for viscous flow. The curves in Fig. 8 for used cast-iron pipes show friction factors well above those for viscous flow at Reynolds' number 1160. It is hard to conceive that the curve of friction factor against Reynolds' number, of any pipe however large or small or however smooth or rough, will intersect the theoretical curve for viscous flow at any one fixed point.

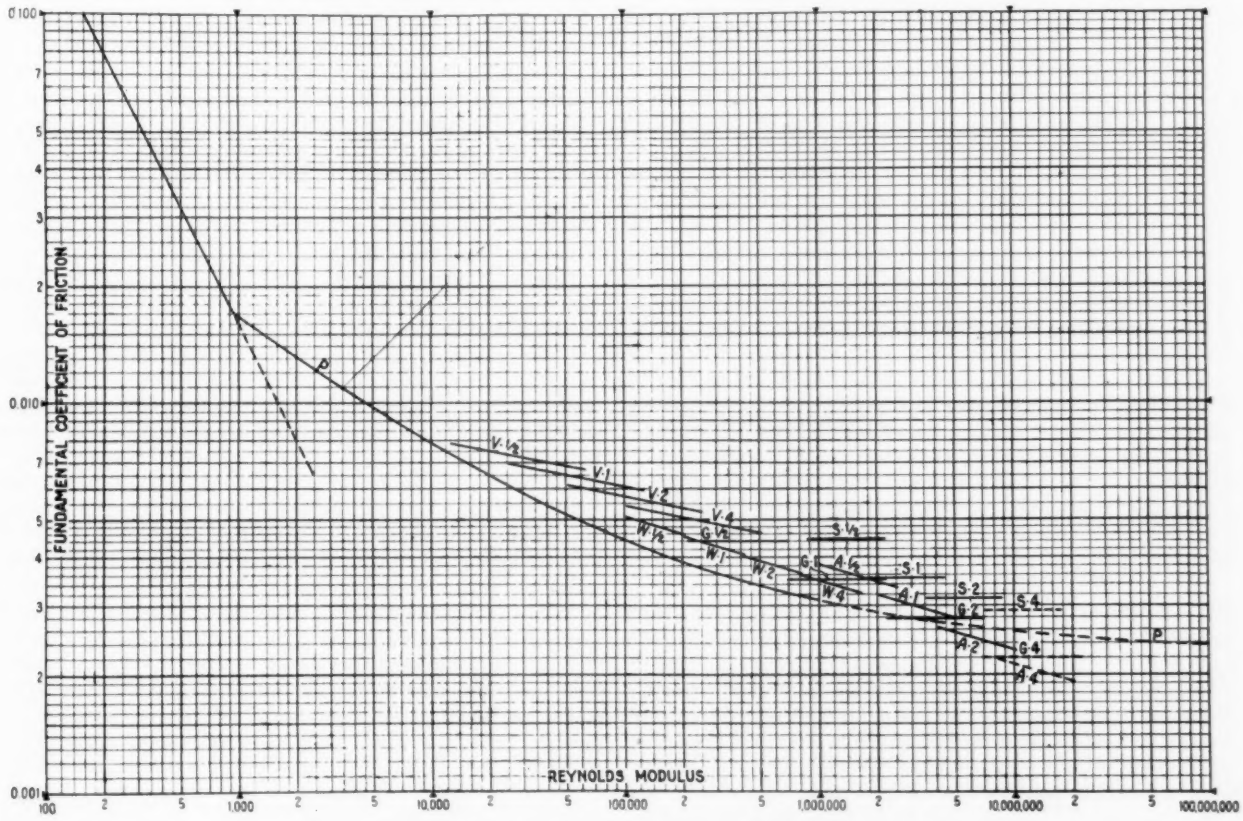


FIG. 17 COEFFICIENTS OF FRICTION IN COMMON PIPE-LINE FORMULAS

(Legend: *P*, oil lines from Heltzel, *Oil and Gas Journal*, June 5, 1930; *V*, ventilating ducts, velocities 4 to 20 fps, Marks' s Mechanical Engineers' Handbook; *W*, water lines, velocities 2.5 to 5 fps, Williams-Hazen formula, $C_w = 140$; *A*, air lines at 100 lb per sq in. gage pressure, velocities 40 to 100 fps, Fritche's formula; *S*, steam lines at 250 lb per sq in. abs pressure and 600 F, velocities 60 to 100 fps, Babcock formula; *G*, natural-gas lines at 40 lb per sq in. gage inlet pressure, 4 to 52 per cent pressure drop in 100 miles, Weymouth formula. Numbers following curve references are inside diameter in feet.)

To this extent the lower chart of Fig. 4 summarizing data on steel pipes is not fully justified.

Section VII of the paper compares the friction factors of formulas that have been used in the past. To supplement that table there is offered a chart (Fig. 17) of the coefficient of friction in common pipe-line formulas which shows the comparison between Reynolds' number and the fundamental coefficient of friction used in these various pipe-line formulas within the range of common practice. The dash lines in the chart show extensions beyond the sizes of pipe commonly used. In this chart the fundamental coefficient of friction is that required in the fundamental equation mentioned earlier in this discussion where the denominator of the right-hand half of the equation is the mean hydraulic depth—namely, one-quarter the diameter. Accord-

ingly the values of fundamental coefficient of friction in this chart are one-quarter of the values given in the author's paper. It should be noted that this chart is on a distorted scale, the length of one cycle of ordinates being twice the length of one cycle of abscissas.

Ordinary formulas, it would appear, take into account to some extent variations in the coefficient of friction and are in form suited for a convenient computation. There is some doubt if it is worth while to take these variations into account more accurately on account of vagaries in performance of pipe lines not fully explainable.

It has been found of great help in evaluating charts of coefficient of friction against Reynolds' number to have appear on the chart a portion of the line for viscous flow. This line will intersect the line of Reynolds' modulus having a value of 1000 at a friction factor of 0.016 or 0.032 or 0.064, depending upon whether the fundamental formulas used are based on hydraulic depth, pipe radius, or pipe diameter, respectively.

Pipe lines as actually installed do not perform as closely to expectations as might be inferred from the author's statements as to the accuracy of his curves. Some pipe lines are as much as 25 per cent below par when measured by the friction factor and some are 25 per cent above par. Tuberculated iron water conduits are so much below par that mere reduction in diameter from the tuberculation fails completely as an explanation of low capacity. Roughness is undoubtedly an important factor, but there is little understanding of the manner in which it works its nefarious purpose. Its effect may be generally explainable as that of a series of shallowly projecting diaphragms, which effect if applied quantitatively does at least account for the behavior of tuberculated pipes. It is hoped that this diaphragm or orifice theory may be fully developed because it seems to offer interesting possibilities with pipes of controlled smoothness. Research concerning the effects of roughness is much needed, and it will certainly be well worth while if as a result we can specify and control roughness to the end that all pipe lines are as good as the best now built, namely, 25 per cent above the average, or even better in the case of very large lines.

R. J. S. Pigott.[41] It may be of interest to indicate the occasion for the analytical research undertaken by the author. Much of the work in which both the author and the writer are engaged involves flow of liquids and gases, and also in a great many cases the flow of mixtures, on which there are few experimental data. As a preliminary to the undertaking of the more difficult problems of mixed flow, a study was made on the subject of purely liquid or purely gaseous flow.

The one point that struck us most forcibly was that in practically all previous dissertations on the subject the theories and formulas were predicated only on a portion of the available data, with the result that the formulated results of many writers differ in form to some extent, but give only slightly differing results. In many cases the reasoning employed to reach these formulated results shows very wide divergences due to the recorded data investigated in each case. We therefore thought it would be desirable to recalculate on the dimensional basis all of the available data so as to show the range of variation and the trends very much more authoritatively.

In his investigations the author soon found that much of the presumably useful data on the subject were lacking in one or more of the required factors such as accurate knowledge of diameter, roughness, and viscosity. As a result, much of the data which have been reported could not be used because accurate information on one or more of these factors was missing or incomplete. However, the remaining mass of data was very much greater than had been correlated before by any one investigator, and it is certainly clear from the author's results that at last we are in a fair position to formulate with some decision the effect of roughness, which has hitherto been largely neglected except by purely empirical methods such as exemplified in the Williams and Hazen well-known tables.

After studying the paper, it becomes obvious that many of the friction-factor curves offered by earlier investigators have shown marked differences to those obtained in this paper, because the roughness was not constant for the different sizes and kinds of pipe for which the curves were drawn.

Another point rather clearly brought out is that all values for the friction factor in the turbulent region exceeding the value at the critical point must be incorrect; in general, it may be noticed that such high values are only found with rough pipe, and in practically every case there is evidence that the diameter was not correctly obtained. When such results are corrected to the true diameters, friction factor always remains under the value at the critical point in the $dv\rho/\mu$ plot. It would seem that this conclusion is of some importance, inasmuch as it insures that the friction-factor lines for all roughnesses of pipe, from perfectly smooth brass to perfectly rough materials, must pass fanwise through the value at the critical point. This differs from the assumption made in the Williams and Hazen formulas and many others of similar type, that the friction-factor curves for varying degrees of roughness are all parallel. This latter assumption has resulted in such anomalies as assigning higher roughness to small brass pipe than to cast iron.

These points are covered more fully in the writer's paper on "The Flow of Fluids in Closed Conduits," delivered before the National Process Meeting in Buffalo, June, 1932. This paper should be regarded more or less as a continuation of the investigation covered by Mr. Kemler's paper. It is expected that the work will be continued to collect the material on loss of pressure due to bends, valves, and other fittings. We are also expecting to carry out some research experiments on the flow of mixtures of gas and liquid in both horizontal and vertical pipes. This problem is extremely important in the investigation of the behavior of flowing oil wells, air lifts, and similar conduits carrying mixed fluids. As yet, we have been unable to find a strict physical basis for the variation of the friction factor with Reynolds' number, but the investigations, so far as they have gone, indicate that a mathematical investigation of the shape of the velocity traverse in a pipe will give the cue for the physical meaning of the Reynolds criterion.

Benjamin Miller.[42] This paper gives the impression that the scientific basis for analyzing the results of experiments on the flow of fluids was originated by Reynolds and that Reynolds also discovered that there are two types of fluid motion. Such an impression would be far from the truth. It is not the purpose of this discussion to minimize the contribution of Reynolds, but rather to place it in its proper light.

As Stanton and Pannell point out in reference 4 of the paper, Newton, Stokes, and Helmholtz preceded Reynolds in the theoretical treatment of the problem, so that Reynolds cannot be said to have developed the equations for dimensional homogeneity and dynamic similarity. Bingham, in the preface to his book "Fluidity and Plasticity," says, "It was in 1842 that viscous flow was first differentiated from hydraulic flow," and gives credit on pages 35 and 36 to Darcy as "probably the first to distinguish clearly between the two régimes."

The importance of Reynolds' work lies in his experimental demonstration of the theories which had been put forth before

[41] Staff Engineer, Gulf Production and Pipe Line Companies, Pittsburgh, Pa. Mem. A.S.M.E.

[42] Henry L. Doherty & Co., New York, N. Y.

and in his determination of the region in which the two régimes overlap. He was also the first to attempt to give explicit form to the relation expressed by $f = \phi(Dv\rho/\mu)$, but subsequent experiments showed that his formula is incorrect.

Why did two centuries elapse between the publication of Newton's theoretical deduction of the criterion for similarity and the publication of Reynolds' experimental verification? And why, after another half century, do we find many engineers unwilling to accept it and use it in their practice? And, on the other hand, how does it happen that it is being used by other engineers? Any answers must be speculative, but such speculation is useful as well as interesting, since it illustrates the conflict between abstract philosophy and concrete empiricism, and the engineer should strive to use the fruits of both.

The writer's answers to these questions involve a subject which is not even mentioned in the paper under discussion—the hydrodynamics of ideal fluids. Although the hypotheses of Newton lead to equations for motion of viscous fluids, the hydrodynamics of the eighteenth century was concerned exclusively with ideal fluids. The simplest experiments showed striking differences between the behavior of real fluids and the predicted behavior of ideal fluids. This was not unknown to the theorists; Hatschek, on page 6 of his book, "The Viscosity of Fluids," quotes Bernoulli: "I attribute these enormous differences for the greatest part to the adhesion of the water to the sides of the tube, which adhesion can certainly exert an incredible effect in cases of this kind." This was published in 1738.

Engineers could not wait for mathematical physicists to develop the complete theory, but while they developed empirical formulas they hoped that the theory would be developed. In 1804 Prony published the results of his researches; from his work Hatschek quotes: "It is regrettable, and even astonishing, that the celebrated Euler, who in the course of his immense labors has often turned his attention to physico-mathematical problems and their practical applications, should not have endeavored to treat the theory of liquids by taking into account the cohesion of the molecules and some kind of friction; even if he had introduced these resistances into the analysis in a purely hypothetical form, it would be interesting to know how he envisaged their effect; but I do not know of any paper by him where they are mentioned."

In 1813 Girard advanced a theory strikingly similar to some heard today. He assumed that there were two kinds of resistance: one, "the force with which all the molecules of the perimeter adhere to the liquid layer which envelops and wets the wall," is simply proportional to the velocity, and would be the only one if the wall were absolutely smooth; its asperities, however, are reproduced in the liquid film and give rise to turbulence, which produces a second resistance term containing the square of the velocity.

In 1823 Navier published the general hydrodynamic equations for viscous fluids. He did not integrate them correctly for flow through a cylindrical tube. This was done later by Stokes (1842).

Before the theoretical development of the formula for resistance in viscous flow was available, Hagen and Poiseuille had started their experimental attack on the problem. The latter's work especially was so elegant and complete that the problem of viscous flow was practically solved by the middle of the last century. Further theoretical investigations were made by Wiedemann (1856) and Hagenbach (1860), but from the time of Poiseuille the main emphasis has been on applications.

By this time we may imagine that the hydraulic engineers were not very hopeful of getting help from the physicists. They had the accumulated experience of more than a hundred years of practical work, whereas the physicists had only just discovered the laws of viscous flow. The viscous-flow laws were of no use to the hydraulic engineers, since they were not working in the viscous régime, and no one knew how far the viscous régime extended. This was the situation when Reynolds attacked the problem. Reynolds found the common limit of the two régimes, and offered a law for the turbulent régime of the form $f = aR^n$, in which f is the coefficient of friction, R the dimensionless group we call now Reynolds' number, and a and n constants; a depends on the dimensions, and for n Reynolds gave —0.277.

It is not surprising that engineers did not immediately abandon the formulas which they had used so long. It is not even surprising that most of them did not consider the work of Reynolds to have any application to their problems. As far as hydraulic engineers were concerned, Reynolds' work might have been forgotten. That it has finally gotten recognition from engineers concerned with the flow of fluids through pipes is probably due to the work of men engaged in a seemingly unrelated field—aviation.

The designers of aircraft had to know more about the resistance to relative motion of solids and fluids. After empiricism had brought aviation into existence, philosophy was called in. This was probably because if the factor of safety were too great the plane would not leave the ground, while if it were too small it would return too suddenly. At any event, reliance had to be placed on models. To use models intelligently required the principle of dynamic similarity, and the application of the principle of dynamic similarity to the problem of surface friction was suggested by Rayleigh in the report of the British Advisory Committee for Aeronautics for 1909–1910. And so it came about that the classic experiments of Stanton and Pannell were carried out, not in a hydraulic laboratory, but in the Aeronautical Department of the National Physical Laboratory of England, the object being "to confirm the existence, under certain conditions, of the similarity in motions of fluids of widely different viscosities and densities which has been predicted, and further, by extending the observations through a range in the velocity of flow which has not hitherto been attempted, to investigate the limits of accuracy of the generally accepted formulas used in calculations of surface friction."

The writer's excuse for the presentation of this discussion of the historical development is his feeling that something of the sort should appear in a paper with this title.

The remainder of this discussion is concerned with the deviations in the paper from the principles and results given in Stanton and Pannell's paper. In the first place, it should be pointed out that the use of the principle of dynamic similarity requires that certain assumptions be made, but those assumptions must be made on other grounds, and they remain assumptions even though limited experiments show that the relations derived with their aid describe the facts fairly well. The introduction of other assumptions must then be justified by experiment also. In this case the assumption is that the resistance depends only on the linear dimensions of the surface, and on the velocity, density, and viscosity of the fluid. From these assumptions may be derived the relations given in Equations [1] and [2] of the paper. (In consistent units, but as has been pointed out the units used in this paper are not consistent, in that pound is used as the unit of mass and force.) The use of the term diameter implies that geometrically similar surfaces are being considered, but the geometrical similarity must extend to those irregularities in the surfaces which constitute roughness.

Classification of pipes as to material and size is therefore equivalent to making the additional assumption that roughness is determined by material and size. Experiments to prove this are lacking, despite the curves presented in this paper. There is no reason to believe that steel pipe, for example, must be rougher

than brass pipe, nor that all steel pipes of the same internal diameter must have the same roughness. To accept curves such as shown in this paper is to deny the possibility of progress.

The writer believes that it would be far better to use the relation between Reynolds' number and the friction factor for smooth pipe as determined by Stanton and Pannell (and extended by Nikuradse, it appears, though the writer has not yet had the opportunity to study his results) as the basis for design, making such allowances for the difference between the smooth pipe and the actual pipe which must be used as experience and tests indicate may be necessary. There seems no reason to believe that commercial steel pipe cannot be made geometrically similar to the brass pipe used in Stanton and Pannell's tests; if designs are based on smooth pipe, the cost of roughness may be evaluated, and there will be a definite incentive to manufacturers to produce smooth pipe.

The few recent studies which have been made on pipe which is not smooth have indicated that for these also the general law connecting Reynolds' number and the friction factor given by Lees holds: $f = a + bR^n$.

If further tests should show that this law is generally applicable, it will require only a few tests to determine the relation for any particular kind of pipe. This has been pointed out by Parry, in the English *Electric Journal*, Vol. 1, No. 4, October, 1920, p. 146; Parry indeed suggests that for all pipes the curves for f against R should intersect the viscous-flow curve in a point, thus anticipating the notion of maximum friction factor brought out in this paper. However, there are indications that for very rough pipe the curve of f against R will rise after reaching a minimum.

The custom of plotting the curve of f against R on double logarithmic charts should be discouraged. On such a chart the curve is almost straight over a short range, and the temptation to make it straight is great. The curves for smooth pipe have been drawn as straight lines in this paper, despite the fact that one of the most important parts of the work of Stanton and Pannell was to prove that Reynolds' index law did not hold.

The introduction of the concept of mean hydraulic radius requires making an assumption which has not so far been proved. Geometric similarity must be discarded if the mean hydraulic radius is to be used. The use of the mean hydraulic radius is an attempt to combine size and shape, and it is quite certain that this can lead only to rough approximations. In cases in which mathematical solutions cannot be obtained, it is best to make model experiments.

The writer would point out that the results of the early experimenters are not necessary to prove the theory and that numerical values obtained by analysis of the results on all but smooth pipes must be used with a great deal of caution.

ALEXEY J. STEPANOFF.[43] One of the reasons why engineers are somewhat reluctant in adopting the modern method of calculation of the pipe flow, based on the scientific principle of dynamic similarity, is that the data available seemed scattered and unrelated. The work accomplished by the author is just as important as the experiments themselves. It is through investigations like this that the results of research find their way to practical application, because it is by the study and comparison of the data obtained by different experimenters that general conclusions can be drawn. It was gratifying to see that the scattered data on pipe flow were found to be consistent and fit into the picture outlined by great investigators. The division of the test results by the author into groups according to the pipe roughness seems very logical and greatly simplifies the problem.

The Reynolds discovery was not appreciated for more than thirty years, for the following reasons: First, at the time of discovery there were no real problems of pumping viscous liquids. Then, the knowledge about viscosity, its properties and measurement, was meager among practical engineers. This is true to a smaller degree even now. Further, the existence of the two modes of flow was known before Reynolds, but he showed the existence of critical velocity and established in an experimental way that both modes of flow were functions of what is now known as Reynolds' number.

Contrary to the author's statement, Reynolds did not test his equation of flow for dimensional homogeneity using an exponential formula with exponent varying from 1.75 to 2. It was not until Lord Rayleigh in 1911 and Stanton in 1914 took up fluid friction in the academic sense that Reynolds' discovery was presented in a form suitable for use by practical engineers.

One of the important things we learned about pipe flow is that all the exponential formulas are both theoretically and in practise wrong. According to the theory, the exponent for the turbulent flow should be 2. In practice it was found that different fractional exponents advocated from time to time cover only different portions of the function to be determined experimentally.

Another important result of the theoretical study of pipe flow is that the resistance of the flow is a continuous function of Reynolds' number only when the flow is dynamically similar, the geometrical similarity being one of the requirements. Hence it should not be expected that the data obtained with circular pipes would check with that for any other form of section, and since roughness does not follow geometrical similarity, a series of lines are obtained instead of one depending on the surface conditions of the pipe. On account of flatness of the curve representing function in Equation [2] for turbulent flow, the results obtained with other than circular sections do not differ much from the pipe data, while for viscous flow the difference is quite material. As a result of dynamic similarity, the velocity distribution across the pipe section is similar for the same values of the Reynolds number. The ratio of the mean to the maximum velocities varies from 0.5 for the viscous flow to over 0.81 at high Reynolds numbers. Anything that disturbs the velocity distribution destroys dynamic similarity, and the value of the function in Equation [2] will be different. This explains why the data obtained with curved, moving, or revolving pipes do not agree with the results at normal conditions. The geometrical similarity as to the length of pipe, especially the form of approach to the observed length in the case of laboratory experiments with comparatively short pipes, was found to have noticeable effect on the observed loss of head.

Having established the function represented by Equation [2] for circular pipes, it is possible by reversing the procedure to use the results for measurement of viscosity when all other means fail, as in the case of air-water mixture. By observing the friction loss for a given pipe, the value of Reynolds' number, and hence the kinematic viscosity, can be determined. The writer experimented some years ago with air-water mixture and found that the friction loss depends not only on the air-water ratio, but on the way air is distributed in water.

Most of the experiments on pipe flow were performed with commercially clean fluids. Presence of foreign matter like gas in oil, or sand and air in water, affects greatly the friction loss, the magnitude of the head lost depending on whether the sand or air is rolled along the pipe or carried in suspension. The friction loss in the first case is considerably greater than in the second. Some data on the sand transportation in pipes will be found in Trans. A.S.C.E., Vol. 57, p. 400, 1906.

From this review of the flow problem in its simplest form, with constant velocities and circular cross-section, one can see that calculation of the friction loss in pipes requires a great deal

[43] Designing Engineer, Byron Jackson Company, Berkeley, Calif. Mem. A.S.M.E.

of knowledge and judgment, and, we quote Gibson, "The engineer who predicts to within 10 per cent the discharge from a pipe under given pressure conditions has every reason to be satisfied with his choice of constants." Mention might be made of the extraordinary difficulties in the calculation of friction losses in hydraulic machinery—centrifugal pumps and hydraulic turbines; where the form of channels varies greatly from inlet to outlet, the velocities vary from section to section, and part of the water path is curved and revolving. In addition to friction losses, there are shock losses, and there is no way to measure even the combined loss. Thus separation of the hydraulic losses and evaluation of different items in a pump or turbine are more or less arbitrary.

The author's statement of the principle of dimensional homogeneity is not accurate. Quoting Buckingham,[6] the principle requires that "all the terms of any correct and complete physical equation must have the same dimension." "Complete and correct" in this statement implies that the subject to be investigated should be sufficiently known to establish what quantities should be included in the equation.

Any mistake in this respect would result in a wrong conclusion and the blame might be put on the dimensional analysis, while all the dimensional analysis does is that it enables one to reduce the form of the equation to the extent that the relationship between the quantities involved becomes more explicit. The equation tested for dimensional homogeneity may have several terms, like Bernoulli's equation, which may not be equal. The dimensional homogeneity of different terms only means that length may be compared with length, mass with mass, etc.

The author does not follow his own notation; the list of symbols gives m for the mean hydraulic radius (English notation), while r is used in the text.

The writer wonders whether the long list of the test data given by H. Hughes and A. Safford in their book on hydraulics, and references attached to Williams and Hazen tables, have been overlooked by the author. It is hardly possible that these data turned out to be unreliable. Also, some interesting test results on circular pipes and annular sections will be found in Mitteilungen des Instituts für Strömungsmaschinen der Technischen Hochschule Karlsruhe, heft 2, V.D.I. Verlag, Berlin, 1932.

The writer hopes that the author will keep his promise to append this paper with some more test data which may be made available since the issue of the preprint of this paper.

R. W. Angus.[44] This paper is valuable in calling attention again to the value of the Reynolds number in estimating the friction loss in pipes. The Reynolds number does not seem to have gained the place in practise that the theory would indicate it should occupy, and this is probably because it does not give all the information with regard to friction in pipes.

If the pipes are absolutely uniform as to their roughness, then the Reynolds number enables one to determine accurately the friction loss irrespective of the nature of the fluid flowing, but it is actually impossible to have pipes of different sizes of the same relative roughness, and this fact largely neutralizes the conclusions that one would naturally draw.

In looking at Fanning's table for the values of f, one is immediately aware of the fact that Reynolds' law has not been followed, and this is usually explained in the very simple way of saying that the pipes are of different relative roughness—a conclusion which may or may not be true.

The author has not included in his results information contained in a paper presented by F. Carnegie, Proc. Inst. M.E., 1930, Vol. 1, p. 473, in which a number of experiments have been given on the flow of steam in pipes 2 in., 6 in., and 8 in. in diameter, with steam velocities from 27 to 146 ft per sec. The steam pressures were in the neighborhood of 200 lb and of 66–88 lb, with about 100 F superheat in most cases. In this paper the Reynolds number and the corresponding friction factor have been determined by Mr. Carnegie, the coefficient of the viscosity of the steam being taken from a paper by Speyerer and reported in the *Z.V.D.I.*, 1925, Vol. 69, p. 747.

It would be helpful, in connection with this important subject, if tables of viscosity of different fluids were available in convenient form.

K. M. Peters.[45] The writer would like to offer some data on the mean hydraulic radius as a criterion for flow in the turbulent region. Using the mean hydraulic radius as a basis of comparison, the curves in Figs. 18 and 19, were recalculated from published

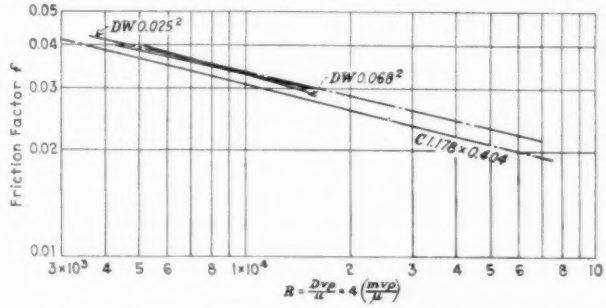


Fig. 18 Flow in Rectangular Sections

(Legend: C = Cornish; DW = Davies and White. The broken-line upper and lower curves are for smooth circular pipe.)

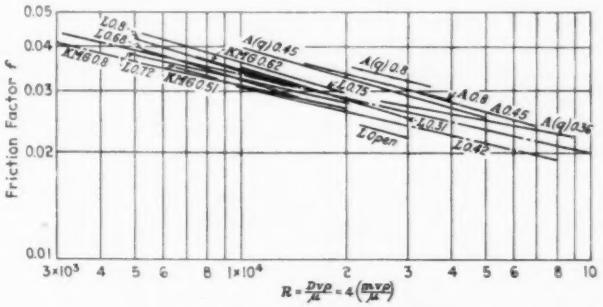


Fig. 19 Flow in Annular Sections

(Legend: A = Atherton; q = air used as fluid; L = Lonsdale; KMG = Kratz, Macintire, and Gould. The numbers give the ratio of core diameter to shell diameter. The broken-line upper and lower curves are for smooth circular pipe.)

data taken from the references given in the summary of tests herewith. In order to make a direct comparison with Mr. Kemler's results, the equivalent diameter was used in these curves, which have been marked similar to his.

The tests on rectangular pipes indicate that the friction factor is the same as that obtained from tests on circular pipes. This could be anticipated from the fact that the dimensions were varied and the same surfaces were kept in contact with the fluid. Because of the similarity in shape, the roughness effect should compare in magnitude to that for a circular pipe.

Annular spaces offer quite a wide variation in friction factor. Some noteworthy conclusions can be drawn from this data, however. Tests by Lonsdale marked open, 0.31, and 0.75; 0.42 and 0.68; 0.72 and 0.8 were made with 2.520 cm, 1.895 cm, and 3.972 cm shell diameters, respectively. The open pipe

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[45] Senior in Mechanical Engineering, University of Pittsburgh, Pittsburgh, Pa.

shows a much lower friction factor than the shell using a small core, which in turn shows a lower value than that for a large core. A similar relation exists for the other two pipe sizes tested by Lonsdale. The reason for this is that the relative roughness is increasing as the core size increases and produces a higher f. Kratz's experiments follow this tendency, with the exception of tests using a large core, which give lower values. The condition of the outside of the core could, however, account for this apparent discrepancy. Atherton's tests correspond to Lonsdale's, but his values cannot be given much weight because of the inconsistency of his data, as can be seen by an examination of his paper.

These data show that the friction factor for open pipes is very near the value for circular pipes. For annular spaces the friction factor will be appreciably higher; the magnitude of this increase depends upon the ratio of core diameter to shell diameter and on the roughness of the pipe itself.

Summary of tests on annular and rectangular brass pipes (water is the fluid used, except that the Lonsdale tests were with air and water and the Kratz tests with water and $CaCl_2$): Annular: "Fluid Flow in Pipes of Annular Cross-Section," by Atherton, Trans. A.S.M.E., Vol. 48 (1926), p. 145. "Flow of Water in Annular Spaces," by Lonsdale, *Phil. Mag.*, Vol. 46, p. 163. "Flow of Liquids in Annular Cross-Sections," by Kratz, MacIntire, and Gould, Univ. of Ill. Eng. Expt. Bul. No. 222. Rectangular: "Flow of Water in Rectangular Sections," by Cornish, Proc. Roy. Soc., Vol. 120, p. 691. "Flow of Water in Rectangular Pipes," by Davies and White, Proc. Roy. Soc., Vol. 119-A (1928), p. 92.

Adolf A. Meyer.[46] The writer was interested in comparing the information presented by the author with the established formulas for computing flow in pipes.

In Fig. 20 the author's relation curves between Reynolds' number and friction factor f for brass pipes are replotted. In addition, similar curves are added based on computed values according to the formulas of Bazin, Manning, and Kutter. Whether these formulas should be applied for flows in pipes is not questioned here, the interest being based on the fact that they have been used and are still being used extensively for the solution of problems of flow in pipes. Coefficients of friction n for Kutter and Manning and m for Bazin for smooth new pipes, corresponding to the condition of brass pipes, have been used.

The resulting curves computed after Bazin and Manning are straight lines as expected, as in both these formulas the loss of head, from Chézy's fundamental relation, varies with the square of the velocity. As f is expressed as a function of $\Delta h/V^2$, there remain for a given diameter only the constants which plot as a straight line. According to Kutter's formula, the shape of the curve is not a straight line. For the small diameter pipes plotted in Fig. 20, the curve drops within the range of the smaller Reynolds numbers similar to the results given by the author for brass pipes, but with increasing slopes they show a tendency to extend as a straight horizontal line. The difference between the curves according to Bazin and Manning on one hand and Kutter on the other hand is due to the fact that in Kutter's formula the coefficient c also is a function of the slope.

[46] Pennsylvania Water and Power Company, Baltimore, Md.

It is interesting to note that for varying diameters of pipes the friction factor varies. If we refer to the author's Fig. 4, we notice a similarity, as the small-diameter pipes show higher friction factors than the larger pipes for identical Reynolds numbers. Evidently from lack of data the author did not carry out this differentiation of pipe sizes in the other diagrams presented.

Looking over the successive diagrams, it will be noticed that with increasing roughness more and more groups of tests made with the same diameter pipe approach the horizontal straight-line curve, or, in other words, show closely the same relation between the loss of head and velocity as do the established formulas. The general picture of all the data presented still apparently shows the downward trend, but the question raised

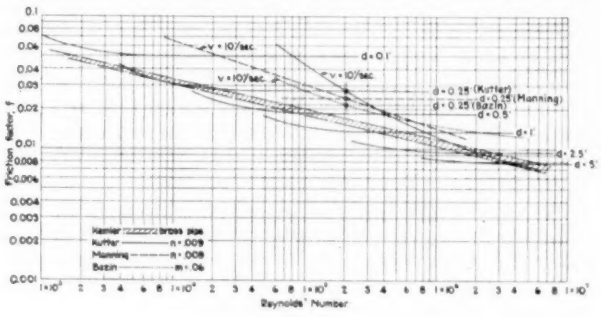


Fig. 20

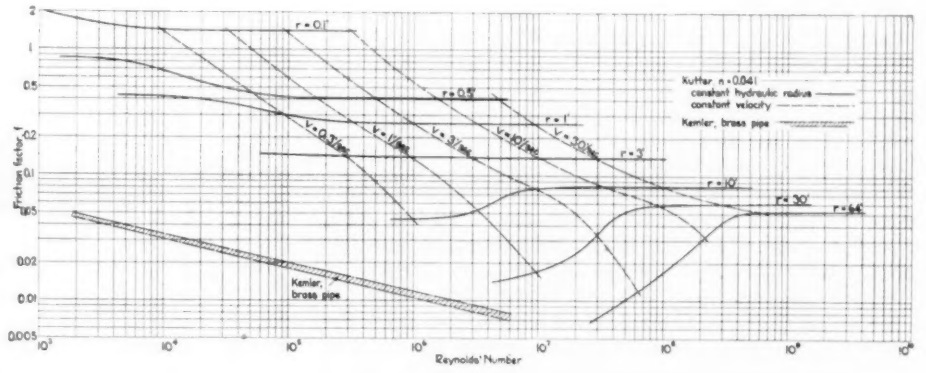


Fig. 21

is whether this trend is due to actual decrease of the friction factor or due to the accidental fact that with increasing Reynolds numbers more and more tests plotted were made on larger pipe sizes to correspond with the high velocities required for the higher Reynolds numbers. When plotting curves of equal velocities for pipes with varying diameters, the downward trend of the velocity curve is clearly apparent in Fig. 20.

What aroused the interest of the writer in the author's paper was above all its application to backwater computations. If for turbulent flow the friction factor would decrease under all conditions with increasing Reynolds numbers, then backwater computations based on Kutter's formula would obviously be wrong.

To present this problem more clearly, Fig. 21 was drawn up. The relations between Reynolds' numbers and the friction factor f computed according to Kutter's formula were plotted for different hydraulic radii between $r = 0.1$ and $r = 64$ and for an $n = 0.041$, which factor was taken from an average figure found in a river on which the writer had occasion to investigate backwater.

Let us assume that during a season of high flow a stretch of

river with a mean $r = 64$ and a mean velocity of 30 ft per sec is observed. The Reynolds number for this flow would be 6×10^6, the friction factor $f = 0.051$. Suppose that by some obstruction downstream the water surface during low flow would be raised to such an elevation as to give the river stretch the same mean $r = 64$ with the velocity reduced to 3 ft per sec. With the Reynolds numbers now ten times smaller, we should expect, according to the data as presented by the author, a friction factor larger than 0.051. According to Kutter, the friction factor is reduced to 0.011.

We have had the opportunity to check computed backwater in the field after ponding took place. The actual observation confirmed the values computed in accordance with Kutter's formula within a very close range. Reference is here made to an article by W. D. Myers on "Comparison of Computed Backwater Profile With the Actual Observed Profile," published in the 1929 report of the Hydraulic Power Committee of the Pennsylvania Electric Association.

Referring to Figs. 13 and 14 of the author's paper, "Flow in Artificially Roughened Pipes," we notice there a few tests which show an upward trend, or, in other words, show the loss of head varying to a power greater than 2.

The suggestion is offered that the information presented by the author be rearranged in a three-dimensional system by plotting Reynolds numbers in one direction, pipe diameters or velocities in another, and the corresponding friction factors in a third. This would show a topographic picture of the friction factor from which the interrelation of the three factors involved could be more clearly understood. With the information as now represented, a warning might be placed not to depend uncritically on the decrease of the friction factor for increasing Reynolds numbers when using the same size pipe.

A. B. Cox.[47] The proposal to establish upper and lower limits for the value of the friction factor, as suggested in the paper, should be a valuable contribution both to theory and practise. The author's curves do not show the points plotted which determined them. Curves with data points plotted are nearly always much more preferable to those without them. This is all the more true here, because some of the methods used by some experimenters for determining the critical flow limits are questionable.

L. Schiller.[48] This collection of voluminous data, particularly on the flow in rough pipes, is especially valuable to us in Europe because it contains a large number of American experimental results which are not generally available to us here. The value $f = 0.054$, for the pipe diameters customarily used and neglecting pipes of extreme roughness, appears to indicate a definite upper limiting value. One must be absolutely clear about the fact—and this is not definitely stated in the paper—that for equal "absolute roughnesses" a further diminution of the diameter will result in higher f values.

As regards the relation of the critical f value, $f = 0.054$ to the "absolute critical number 1160," the following should be pointed out. According to the designation of the author, $R = Dv\rho/\mu$, the critical number is not 1160, but 2320. To this value belongs $f = 0.038$, not 0.54. With regard to the resistance law for *smooth* tubes one may say today that it has been confirmed to a value of $R = 2 \times 10^6$ within 1 per cent, by the recent experiments at Leipzig and Göttingen (Hermann, Nikuradse)[49] which check each other. The resistance law as expressed by Hermann is $f = 0.00540 + 0.396R^{-0.300}$

As to the question of *rough* pipes, there is lacking any experimental proof of the law of similarity. A rigid proof may be made most easily with relatively coarse systematically arranged roughnesses. We in Leipzig have carried out experiments on two tubes of 5 and 10 cm diameter with regularly spaced reducing flanges built into them. In these experiments complete similarity has been maintained as far as it may be attained practically (test to 0.03 mm). The similarity law has been confirmed to within 2 to 3 per cent.

Author's Closure

The comments of Mr. L. Schiller are very interesting, as they give the European opinions on this subject. His comments relative to the experimental proof of the law of similarity as applied to rough pipes are most welcome. The fact that complete similarity has been maintained in so far as it may be attained practically gives the necessary assurance that tests on rough pipes made with one fluid may be applied to another fluid. While this law has been practically applied, this is apparently the first series of tests to definitely establish that the law held.

The question raised by Mr. Baumeister is the final result in which the practicing engineer is interested. Until it is better understood just how roughness affects pipe flow, it will be impossible to make any rigid statements as to the values that should be used. The coefficients such as Mr. Baumeister gives have to be used with care, since it amounts to saying the roughness has the same effect at all Reynolds numbers. As long as there is a viscous film on the pipe, the effect of roughness will necessarily be less than when this film disappears. Since the viscous film varies with the Reynolds number, curves of the type such as Mr. Pigott gives in his paper seem to be a reasonable solution. Fig. 22 and Table 5, which are taken from the data in Fig. 1 and Table IV, respectively, of Mr. Pigott's paper, illustrate the type of curves just referred to. These curves which Mr. Pigott gives are the best information available at the present time on friction factor for rough pipes.

The question as to whether the friction factor depends upon Reynolds' number only, and is not influenced by the diameter (assuming the roughness effect the same) or velocity, as brought out by Mr. Spitzglass and Mr. Meyer, is an important consideration. The results on brass pipes for different fluids and pipe

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[48] Division of Applied Mechanics and Thermodynamics, Physics Institute of the Leipzig University, Leipzig, Germany.

[49] L. Schiller, "Strömung in Röhren," Handbuch der Experimental-Physic, Band IV, 4, 1932.

TABLE 5

(Dimensions in inches)

	Curve No. 1	2	3	4	5	6	7	8	9	10	11	12	13	14
Steel and wrought iron	Over 72	48-66	42-14	12-6	5-4	3-2	1½	1½-1	¾	½	⅜	¼	⅛	..
Galvanized	...	...	30	24-10	8-6	5-3	2½	2-1½	1¼	1	¾	½	..	..
Best cast iron, Cement, Light riveted, Sheet ducts	...	...	96-48	48-20	16-12	10-5	4-3	2½-2	1½	1¼	1	..	..	..
Average cast iron, Rough formed, Concrete	...	...	Over 96	96-42	36-24	20-10	8-6	5-4	3	...	..	..	..	..
First-class brick, Heavy riveted	...	...	Over 220	204-84	72-48	42-20	18-16	14-10	8		4	..	..	..

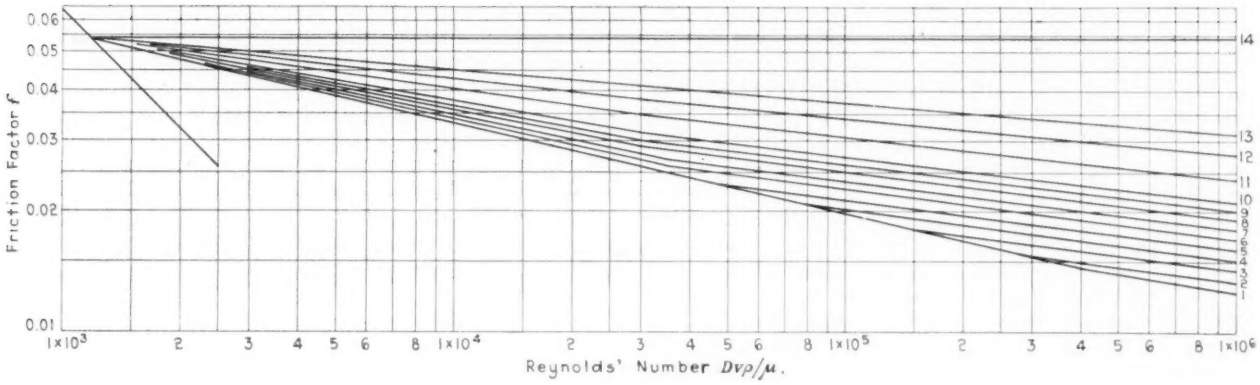


FIG. 22

sizes indicate that the friction factor depends only upon the Reynolds number for smooth pipes. The suggestion of Mr. Meyer that a three-dimensional plot be made is very interesting, but to obtain reliable results a series of tests with this as an objective would have to be made, otherwise variations in roughness and viscosity might influence the results.

Mr. Stepanoff and Mr. Pigott have brought out the point that the velocity distribution is an important factor. It seems to the author that a study on rough pipes which would give data on the velocity distribution, friction factor, and Reynolds' number would add much to our understanding of the behavior of rough pipes. A microscope study of the pipe surfaces should be made to see how it influences the velocity distribution. Such a study would have to be very complete in order to give conclusive results.

The author recognizes that the field of viscous flow has only been touched upon in the paper, but the limitation that had to be placed on this research problem and the fact that the viscous law has been generally accepted are the reasons for its omission. Most of the emphasis was placed on turbulent flow, since it has the larger field of application and is the field about which we have a limited knowledge. The comments of Prof. Glen Cox, therefore, add welcome information in this field.

The comments of Mr. A. C. Chick concerning Mr. Freeman's results are of considerable interest. An examination of the summarizing curves for steel pipes will show that Mr. Freeman's test results fall almost on the average curve for each case. His tests cover the entire range of pipe sizes and were apparently made with exceptional care. The making available of the complete data should be a valuable contribution to the literature.

The historical development of hydraulic flow equations as given by Mr. Spitzglass in his paper, "Similarity: Limitations in Its Applications to Fluid Flow,[50] together with that given by Mr. Miller, gives a brief outline of the historical background of this paper.

Mr. Miller's criticism of the use of various curves for different size steel pipes is misleading. There is reason to believe that ordinary steel pipe is rougher than brass pipe because of the difference in method of manufacture. There is no better proof of this fact than the test results given in the paper, which show this to be so, and in the final analysis there is no other proof possible. The author would agree to a statement that it is possible to make steel tubing as smooth as brass pipe, but it remains that it is not usually made that smooth. While there is a definite saving possible in using smooth pipes because of the lower friction losses, the pipe usually cannot be kept in a smooth condition when subjected to abrasive or corrosive action. Under such conditions the pipes soon become rough, and the extra price that must be paid for smooth tubes cannot be justified on a saving in power costs alone.

It is true, as Mr. Rhodes brings out, that the making of all the curves to pass through the value of Reynolds' number equal to 1160 has no proof. However, with the qualifications placed upon it there seems to be some justification for using it. These qualifications are that the actual diameter be used and that the roughness be of an order of magnitude such as not to cause contraction and enlargement losses. A study of rough pipes, as outlined before, would do much to clear up this theory.

The curves of Mr. Peters give additional data on the mean hydraulic radius as a criterion for flow in the turbulent region.

The suggestion of Mr. A. B. Cox that the points for the curves should be plotted is an impossibility, as this would have required about 300 figures to show accurately the data for all of the curves.

There has been some confusion concerning the value $R = 1160$ as given by Schiller. The value as obtained by Schiller was 580 on the basis of using the radius in Reynolds' number. The value of Reynolds' number based on the diameter would then be 1160, or the value used in this paper. In all cases the use of the term Reynolds' number in this paper means $Dv\rho/\mu$.

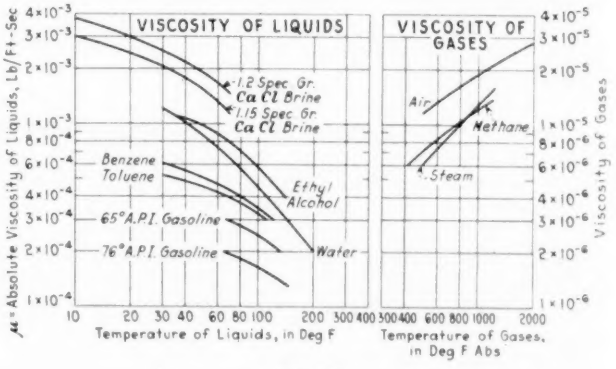


FIG. 23

Curves of viscosity as suggested by Mr. Angus are given in Fig. 23. The data on viscosity were taken from the Handbook of Chemistry and Physics, the Smithsonian Physical Tables, and *The Petroleum Engineer*, November, 1932, p. 26. The data given by Mr. Angus and Mr. Van der Pyl have not been replotted for reasons of economy. The references referred to by Mr. Stepanoff were not used because the complete test data were not available.

In conclusion, the author wishes to express his appreciation for those who have added their experiences and comments. It is hoped that this paper will indicate the fields of research needed to increase our knowledge of turbulent flow.

[50] Trans. A.S.M.E., vol. 52, 1930, paper HYD-52-7c, pp. 111–121.

Research Institute for Hydraulic Engineering and Water Power

Discussion of Paper by Hunter Rouse, Cambridge, Mass. [Trans. A.S.M.E. (1932), HYD-54-3], Describing Large Open-Air Hydraulic Experiment Station Near Munich, Germany

C. M. ALLEN.[1] Inasmuch as the paper entitled "Research Institute for Hydraulic Engineering and Water Power," by Hunter Rouse, was published without being presented at a meeting, the author did not have the benefit of the usual criticism and comment that are a valued feature of oral discussion. Hence it is necessary to call attention to certain statements concerning the salt-velocity method of water measurement appearing in the paper that should be cleared up.

The last two paragraphs of the paper practically say that the current meter is by far the best method of water measurement. This conclusion is so contrary to the data set forth in the paper, is so contrary to American practice, and the whole matter is so important to manufacturers and engineers all over the world that it should not go unchallenged.

In the Walchensee laboratory tests the results of four different methods of water measurement were compared with those obtained by the volumetric method. The measuring basin, the water-level gages, and the test apparatus used in the volumetric measurement appear to be correct in theory and practice and no doubt they were properly installed and carefully observed during the test. The values for discharge by this method should be within a small fraction of 1 per cent of the true values.

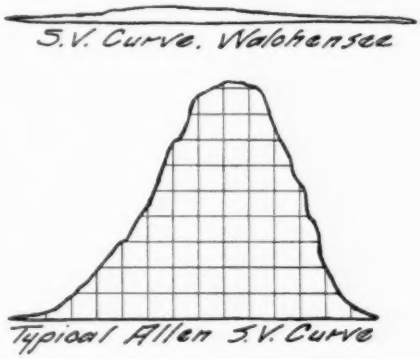
The four methods which were compared with the volumetric are the weir, current meter, salt velocity, and salt titration. I assume that the laboratory personnel was familiar with the measurement of discharge by weir and by current meter, and in the absence of any knowledge of the matter, by salt titration.

The author briefly describes the Allen salt-velocity method as conducted at this laboratory, when as a matter of fact, basing our opinion upon the description and results as given, it appears to be a long way therefrom. The detailed description of how these tests were made is so incomplete that it is difficult to offer much helpful comment. For instance, we should like to know the size and type of injection valves, whether a-c or d-c current was used, what voltage was used, and how the height of the salt curves was intensified. More information about the electrodes would be in order and a sample salt curve would be very desirable.

Apparently the personnel of this new laboratory was attempting a few of the procedures that were tried out over ten years ago in the Alden Hydraulic Laboratory, most of which were discarded because of errors,[2] and had no knowledge of developments since 1923 and consequently did not use the accepted standard methods. It is not surprising that, as the author states, they did not know the proper location of the electrodes. That their very first test showed results within 1.2 per cent of the volumetric results speaks very well for the salt-velocity method in the hands of inexperienced men.

The author states that the purpose of these tests was to compare the leading European and American methods of measuring penstock discharge, and that "these experiments were made by recognized authorities in the field of practice." We cannot agree with this statement as it applies to the salt-velocity method.

These pipe-line tests were made prior to the laboratory tests, and, so far as we know, no one connected with them had had any practical experience with the salt-velocity method. Mr. Howland, of the British Pitometer Company, Ltd., London, made both the pitometer and salt-velocity tests. His experience with the salt-velocity method was confined to a few hours' visit to the Alden Laboratory. It was understood that he would be supplied with special instructions before a test was made. Instructions were cabled, but they were not followed. If Mr. Howland had been able to do as instructed in the one cablegram sent, he would have gotten even better results than he did. If he could have had full instructions and an opportunity to carry them out, he would have made an exceptionally accurate test, because the conditions at Walchensee were ideal for a perfect test.



FIGS. 1 AND 2 SALT-VELOCITY CURVES

In the conduct of the test only one pop valve was used for salt introduction, and that was located in the center of the pipe at the lower end of a bend where the distribution of instantaneous velocities was very bad for a pipe line. Standard practice is to install two to four pop valves at the upper end of the bend or, preferably, well downstream in a straight section of pipe. The Walchensee penstocks were about 1200 ft long and afforded ample room for an ideal test section. No advantage was taken of the total length of penstock.

Four electrode stations were used. Standard practice would have used only one. The two upper stations at Walchensee were improperly located, the electrodes were of unsuitable design, i.e., without making complete traverses, and the results were rejected for obvious reasons. These two sets of electrodes never should have been installed, but the runs in which they were used constituted more than half of those made.

The third electrode was so close to the fourth and final one that the two salt curves overlapped. A correction was made for this,

[1] Professor of Hydraulic Engineering, Worcester Polytechnic Institute, Worcester, Mass. Mem. A.S.M.E.

[2] See "The Salt-Velocity Method of Water Measurement," by C. M. Allen and Edwin A. Taylor, presented at the A.S.M.E. Annual Meeting in December, 1923, and published in MECHANICAL ENGINEERING, January, 1924, p. 13.

but the third electrode was unnecessary and should never have been installed. For research purposes duplicate sets of electrodes are sometimes installed, but they are always located far enough apart to prevent interference of the salt curves.

The results from the two lower electrode stations were computed and appear in Table 3 of the paper. Direct current was used, while only alternating current is recommended. Owing to the use of direct current, voltage regulation by resistance, long-line resistance in the electrode leads, and to the tiny electrodes used, the salt curves from the Walchensee instrument were very difficult to compute accurately. The accompanying Fig. 1 shows a sample salt curve from the Walchensee test and Fig. 2 one from a typical salt-velocity test properly conducted. The Walchensee curves could and should have been five times as high as they were. The diameters of the pipe and the lengths of the test sections which produced these two curves were about the same in each case.

In the next to the last paragraph of the paper the author discusses the "poor results" obtained at Walchensee by the salt-velocity method and the "very dependable results" by current meter. The German Research Institute makes the current meter standard for accuracy of water measurement. Obviously the author has never computed the actual percentage of variation in values for Q at Walchensee by current meter and by salt velocity.

Referring to Table 3 of the paper and assuming the current-meter results as correct, then the results by salt velocity vary from current-meter values as follows: Experiment No. 7, — 0.25 per cent; No. 8, + 1½ per cent; No. 9, + 0.033 per cent; No. 10, — 0.2 per cent. Three out of four runs show a variation of less than 0.25 per cent, and obviously something is

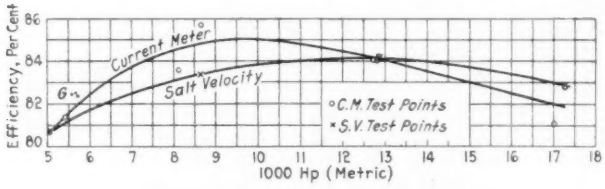


FIG. 3 WALCHENSEE TESTS, 1929

wrong with the values given for the other run, No. 8. While the values for Q by current meter and salt velocity vary by 1½ per cent, the corresponding values for efficiency vary by 2¾ per cent. This is impossible.

If the author will plot all the values for Q and efficiency against either gate opening or horsepower for the two methods, he will conclude that the salt-velocity values look more reasonable for experiment No. 8. The current-meter value for efficiency of 85.66 per cent looks wild for this run.

Fig. 3 is a curve sheet showing efficiency on horsepower plotted from Table 3 in the paper but omitting the pitometer results. It is Fig. 40 of the paper with the actual test points shown. In making up Table 3 the salt-velocity values for runs Nos. 1 to 5, inclusive, were rejected by the author because the wrong electrodes were used. Run No. 6 was rejected on account of errors in load, and runs Nos. 11 and 12 were for the Gibson method only.

Which shows greater variation, the nine current-meter points or the four salt-velocity points? The current-meter point at 8664 hp, or run No. 8, is under suspicion. If that value and one low value at full load are ignored, then the current-meter curve and the salt-velocity curve would practically coincide. So much for "poor results" by the salt-velocity method and "very dependable results" by current meter.

The author states that "the salt-velocity method showed excessive departure from its own average of readings except on the stretches from the injection point to the third and fourth electrodes." Of course, they would with the salt introduction made with one pop valve at the center of the pipe and at the lower end of a bend and with small traversing electrodes (2 in. long × 1¼ in. wide) but with no traverses made. The plant operator could not arrange for making complete traverses before the regular test.

One set of standard electrodes should have been used. The first and second electrodes should never have been installed. Electrode No. 3 was actually an injury to the test, since it was so close to No. 4 that the two salt curves overlapped. It was unnecessary for any field testing, but if installed for research purposes, then it should have been located 100 ft upstream or, better yet, located where it was and electrode No. 4 moved from 100 to 400 ft downstream. All of these things would never have occurred in a properly conducted test, and this discussion would probably have been unnecessary.

In 1930, Dr. Otto Kirschmer, head of the Research Station for Water Power and Hydraulic Design, published the results of these Walchensee tests. At that time he was at the head of this German Research Institute. The author's Table 3 is a copy of Dr. Kirschmer's table of results. The following is quoted from the original paper: "Fig. 19 shows further that the salt-velocity method does not vary very much from the meter method." Dr. Kirschmer also stated that the salt-velocity method would have given better results had the *standard type* of electrode been used. In view of both of these statements, how has the author arrived at his conclusions in regard to the salt-velocity method?

It should be emphasized that, in common with all other methods of water measurement, only those who are qualified and who have had experience with a method should use it. If others desire to use it or experiment with it, they should acquaint themselves with the latest developments and should not publish their results unless all the conditions and facts are fully described.

A. W. K. BILLINGS.[3] The tests under discussion were initiated in connection with the efficiency guarantees in a contract with a European manufacturer for an impulse wheel of 70,000 hp. maximum capacity for a plant near São Paulo, Brazil.

For many years in connection with plants in Spain, Mexico, and Brazil we have had difficulties with the various methods of specifying and of testing efficiency; these were particularly evident in comparing guarantees of European and American manufacturers. As a result of this experience and after careful study we have adopted and now insist upon the salt-velocity method of testing, and, for low-head plants, alternatively the Gibson method.

The purchaser, in continuing this policy, therefore required the German manufacturer to satisfy himself of the reliability of the method before the informal award for the unit in question would be confirmed by a written contract. The manufacturer arranged with the Research Institute to do this work, and the scope of the tests was extended by it to include the pitometer and salt-titration methods.

The Walchensee plant has an important load, and the management was naturally somewhat reluctant to shut down the units for the periods desired and to prepare the penstocks to receive the electrodes. These limitations and the natural desire of the manufacturer to expedite the comparison resulted in these first tests being made without attention being given to certain details in which changes were indicated by Professor Allen. The results, however, satisfied the water-wheel manufacturer, and

[3] Vice-President, Brazilian Traction, Light, and Power Co., Ltd., Toronto, Canada. Mem. A.S.M.E.

the formal contract was made on the basis of the salt-velocity method.

It was not expected initially that these tests would be published, and the special motive and circumstances should be borne in mind in discussing the results.

In our work we have found the current-meter method laborious under the very best conditions, even though accurate in expert hands, while in turbulent flow and with entrance conditions far from ideal, as in many commercial plants with large units, it is subject to large errors and difficult of interpretation. We have found that both the salt-velocity and Gibson methods check satisfactorily with each other, and, where possible, with volumetric measurement; and we now decline to use, for official acceptance of large units under practical conditions, others such as the current-meter, weir, or titration methods.

It is our opinion that European engineers, while conservative and reluctant to adopt these new methods, will gradually come to accept them, especially for testing large units where the weir and titration methods are impracticable and the current-meter method leads usually to controversy regarding the characteristics and accuracy of the various types of meter selected.

Edward S. Cole.[4] This paper gives an excellent description of the Walchensee Hydraulic Laboratory and of certain flow measurements made in the pipe line of a nearby power plant which it was the writer's privilege to witness; but with several of the statements and conclusions concerning the pitometer tests he must disagree.

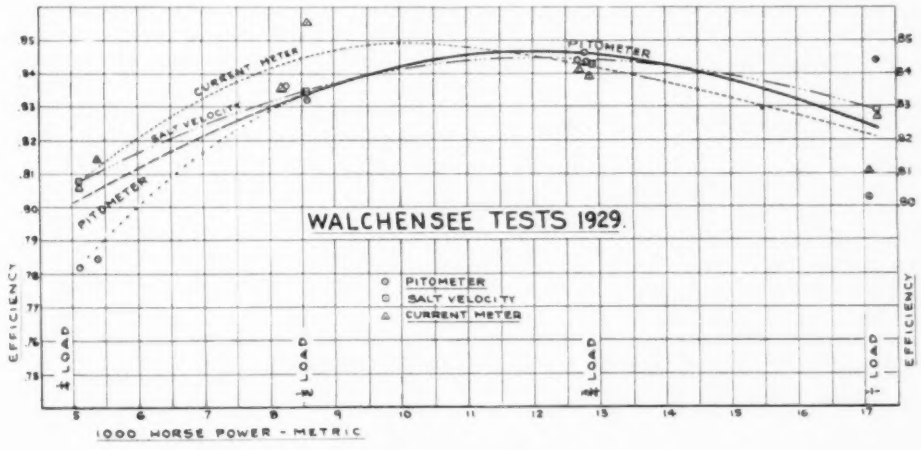


Fig. 4 Walchensee Tests, 1929

The accuracy of a pitometer gaging depends upon four things: (1) Calibration of the instrument; (2) ability to read the axial component of angular flows; (3) selection of a suitable gaging point; and (4) experienced handling by the engineer conducting the tests.

These pitometer measurements at Walchensee were unofficial but were made by The British Pitometer Co., Ltd., for its own information. The instrument had been calibrated by a well-known hydraulic laboratory with a high degree of accuracy, except at the lowest rate of flow where the laboratory now admits a probable error of 2 or 3 per cent.

The gaging point was quickly shown to be unsuitable by the first velocity traverses made, but local authorities would not permit a change. A vertical bend about 100 ft above the gaging point caused a disturbed flow, and it would have been easy to secure a longer tangent by moving it farther down the pipe line.

The author makes reference to unstable U-tube readings, but these were probably due to the gaging point with its disturbed flow. He speaks of "supporting rods" for the pitometer and current meters, but no supports are needed by the pitometer, which is introduced through a valve attached to the penstock wall. This may be done without unwatering the line if necessary.

The pitometer gives velocity readings within $^1/_4$ in. of the pipe wall, a decided advantage in computing the velocity traverse, since current meters can hardly read closer than 2 in. from the wall, involving uncertainty as to this outer 2-in. ring, which has about 12 per cent of the total pipe area and is the region of greatest velocity change.

The author makes a claim as to the relative speed of taking pitometer readings, but in the writer's opinion speed is largely a matter of organization. The pitometer responds to changes of velocity quite as quickly as the current meter.

In the original report of Dr. Kirschmer the statement is made that "Pitot measurements should give higher velocities at low flows and lower velocities at high flows than current meters." This evidently is based upon the assumption that the "pitot" has a uniform coefficient throughout its range, which is certainly not true of the pitometer.

Now as to the pitometer results (see Fig. 4):

At one-quarter load the two pitometer tests agree closely with each other, but are about 2 per cent below the salt-velocity and current-meter values. This is accounted for by the laboratory error at that rate of flow, as noted above.

At one-half load there were two pitometer tests which came within $^1/_2$ per cent of each other, agreeing closely with the one salt-velocity test and with the lower of the two current-meter values, the upper of which, by the way, seems to be about 2 per cent too high.

At three-quarters load the three pitometer tests agreed with each other and the single salt-velocity test within less than $^1/_2$ per cent. The two current-meter values were also very close, and certainly this agreement of the three different methods of measurement is most interesting.

At full load the average of the two pitometer tests agrees closely with the average of the two current-meter values and with the single salt-velocity test, but the variation in pitometer tests at full load probably was due to the disturbance of the upstream bend.

A considerable number of pitometer gagings made in waterpower penstocks of American and Canadian plants have demonstrated the extreme simplicity and high degree of accuracy of the method.

Marcus Mousson.[5] The tests at the Walchensee development are the first in history where so many practical testing procedures of European and American origin have been compared. The physical layout of the plant properties is equally favorable for all methods used, which is seldom the case at other hydroelectric developments. In spite of this the results obtained

[4] Hydraulic Engineer; President, Pitometer Co., New York, N. Y. Mem. A.S.M.E.

[5] Hydraulic Engineer, Safe Harbor Water Power Corp., Baltimore, Md.

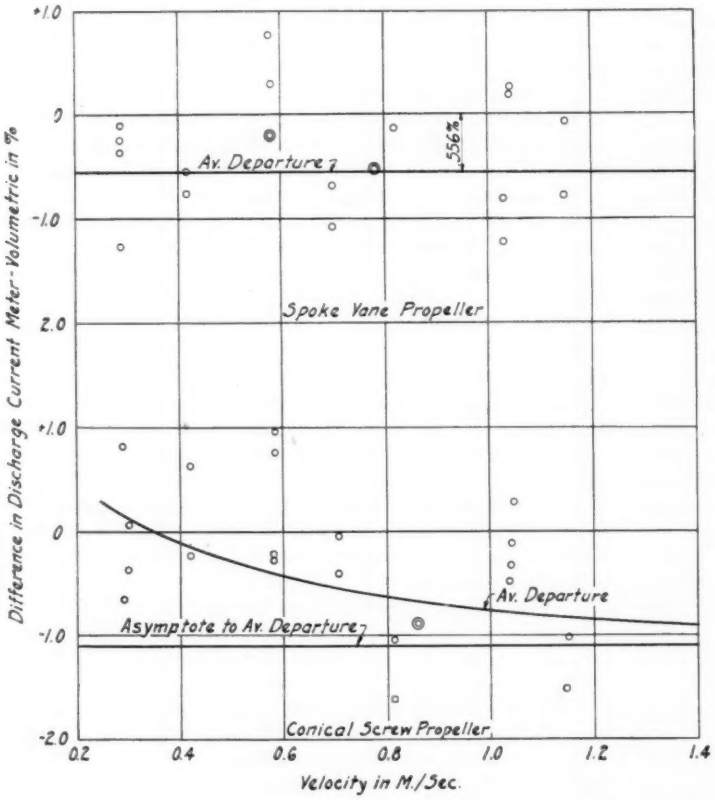


FIG. 5 AVERAGE DEPARTURE CURVES OF SPOKE-VANE AND CONICAL-SCREW METERS FOR WALCHENSEE FLUME TESTS

and the conclusions drawn therefrom need a more careful analysis to avoid being misleading.

Unfortunately, some European engineers have concluded from the results that the Allen salt-velocity, Gibson, and pitot-tube methods will be permanently abandoned in the near future. However, it must be said that in America we see no weakening of faith in these methods where conditions are suitable, but rather a development of current-meter application for conditions where satisfactory tests by means of other methods do not appear to be possible.

Because of discrepancies that have occurred in the past between various testing methods, some prejudice may have sprung up against one or another method. It has been too often specified in turbine contracts, and has frequently been standard practice, that the acceptance tests should be made by means of two methods; though the physical layout of the water passage may have not been favorable for the application of one of them.

There are enough data at hand to substantiate the fact that American and European methods give identical as well as reliable results when the layout is favorable to the same degree for the various methods. Much harm may be done to any method when the physical conditions are unsuitable or if the test personnel should lack in necessary familiarity with the methods and the difficulties which may be encountered.

It would be very desirable if all essential precautions were to be taken before actual construction of a hydroelectric development to guarantee a water passage suitable to the method by means of which the turbine acceptance tests will be made. Assumptions that hydraulic conditions will be satisfactory alone are not always justifiable, as results of model tests may show that alterations in design are desirable and essential.

Enough experience in current-meter and salt-titration procedures was at hand at the Walchensee to insure reliable results with these methods, but it is hard to believe that the experimenters had sufficient knowledge and practice with the Allen salt-velocity and Gibson methods to obtain satisfactory results, as extensive experience is essential to the successful application of these typically American methods.

Some comparative measurements have been made at the hydraulic laboratory of the Pennsylvania Water and Power Company at Holtwood. When the laboratory was put in service, the venturi meter measuring the discharge was calibrated by means of the Allen salt-velocity method. Conditions suitable to this particular method are provided by a straight penstock section of sufficient length. Before the turbine tests at Safe Harbor, two current meters were tried out in the flume of the above-mentioned laboratory and a very close check was arrived at between discharges measured by the venturi meter, calibrated by the Allen method and the current meters. The average discrepancy established was less than 0.10 per cent.

A careful analysis of the current-meter measurements presented in the paper produces some valuable new data and permits fundamental conclusions to be drawn regarding the application of current meters.

In Fig. 5 are plotted departures of discharges measured by means of current meters from volumetrically determined quantities in per cent versus mean velocities at the current-meter measuring section for the flume tests (runs 1 to 28, inclusive). It is apparent that the spreading of the test points is quite large and that the distribution is not alike for both meters

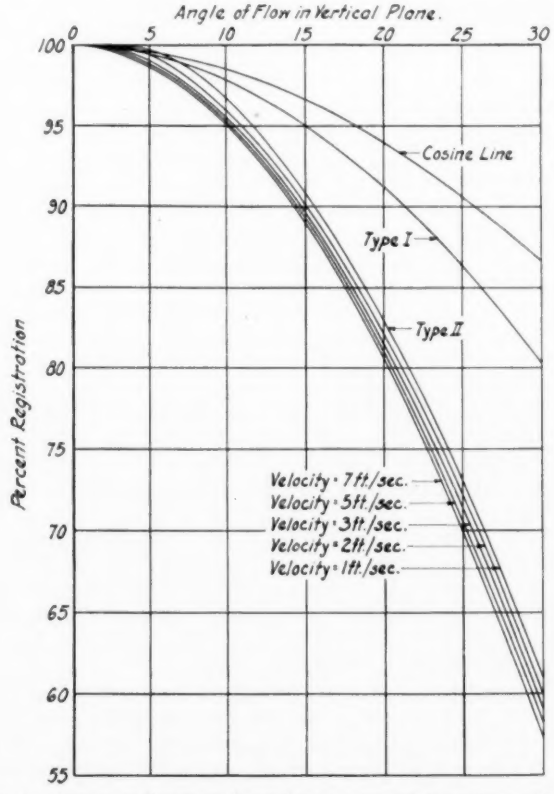


FIG. 6 AVERAGE OBLIQUE-FLOW CHARACTERISTICS OF SPOKE-VANE AND CONICAL-SCREW PROPELLERS USED AT SAFE HARBOR

There is a noticeable and actual trend toward minus values for the conical type with increasing velocities. Hereafter, the spoke-vane and conical-screw propellers will be referred to as Type I and II, respectively.

A detailed investigation has been made on the behavior under oblique-flow conditions of Type I and Type II propellers of Ott manufacture in connection with the calibration of the current meters for the Safe Harbor turbine field tests.

From Fig. 6 one may realize that there is a fundamental difference between the oblique-flow characteristics of the two propeller types aside from the magnitude of underregistration. The greater the velocity the more Type II underregisters for a given obliquity. On the other hand, for Type I they are not affected by velocity changes.

Based on the above and the following reasons, it is safe to state that both meters used at the Walchensee flume tests indicate oblique flow at the measuring section, independently of one another. The increase in underregistration of Type II with increasing velocity could only have been caused by angular flow. On the other hand, Type I indicates also obliquity, as only four test points show overregistration (tests 12 and 25). If the flow had been laminar, a more equal distribution between plus and minus departures would have been obtained for the Type I propeller.

It is not believed that turbulence was responsible for the results shown in Fig. 5, as it would have affected both current meters more and more with increasing velocity. The average departure curve for the Type I propeller is parallel to the horizontal axis of the coordinate system, and the same curve for the Type II is approaching an asymptote also parallel thereto, located for this particular case in the vicinity of approximately minus 1 per cent, or about twice that for Type I.

Fig. 7 shows the oblique-flow characteristics of Type I and Type

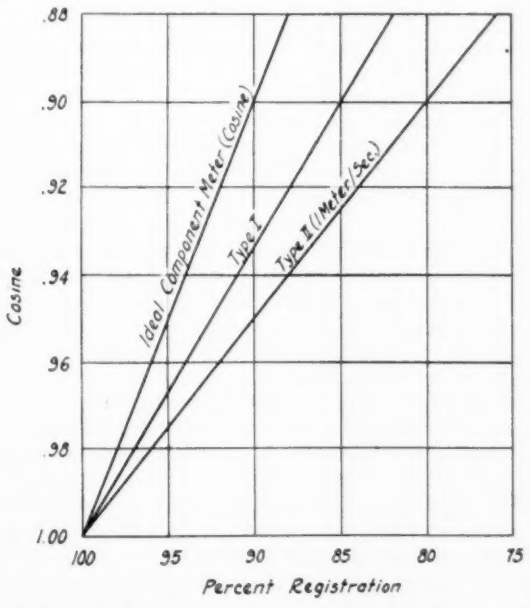


FIG. 7 OBLIQUE-FLOW CHARACTERISTICS OF SPOKE-VANE AND CONICAL-SCREW PROPELLERS USED AT THE WALCHENSEE, MOUNTED ON CURRENT-METER BODY TYPE V "TEXAS"

II propellers used at the Walchensee for a velocity of 1 meter per second. It may be noted that the Type II propellers tend to overregister slightly at small obliquity and very low velocities (see Fig. 6). The results as obtained at the Walchensee are quite logical and speak well for the precision with which the current-meter work has been done; although it has to be stated that for practical use the Type I propeller is more reliable, as the velocity range in general will be well above the one chosen for the Walchensee tests.

The experimenters averaged the departures of the two meters without paying any attention to the oblique-flow characteristics. In view of this fact European engineers have been puzzled by the

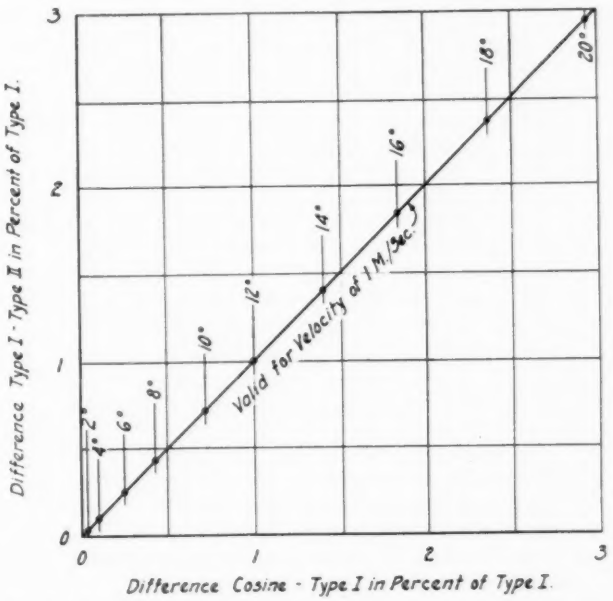


FIG. 8 CORRECTION CURVE FOR MEAN VELOCITIES, FOR MEASUREMENTS MADE WITH THE CURRENT METERS USED AT THE WALCHENSEE

result as presented, as it has been shown in the past that the Type I propellers of Ott manufacture have an oblique-flow characteristic closer to the cosine line than the Type II, and therefore the Type I propellers have been used more or less exclusively for turbine acceptance tests. It was suspected that algae or grass were caught in the spokes during the tests at Walchensee and that their presence slowed up the Type I propellers sufficiently during various runs to produce the established average deficiency for this type.

In connection with these tests it has been stated by Dr. Kirschmer and Mr. Esterer (*Z.V.D.I.*, Vol. 74, p. 1504) that none of the methods used at the Walchensee showed any dependence upon changes in velocity. That this statement does not hold true for the Type II current meter has been demonstrated by the ratings at the Bureau of Standards shown in Fig. 6, as well as by the average departure curve for the same type in Fig. 5.

To substantiate further these conclusions, a correction curve for mean velocities, discharges, or turbine efficiency is given in Fig. 8. The curve is derived from the original oblique-flow characteristics of the two meters used at the Walchensee put at our disposal by Dr. Ott, manufacturer of the meters, and presented in Fig. 7. This curve is true for velocities in the vicinity of 1 meter per second and will differ very slightly from the correction curves for higher velocities. As the underregistration of the Type II propeller is twice that for the Type I, the curve is a straight line, inclined at 45 deg, when the scales for the two axes of the coordinate system are chosen identical.

Special attention should be given the fact that the curves given on Fig. 7 are straight lines, because it is of fundamental importance. This characteristic may have been overlooked, because angle of flow has usually been plotted versus per cent registration.

Let it be assumed that there is a certain constant discharge at a gaging section. Then the mean ordinate vertical to the plane of measurement will be of constant magnitude.

When current meters with a straight-line oblique-flow characteristic, such as those in question, are used, we shall, theoretically, obtain at all times a discharge of the same magnitude; though the measured discharge will deviate from the true discharge a definite amount, depending upon the oblique-flow characteristic of the meter type used for the measurements. That the difference theoretically would be constant is due to the straight-line relation between cosine and per cent registration.

The correction curve shown in Fig. 8 will also be a straight line due to the individual linear oblique-flow characteristics of each propeller type. Therefore it is not necessary to correct for the velocity components registered in each current-meter location, but we may make use of the correction curves in reference to the established mean velocities or discharges obtained with each type of meter.

It is advisable to establish separate discharge curves obtained by means of each type of meter before the correction is applied, in order to eliminate the test-point spreading and because it is not essential that the true discharges for Types I and II current meters be alike, as the measurements do not have to be made simultaneously.

In connection with turbine testing, another procedure may be adopted—that is, to establish first the efficiency curves based on the discharges measured by means of each type of meter; the true efficiency may then be determined by means of the correction curves. A procedure of this kind is permissible, as efficiency is proportional to discharge.

It is apparent that an average correction of 0.5 per cent for the Type I measurements is necessary, as the vertical distance between asymptote for the Type II departure curve and the average underregistration for the spoke-vane current meter are of the same magnitude. (Correction was made by means of asymptote, as the oblique-flow characteristics of Type II for 1 m per sec only is known.) The results obtained by the Type I propeller justify this assumption, because a correction of 0.5 per cent would almost give a complete check with the volumetric measuring method, as the average underregistration would be 0.05 per cent, a result of utmost precision.

Based on the foregoing it may be realized that the two-type current-meter method is essential for results of high accuracy, even for the most ideal approach conditions. Results obtained with a water passage less favorable than the one at hand for these tests may be seriously doubted regarding their precision if the measurements have been made with but a single current-meter type. It has to be borne in mind that for gaging sections generally available for turbine acceptance tests the approach conditions are quite different, and most probably the mean angularities will be in excess of the one established for this particular set-up.

From the correction curve given in Fig. 8 it may be seen that the average angularity was approximately 9 deg. A value of this magnitude is astonishing for the ideal approach provided; although this result still further strengthens the present-day theory that laminar flow does not exist in open flumes and penstocks, but that the individual water particles follow an irregular path deviating by a large amount from the direction of the flume axis horizontally as well as vertically. This phenomenon is the very reason for the success of the salt-velocity method, and its existence has been demonstrated in connection with that method.

As in general approach conditions at low- and medium-head plants are not as ideal as those which were available for the flume-gaging section of the Research Institute, it had been adopted as standard practice abroad to provide temporary roofs or intake pier extensions in order to straighten out the flow to a satisfactory degree. However, based on the conclusions arrived at above, it may be safe to state that the magnitude of corrections due to obliquity may exceed 1 per cent even when special provisions have been made. This explains to a certain extent the high turbine efficiencies reached abroad.

It is to be regretted that the conical-screw meter alone was used for the penstock tests at Walchensee, as valuable data could have been obtained regarding the magnitude of obliquity to be expected in penstocks. It may be safe to assume that the mean angularity for straight penstock sections will be of the same magnitude as the one established for the flume at the Walchensee.

It has been pointed out above that the spreading of the test points is considerable for the flume tests. The magnitude of the spread is about 2.3 per cent for the spoke-wheel propeller (disregarding test 18-B) and about 2.6 per cent for the conical-screw meter. Most probably individual differences of smaller magnitude would have been obtained if no water had been wasted at the downstream end of the set-up, because the volumetric basin could not hold the total discharge passing the current-meter section during the time necessary for the measurements.

This difficulty could have been easily overcome in using more than one current meter of each type. It is believed that more consistent results would have been obtained in using five meters simultaneously mounted on a supporting rack. Arrangements of this type have been adopted as standard practice for turbine testing. By means of such a set-up the time interval necessary for the measurements would have been only 15 per cent of the one used for the tests and, furthermore, no water would have had to be wasted because the total discharge for one test run could have been retained in the basin.

Well-organized current-meter measurements may be made with a smaller spreading of test points, which is essential for this particular measuring method on account of the considerable time necessary to compile the data, and, in addition, because only a minimum of test runs is permissible from an economic point of view.

It is general practice abroad to sketch in the velocity diagrams close to the wall without actual measurements by means of pitot tubes. It is not necessary to adopt for this a pitot tube of the differential type, as one is not interested in absolute velocity values but in the shape of the velocity curve. When a supporting rack is available, pitot tubes, provided with one impact opening, may easily be located between the first meter and the wall, as well as between the first and second current meters. The shape of the velocity diagram derived from the pitot-tube measurements will then be tied in with the absolute velocity values obtained by the current meters. When such an arrangement is used, enough overlap is assured to make this tie-in possible. The pitot-tube tests can be made either simultaneously with the current-meter ones or separately for various discharges at the gaging section. A similar procedure is desirable for the bottom of an open flume, as was available for these tests or for power-house intakes. The total area of the gaging section at the Walchensee flume tests was relatively small and the area affected by wall and bottom is an appreciable percentage of the total area of water passage. Pitot-tube tests of the kind outlined above may have been of some assistance and may have affected the precision of the individual test points to a certain extent.

What has been outlined above is also important for the penstock measurements. The layout of the latter tests in reference to the current-meter setting cannot be called satisfactory. Only two diagonals intersecting at right angles were explored with the current meters, and therefore approximately 40 per cent of the penstock cross-section was not investigated. Too much leeway is left with such an arrangement. It would have been more desirable to install a revolving current-meter supporting rod

pivoted at the penstock center with enough current meters to cover the total diameter simultaneously.

One may conclude that too much weight was given to the distribution of the test points as shown on Fig. 5 and might call the distribution merely accidental. Fig. 9 gives the theoretical average departures from the true discharge for a wide velocity range to be obtained with the Type I and II propeller meters used at Safe Harbor for various mean angularities. The departure curves are based on the calibrations made at the Bureau of Standards. The plotting thus obtained is identical with the actual results at the Walchensee (see Fig. 5) and speaks well

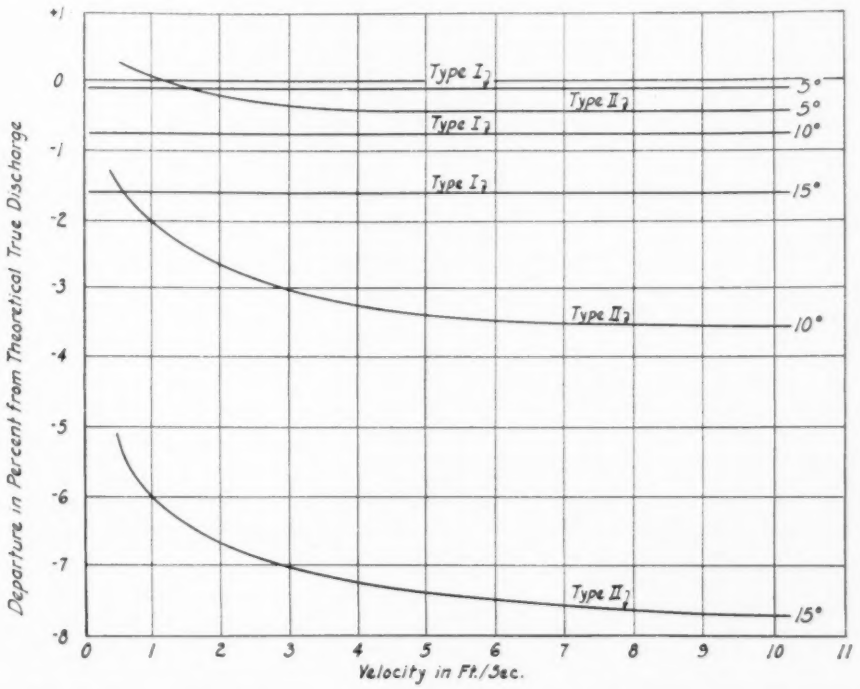


FIG. 9 THEORETICAL DEPARTURE CURVES FOR VARIOUS MEAN ANGULARITIES OF OTT CURRENT METERS USED AT SAFE HARBOR

for the precision with which the current-meter work has been done. The slight difference in shape of the Type II propeller departure curves is due to the fact that the oblique-flow characteristics of the Type II propellers used at the Walchensee and Safe Harbor are different, because the nominal pitches of the meters were 25 cm and 50 cm, respectively.

It has been pointed out by Dr. Kirschmer (*Z.V.D.I.*, Vol. 74, No. 7) that the established average current-meter departures of —0.3 and —0.6 per cent do not justify any doubt regarding the reliability of the current-meter method, because the magnitude of this deficiency is within the range of calibration discrepancies obtained at various rating stations.

Based on the analysis presented in the present discussion, this statement is not valid but needs some clarification. There are quite a number of calibration stations in existence, but it is believed that the most reliable ratings are made by Dr. Ott in Kempten, Germany, and by the Bureau of Standards in Washington, D. C. For velocity ranges generally available at current-meter gaging sections in connection with acceptance tests, these two stations check one another within 0.1 per cent. The main reason for this close agreement is that especial precautions have been taken at low-rating car speeds. All other rating stations have some pecularities which seem to make them unsuitable for current-meter calibrations in connection with turbine acceptance tests. The coinciding of the ratings obtained by Ott and the Bureau is very desirable, as efficiencies are measured by means of an identical scale, both here and abroad.

It has often been advocated to calibrate current meters in flowing water, and such a scheme has been mentioned in connection with the National Hydraulic Laboratory. Some engineers in this country have even seriously doubted the justification of current-meter measurements as long as this rating procedure was not adopted. However, based on the foregoing analysis, it may be realized that when two types of current meters are used, rated in still water for straightforward and oblique flow, true discharge measurements will be obtained.

The small discrepancy mentioned previously between the salt-velocity and current-meter methods established at the Holtwood Laboratory is based on the two-type current-meter method. Two series of runs were made, with different approach conditions for each series. The differences obtained between the two methods in both series of test runs were alike, namely, —0.08 per cent, taking salt velocity as standard. The conventional average differences, that is, the differences for each propeller type alone for both series of runs, were —0.35 per cent and —1.01 per cent for Type I and Type II, respectively.

It is hoped that the material here presented will be helpful in showing that the two-type-propeller current-meter method is essential, as turbine efficiencies obtained with one type, for instance, the spoke-vane type, are too high, even when all precautions have been taken to straighten out the flow by means of temporary roofs or intake pier extensions, and even in the case when the measurements are made in penstocks.

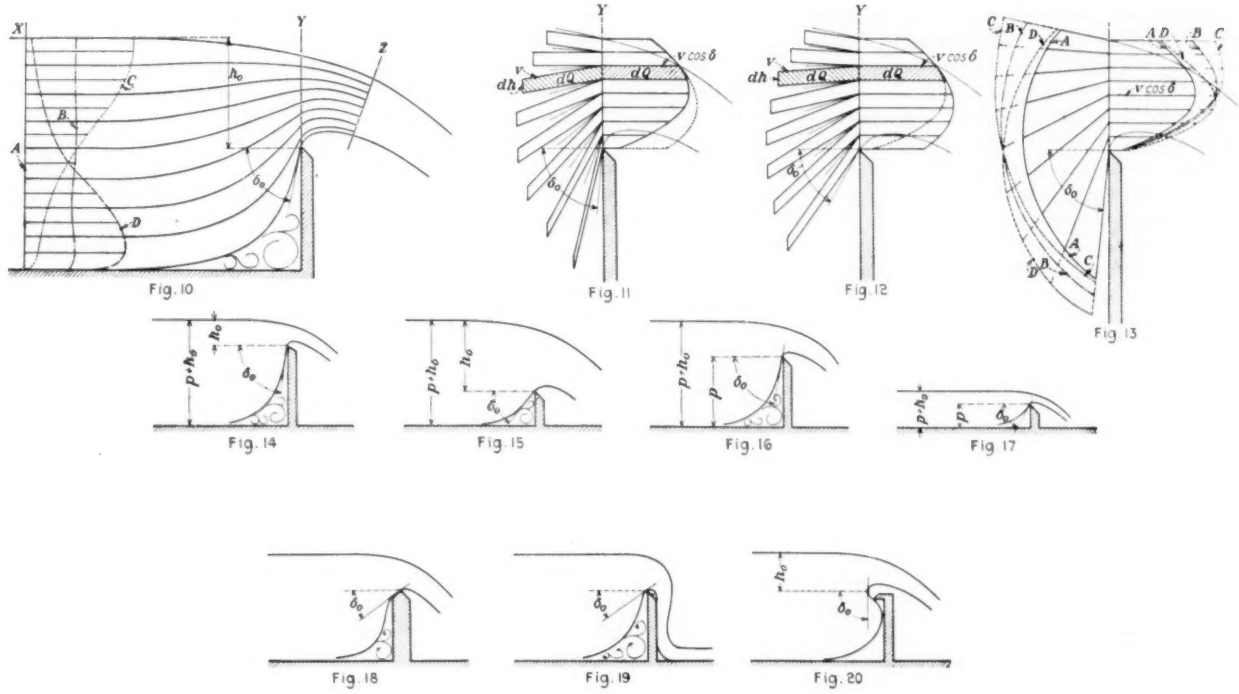
Another major advantage of the two-type current-meter method is that temporary installations may be omitted, as the influence due to oblique flow can be estimated. Such a procedure saves considerable expenditures and permits making the acceptance tests under actual operating conditions.

A. OTT.[6] It may be regarded as an open question whether or not too much is expected of the gaging method by means of a "standard weir," keeping in mind the various factors influencing the discharge. Based on tests in a small glass flume, the standard weir formula may be reconstructed so that the effect of the individual influences are expressed within the formula itself.

Fig. 10 illustrates the flow over a sharp-crested weir without side contraction. At section *X*, located a certain distance upstream from the weir, the water filaments are moving nearly parallel, although a different velocity is prevailing in the various flow layers. The vertical velocity diagram corresponding to this assumed flow condition will have, in general, a shape corresponding to one of the typical cases, *B*, *C*, and *D*, shown in Fig. 10.

In approaching the weir the flow filaments are gradually concentrated more and more until the maximum angular convergence is reached at the nappe section *Y*. Variable eddies are in existence between the lowest filament of flow and the physical

[6] A. Ott Math. Mech. Institut, Kempten, Germany.



FIGS. 10–20

boundary—that is, the bottom and the upstream side of the weir. Small, thin flow filaments, that is, horizontal whirls, are separated from the eddies and rise along the upstream side of the weir. These rising horizontal whirls naturally influence the lowest layer of the nappe at the crest of the weir, causing a change in the direction of the lowest flow layer, which will be deflected to an angle in reference to the upstream face of the weir.

The angular convergence decreases rapidly as soon as the nappe has left the crest and at section Z the flow filaments are again more or less parallel. Formerly section Z was usually taken as a base for the theoretical investigations to establish the discharge formula of weirs. Therefore a water quantity was estimated by multiplying the area of cross-section of the nappe by the velocity existing at this section. One assumes that the thickness of the nappe at this section bears a certain definite relation to the more easily measured water depth above the crest h_0, in order to avoid the difficult measuring of the nappe thickness at section Z. This definite relation mentioned above is represented by the coefficient of contraction.

It is believed that section Y would furnish better information of the discharge conditions, as the various elements of flow may be studied easier at that location.[7] This will be substantiated by the following simple illustrations:

In Figs. 11 and 12 are shown the discharge quantities for two flow conditions over the weir. The quantities are represented by the respective areas. Depth and velocity are taken alike for both cases, although it is assumed that the individual flow filaments have a different slope. To the left of section Y are plotted the partial quantities dQ, represented by small parallelograms lying in the direction of flow. Their long side may correspond to the velocity v, whereas their small side, being parallel to section Y, may represent the element of height dh; whence their volume becomes $dQ = v \times \cos \delta \times dh$.

To the right of section Y the same strips of area are changed into parallelograms of rectangular shape. Their height is dh and their length is equal to $v \times \cos \delta$. The sum of all the rectangles is equal to the discharge when the smooth curve is taken as the boundary for the area representing discharge quantity, and not the staggered line:

$$Q = \int v \cos \delta \times dh$$

Comparing Figs. 11 and 12 it will be seen that the quantity of discharge increases with increase of deflection of the lowest filament of flow away from the crest (that is, for smaller values δ_0). As the deflection is caused by the whirls rising along the upstream wall of the weir, and the whirls in turn become larger with increased roughness of the wall and increased viscosity of the fluid, one arrives at the conclusion that the discharge quantity will increase with increasing wall roughness and increasing fluid viscosity.

In Fig. 13 are shown four areas, A, B, C, and D, each corresponding to a discharge taking place under different conditions for the velocity of approach. The direction of flow is assumed to be identical for all four cases.

Referring to Fig. 10 the area A corresponds to discharge without any velocity of approach, and B refers to a case with an even distribution for the velocity of approach; areas C and D correspond to discharge conditions where there is velocity of approach in excess of the average near the top and bottom, respectively.

It is easy to visualize that the magnitude of increase in discharge depends very much upon the characteristic of the velocity of approach. Bottom flow increases the discharge the least, top flow the most, and the influence due to evenly distributed flow of approach lies between.

In establishing a discharge coefficient C for weir of given shape and in assuming this discharge coefficient to be the product of two factors, that is, the coefficient of convergence C_c and the coefficient of velocity C_v, one may reach the following conclusions based upon what has been mentioned above and by taking into consideration a few more graphical illustrations.

[7] Detailed analysis given in "Ausfluss, Durchfluss und Ueberfall," by Dr. A. Ott, *Wasserkraft und Wasserwirtschaft*, vol. 27 (1932), no. 18, and vol. 28 (1933). Special reprints obtainable from the author.

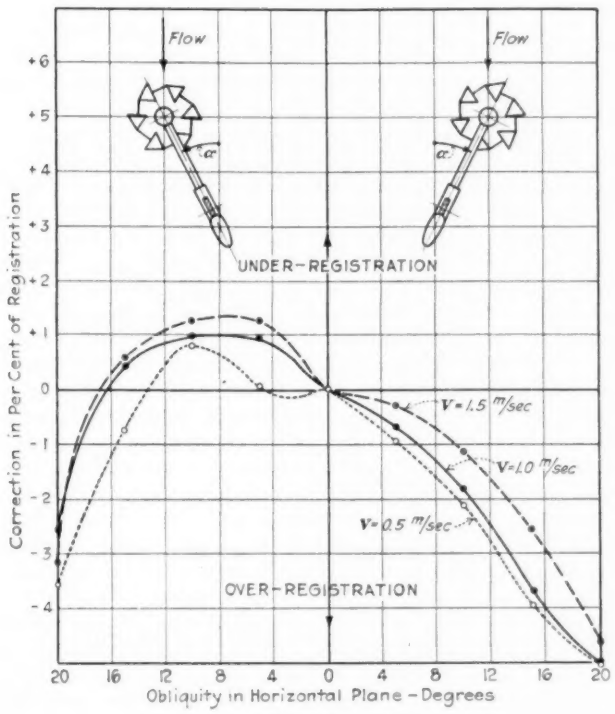


FIG. 21

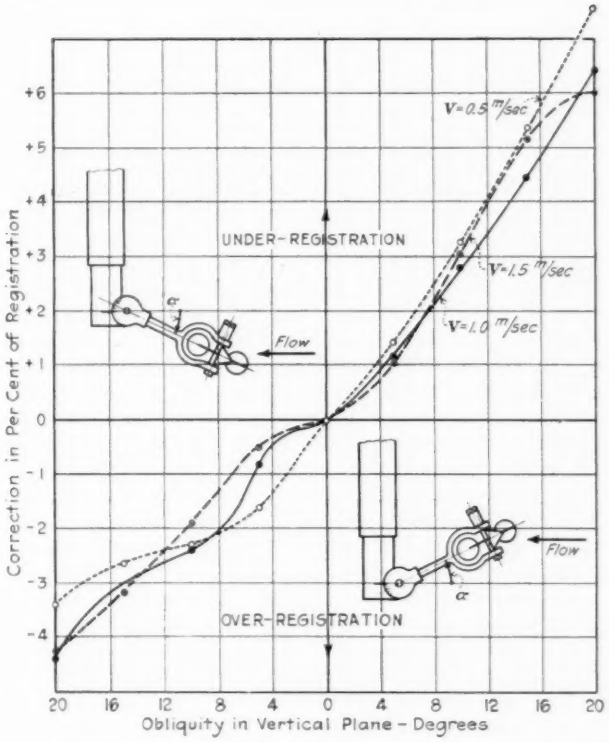


FIG. 22

C_e will be increased by any of the following factors: increase in the ratio $h_0/(p + h_0)$ (Figs. 14 and 15); decrease in the weir height p (Figs. 16 and 17); rounding off the crest (Fig. 18); suppressed nappe caused by insufficient ventilation (Fig. 19); increase in roughness of the weir surface or increase in viscosity of the fluid causing enlargement of the whirls (Figs. 10–12).

C_v will be increased by variation in velocity of approach distribution, being influenced the least by excessive bottom flow (curve *D*, Fig. 13), more by a uniform distribution (curve *B*), and the most by excessive top flow (curve *C*), and by insufficient ventilation causing a depressed nappe (Fig. 19).

C_v will be decreased by increasing angular convergence of the flow filaments, therefore slowing down their velocity.

Considering all these points, the number of which could be still enlarged as there are other influences crosswise to the direction of flow not mentioned above, it is not to be wondered at that discrepancies have been found in various discharge measurements made with weirs.

Referring to the comparative current-meter measurements, it is to be regretted that the experiments were conducted with the large Price meter, type No. 600, which is hardly used any longer; although some advantage may be derived therefrom. As this type of meter was used, the results obtained by the Research Institute may be compared with data published by B. F. Groat in 1913.[8] Mr. Groat made an intensive comparative study between a Price meter, type No. 600, and a Haskell screw-type meter. The Haskell meter had a characteristic somewhat similar to the Ott meter "Texas" equipped with a conical screw propeller and used by the Research Institute. In using both types of meter, that is, the Haskell and Price, under identical conditions, Mr. Groat found that measuring discrepancies of 1:6 are found respectively for the two types, and that therefore the result obtained by the Haskell meter should be increased by $^1/_7$ of the difference between the results of both meters.

[8] B. F. Groat, "Characteristics of Cup and Screw Current Meters," Trans. A.S.C.E., vol. 76 (1913), pp. 819–870.

Applying this conclusion to the results of the Research Institute, it is found that: Error of cup meter = +1.845 per cent; error of screw meter = —0.302 per cent; difference between both meter types = 2.147 per cent; one-seventh of this value is equal to 0.307 per cent; therefore the error of the screw meter after correction = —0.302 + 0.307 = +0.005 per cent. Mr. Groat's conclusion has been substantiated by the experiments of the Research Institute.

It is believed that one of the main reasons for the overregistering of the cup meter is that the cup wheel does not correctly integrate pulsations in flow because of the flywheel effect of the cup wheel, as well as because of the different action of front and rear side of the individual cups. Therefore somewhat larger velocity values are always established, which correspond to the average for the energy of flow. For flow without pulsation and obliquity, identical results may most probably be obtained with cup and screw meters, assuming that the mounting of the meter and its calibration are carried out in the same fashion.

The writer made some experiments with the new improved Gurley-Price meter, No. 622, and arrived at conclusions in regard to its behavior somewhat different from those generally accepted. The cup meter does not always register more than the true cosine component in oblique flow. It may over- or underregister for oblique flow from the left or the right, and from above or below, respectively.

Fig. 21 shows the corrections to be made on the measurements under angular-flow conditions and velocities of 0.5 m per sec, 1.0 m per sec, and 1.5 m per sec, in order to obtain the true velocity component vertical to the plane of measurement. It will be seen that the corrections are negative for angular departures to the right, but are mostly positive for those to the left; and further, that the meter is very sensitive for variations in flow.

In Fig. 22 are given the correction curves corresponding to oblique flow in the vertical plane. For this type of obliquity the differences are still more pronounced. For an obliquity of

10 deg to the horizontal and a velocity of 0.5 m per sec, this current meter over- or underregisters 3 per cent and 2.5 per cent for angular flow from below and above, respectively. The corrections may be estimated in accordance with a previous description given by the writer.[9]

CHARLES H. PIERCE.[10] It is not surprising that the outcome of the tests on the weir measurements "seems to substantiate the doubt that exists with regard to the use of the sharp-crested weir formulas for large discharges," as stated by the author. In fact, the several weir formulas that were used for comparison with the results obtained by the Research Institute differ as widely among themselves as they do from the volumetric results by the Research Institute for the range of head covered by the experiments.

It is now considered good practice to test the accuracy of weir measurements by means of the current meter. The use of the small Price current meter has enabled engineers of the United States Geological Survey to detect errors in weir computations, investigations generally showing that the errors were due to the use of weir formulas that were not applicable to the weirs actually used in measuring flowing water.

The relative accuracy of current-meter measurements as compared with computations of discharge by weir formulas is well illustrated by the results obtained by the Research Institute, the measurements by the two types of Ott meters being far more accurate than the results obtained by the weir formulas.

Although the author shows that the average percentage departures from the discharge as determined by the basin measurements for the 28 experiments was —0.302 for the Ott screw-propeller meter and —0.556 for the Ott vane-propeller meter as compared with +0.062 for the titration method, yet the consistency of the results of the meter measurements indicates clearly that very good work is done by the meters and that when carefully rated and properly used they can be depended upon for measurements of discharge with a high degree of accuracy.

It is unfortunate that the current meters used in the experiments by the Research Institute were limited to one type, with the exception of one antiquated large Price meter, 27 years old.

The large Price current meter was first designed and constructed by W. G. Price in 1882 and was used by him in measuring the flow of the silt-laden Ohio River at Paducah, Kentucky. The conditions under which Mr. Price was working required a meter that would function with a fair degree of accuracy at depths of 100 ft or more in water carrying as much as 5 per cent of silt and sand. The meter was not designed for use in measuring the discharge of small streams with a high degree of precision. About 30 years ago the design was modified so as to provide for the use of a smaller bucket wheel with cups of a different shape and a higher speed of rotation, although the fundamental features of vertical shaft, differential-action cups, conical pivot point, and silt-protected bearings were retained. During recent years various minor changes have been made in the construction, such as the provision for electric contact at every revolution or at every fifth revolution without changing the contact chamber, the use of stainless steel for shaft and penta gear, a pivot of larger diameter ground to a 90-deg angle cone, and specially heat-treated and tested metal for the pivot and bearing seat. It is now possible to assemble Price meters for use in low-velocity measurements which can be calibrated down to a velocity of about 0.05 fps and which show a continuous spin of $5^1/_2$ min in air before rotation of the cups is stopped by frictional resistance.

As the author has pointed out, the accuracy of current meters depends upon the ratings of the meters prior to use, with possibly a check rating after use, because only with careful and dependable ratings can meters be expected to give reliable results.

In general, the possibility of errors in current-meter ratings, as measured by percentage departures of individual observations from the mean curve, may be expected to increase as the velocity decreases. This is particularly noticeable for low velocities and is characteristic of the ratings or calibrations of all instruments where the rotating speed is the variable that is being calibrated, and it is equally true for current meters of the cup or screw type and for rotations about vertical or horizontal axes. As the frictional resistance of the moving parts of the instrument is decreased through improvement in design, better workmanship, superior metals, or other factors, the range of velocities in which frictional resistance may cause uncertainties in the rating is correspondingly reduced.

In the small Price meters as now assembled and used by the United States Geological Survey in stream gaging, the frictional resistance of the moving parts has been so greatly reduced that it appears to have no appreciable effect on the accuracy of ratings for velocities above about 0.8 fps, and in small Price meters specially adjusted for measurements of lower velocity its effect is still further reduced.

W. J. RHEINGANS.[11] The tests made to compare the various methods of measuring water which the author describes are of interest only as applied to open flumes and under ideal conditions of straight, undisturbed flow. It is to be noted that rain, high winds, and ice made testing impossible. Also that ground-water seepage was measured, although unfortunately no data are given as to the amount thereof.

In general, it is not surprising to note the apparently poor results of the Allen method as compared with the Ott meter, since the Europeans have for years been partial to the Ott current meter.

The results given substantiate the writer's contention and experience that large weirs in connection with high heads such as are used at power plants are either not very accurate or need special weir coefficients and formulas.

The metering section was ideal, having a long, straight channel, undisturbed flow, and no unnatural flow such as is found at turbine-plant intakes. But even under these conditions the Price meter read 2 per cent high, which is a strong argument against using it. What would it read under flow conditions found around power plants? The excuse that the meter was 27 years old does not mean much provided it was in good working condition and properly rated. However, since long straight channels of this sort are seldom found for a turbine test, the test made is only of passing interest.

In open-flume testing, elevation of water surface for determining areas becomes important, and no doubt wave action etc. can lead to large errors when the salt-velocity method is used.

Determination of the center of gravity of the salt curve is the main feature in the salt-velocity computations, and the method used, which consisted of burning the current intensity into paper with a spark gap, is not believed to be very accurate. A spark has a tendency to move back and forth, never going twice to exactly the same spot, and does not give a well-defined curve. Further, the personnel were apparently not familiar with the latest developments in the salt-velocity method, and therefore did not handle the tests or apparatus properly. For instance, the large minus error obtained in going from the pop valve to any of stations indicates an error in the timing of the salt intro-

[9] Trans. A.S.C.E., vol. 95 (1931), p. 829, Equation [8].

[10] Senior Hydraulic Engineer, U. S. Geological Survey.

[11] Test Engineer, Allis-Chalmers Manufacturing Company, Hydraulic Department, Milwaukee, Wis.

duction due either to water hammer or incorrect methods.

From tests at Davis Bridge, Mt. Shasta, and Big Creek, the writer has gathered enough data to convince himself that the error from pop valve to either of two electrodes or from electrode to electrode does not vary more than $^1/_2$ of 1 per cent and certainly not 4 or 5 per cent, as shown by the tests in Germany.

Either the open flume or lack of salt-velocity test knowledge is responsible for the large errors they found at Obernach, and the writer would hesitate to place much reliance on the results there with this method. Later, with tests in penstocks much better results were obtained in comparison with the other methods used.

In the salt-titration tests conditions were again ideal for accurate results. There was a good mixing, which is obtained at all power plants. However, there was exceptionally smooth flow at the sampling station, which is seldom the case at a power plant. Therefore the results can hardly be used for indicating possible power-plant results due to the highly disturbed and fluctuating flow usually encountered in the tailrace.

Individual tests made showed variations of 1 to 2 per cent, although the average for all the tests happened to be small. However, this means that five or six tests would have to be made at each gate to obtain any sort of accuracy.

The lack of knowledge of the fundamentals of the salt-velocity method by the Walchensee personnel is shown by the method of using the electrodes. Four short (2-in.-long) electrodes on the outer diameter of a 6-ft penstock will not give accurate results. Further, the velocity measurements by current meter and pitot tube were made only on one diameter, which would hardly give good results.

The disadvantage of the length of time required to make a pitot-tube traverse is pointed out, as well as the bothersome U-tube fluctuations. This is where the photoflow method could be used to advantage; it requires considerably less time than the current meter, and would probably offer less obstruction to the flow.

At Walchensee the current meter gave 1 per cent less discharge than the salt-velocity method, while at Obernach the salt-velocity gave from 1 to 5 per cent less discharge than the Ott meter. Note the close agreement between the pitot-tube and the salt-velocity methods. The pitot tube gives high discharges evidently at low flows, which fits in with Mr. Proebstel's statement that the lower the velocity, the lower the pitot-tube coefficient. Since the current meter gave 1 per cent better efficiency than the other methods, it would naturally be considered the best method.

The pitot-tube coefficient used is not given in the paper, neither the method of measuring static head etc., so we cannot draw any conclusions from these tests except that in general the methods agreed fairly closely. It is unfortunate that the Gibson method was used for only one point on the curve.

The writer agrees with the author's conclusion that performance of instruments under actual flow conditions is needed. The tests at Obernach so far have not fulfilled that need.

E. B. Strowger.[12] Reference has been made in the paper to tests using the Gibson method on one of the Walchensee units which resulted in only two usable diagrams. These two runs showed fair agreement with the current-meter results when compared with the current-meter rating curve, even though they involved a low turbine gate setting and consequently relatively low penstock velocities and probably relatively low velocities for the current meters. It was of course unfortunate that an incorrect spring adjustment in the apparatus precluded the use of the remaining Gibson diagrams.

[12] Assistant Hydraulic Engineer, Buffalo, Niagara, and Eastern Power Corp., Buffalo, N. Y.

The European designs of the Gibson apparatus have been ingeniously conceived and from appearances have been made with a high degree of precision. The results obtained at the Brunnenmühle plant by J. M. Voith using the Voith-Brecht apparatus and those obtained by the use of the Thoma apparatus (designed by Volkhardt and Deckel) were in close agreement with the corresponding weir and current meter results, respectively, taken under great care.[13] It is reassuring to one who has had the opportunity to apply the Gibson method of water measurement to a great many water-power plants to know of independent investigations of the method which show such good results.

It is the usual American practice in testing a turbine to make several runs at each gate setting, resulting essentially in a rating curve determined by averaged points. In this way accidental errors tend to be compensated. A casual or random error does not introduce a dangerous error in an average deduced from a series of readings. Accidental or casual errors of observation, as a practical matter, cancel out.

During the past summer the writer has made a series of experiments for the purpose of investigating the effect of variation in shape of the pressure-time diagrams obtained by the Gibson apparatus. These experiments were made by comparison of water measurements by the Gibson method with volumetric tank measurements using a quantity of water of about 4 cfs. As the effects to be considered were of relatively small proportions, it was necessary to reduce the probable error of the volumetric measurements by using a tank of unusual dimensions, namely, 57 ft long, 18 ft wide, and 4 ft deep. The time for filling with the quantity of water being measured ran from 10 to 15 min. The average variation between the volumetric and the Gibson measurements in one series of 15 runs was —0.2 per cent and in another series of 9 runs +0.4 per cent.

In May, 1931, the writer made an independent Gibson test of a power plant in New York State which had previously been tested by the Allen method in October, 1929. The station contained two Francis-type turbines operating under a head of about 200 ft. The results obtained by the two methods were practically identical.

In the long run, all methods of water measurement must depend upon true principles, and where necessary, proper coefficients will eventually be determined. But even if the principles employed are true, it does not follow that minor errors may not be introduced in isolated cases. Too much attention, therefore, should not be paid to a few scattered observations, particularly when means are not available for checking them.

Author's Closure

This paper on the Research Institute was written at the request of the late Dr. John R. Freeman to form part of a proposed supplement to his treatise, "Hydraulic Laboratory Practice." He had visited the experiment station in the summer of 1930 while the author was spending four months on the Walchensee studying the methods of outdoor hydraulic experimentation. Highly pleased with the spirit and capability of the research staff and with the layout and equipment of the new station, Dr. Freeman felt that his proposed supplementary publication describing more recent laboratory undertakings would be incomplete without extensive mention of the Research Institute.

It will be recalled by those familiar with Dr. Freeman's work that its primary purpose is the detailed description of laboratory layout, equipment, and experimental procedure; results of individual experiments are of secondary importance, as they are generally available in printed form in engineering magazines and

[13] "Water Measurements in Large Power Plants" (Wassermessungen bei Grosskraftanlagen), H. F. Canann, *Wasserkraft Jahrbuch*, 1930–1931.

handbooks. Of similar nature is the interest of a student traveling abroad on such a scholarship as those established by the various Freeman funds to acquaint young American engineers with modern European laboratory technique. Hence the author submitted to Dr. Freeman in the fall of 1930 an extensive description of the Research Institute at Obernach, approved by the institute director, stressing those features of interest to American laboratory men.

Since the proposed supplement to "Hydraulic Laboratory Practice" was not published, Dr. Freeman sent this paper to The American Society of Mechanical Engineers, by which it was printed in the Transactions of May 15, 1932, and then presented formally for discussion at the annual meeting the following December. Judging from the discussion that has been submitted, the primary interests of the readers were neither those of the author nor of his intended audience, for the remarks have been confined almost entirely to the necessarily brief account of certain investigations which were conducted by the Research Institute in the first years of its existence.

Inasmuch as the author is not responsible for the methods of experimentation and interpretation of results (which material was taken directly from publications of the institute), he is not in a position to defend this section of the paper against criticism. However, copies of the various discussions have been sent to the headquarters of the Research Institute at Munich, Germany, in order that the proper authorities would have the opportunity to reply to the numerous criticisms. The following remarks have been prepared under the direction of Dipl.-Ing. B. Esterer, of the Research Institute, and translated from the German by the author:

Comment on the Discussions

Mr. Allen's discussion, paragraph 4. It is obvious that the application of the salt-titration method followed an intensive study of literature on the subject and extensive laboratory investigations, through which the staff of the station had become familiar with the method.

With regard to the Allen method, the Research Institute had gained experience through the experiments at the Walchensee Power Plant, which showed that there was much room for improvement in the way those tests were conducted. Furthermore, for the experiments at Obernach all literature dealing with the Allen method was carefully studied.

If the results of the salt-titration method were better than those of the Allen method, this shows that the former method is easier to apply and control than the salt-velocity method.

Paragraph 5. A detailed description of the experimental layout is to be found in the publications "Vergleichswassermessungen am Walchenseewerk" (Comparative Discharge Measurements at the Walchensee Power Plant), V.D.I., 1930, No. 17, and "Die Genauigkeit einiger Wassermessverfahren" (The Accuracy of Several Methods of Discharge Measurement), V.D.I., 1930, No. 44, which Mr. Allen apparently has not used for his discussion. In the latter publication it is pointed out that a complete description is to follow.

Paragraph 8. It is much to be regretted that, as was afterward apparent, the British Pitometer Company entrusted with carrying out the Allen method did not seem to have sufficient experience with this type of measurement. The very purpose of these experiments was to have the various methods conducted by groups that were completely in charge of the individual methods. Once the British Pitometer Company had taken over these measurements, it was to be assumed that the method would be carried out with due regard to all previous experience.

Each group had the opportunity to arrange its apparatus as dictated by its experience or instructions. At the same time it is evident that due consideration had to be given to local circumstances—for instance, to existing openings in the penstock for installing electrodes etc., to avoid excessive drilling—which is obviously essential in measurements at any power plant in operation.

Paragraph 9. The number of injection valves, their location, and the arrangement and type of electrodes were left to the discretion of the British Pitometer Company. If the arrangement was unsatisfactory to Mr. Allen, this should have been clarified by securing the opinion of Mr. Allen before starting measurements.

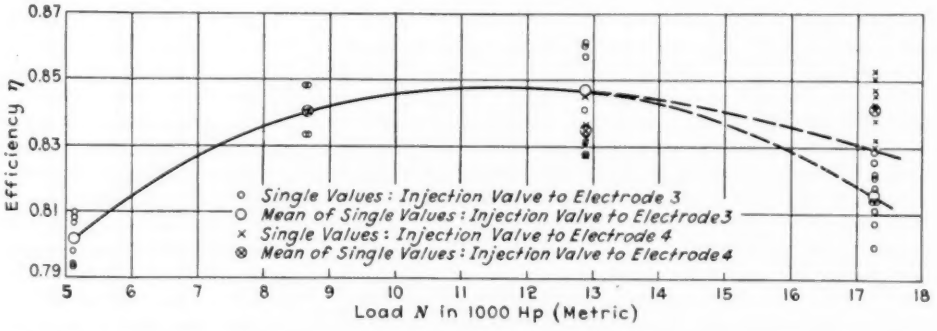


Fig. 23 Comparative Discharge Measurements, Walchensee Turbine Efficiency η Versus Load N (Turbine 6), Salt-Velocity Method

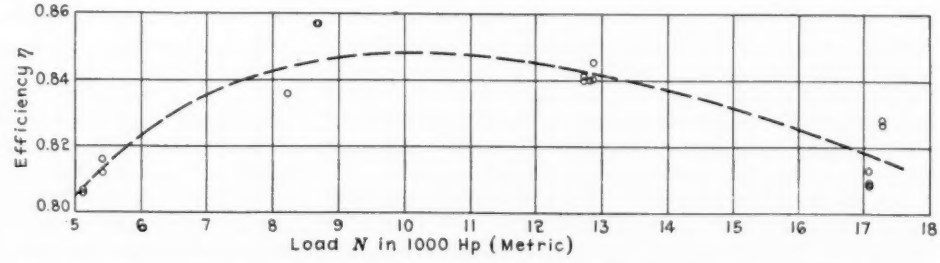


Fig. 24 Comparative Discharge Measurements, Walchensee Turbine Efficiency η Versus Load N (Turbine 6), Current-Meter Method

The arrangement of a number of electrodes along the penstock served the purpose, so far as we know, of determining to what extent the results depend upon the choice of position and interval, since no prior certainty existed as to the most favorable distance between point of injection and electrode.

Paragraph 10. Short electrodes instead of the well-known and time-proved transverse electrodes were used by the British Pitometer Company without further explanation. Hence it was to be assumed that these electrodes had been sufficiently tested for this purpose. The disadvantages resulting from the use

of these short electrodes have been pointed out in the article "Vergleichswassermessungen am Walchenseewerk."

Paragraph 11. The disadvantages of the chosen distances from point of injection to the electrodes are also brought out in the above-mentioned article.

Paragraph 12. It is true that the method of recording the salt curves in the experiments at the Walchensee plant was unfavorable. The Research Institute, therefore, avoided these disadvantages during its own investigations at Obernach by means of the method described in the publication "Die Genauigkeit einiger Wassermessverfahren."

Paragraph 13. Considering the apparatus used for the Allen method, the results must be rated as comparatively poor. It is certain that better results would have been obtained by more satisfactory apparatus; apparently the experience of the British Pitometer was not sufficient to devise at the outset such better apparatus. The results of the Allen method agree so well as they do with the efficiency plot because all results too much in error were omitted and because all values given in the table consist of mean values of the results for the intervals from the injection point to electrode 3 and to electrode 4. At the same time it must be realized that all the results of the current-meter measurements were plotted, without omission of poor values.

In the accompanying two diagrams (Figs. 23 and 24) the results of the Allen salt-velocity method and of the current-meter method, respectively, are plotted to give curves of turbine efficiency against load. In order to show to what extent the individual values for single injections in the Allen method departed from the mean, not only the mean values but also the individual values have been plotted. Similar individual values were also plotted for the current-meter measurements. It is shown that the error for single injections in the Allen method is at least fully as great as for the current meter. Under three-quarters and full load the results for the distances from the injection point to both electrode 3 and electrode 4 were averaged, while under lower loads only the average for the distance to electrode 3 was used. If one draws a curve through the average values for the distance to electrode 3, there results (the lower branch shown in the diagram, Fig. 23) an appreciably greater decrease in efficiency than by using the average value (upper branch) of the means for the two distances. With regard to Table 3 in the article, the following is to be noted: Experiment No. 8 contains an error, to which Mr. Allen calls attention and which unfortunately was not discovered previously. The efficiency should be 84.2 per cent instead of 83.4 per cent. Hence the curve of efficiency for the salt-velocity method is moved upward, near to the curve for the current meter.

Further checking also gave for the Allen method in experiment No. 7 somewhat different values—namely, for the discharge, 2.491 cu m per sec, and for the efficiency, 80.14 per cent. This error occurred through an erroneous mean value in the computation of Mr. Allen.

These corrections and the resulting change in position of the efficiency points will answer a number of Mr. Allen's remarks.

Paragraph 18. In full accord with Mr. Allen's opinion about the use of short electrodes and his suggestion to use instead transverse electrodes, are the remarks in the publication "Vergleichswassermessungen am Walchenseewerk," as was noted in Mr. Allen's further discussion. Concerning Mr. Allen's suggested location of electrodes, it must be noted that in most cases of installation of apparatus one generally does not know how long the expected salt curves will be and hence what position of the electrodes would be most favorable.

Last Paragraph. We add the following to the final remark of Mr. Allen: If the British Pitometer did not apply the method in accordance with the latest developments and all previous experience, this might in part be a result of the fact that it is on the whole very difficult with this method to ascertain at the outset, without lengthy preliminary experiments, the best manner of performance. If this method requires such exceptional experience, then this limits accordingly its field of application. It might thus remain undisputed that insufficient experience existed on the part of the British Pitometer Company; in this case, however, the company should have refused to carry out the measurements. With respect to the publication of complete descriptions and numerical results, it was pointed out in the article "Vergleichswassermessungen am Walchenseewerk" that this was only a preliminary report. It is evident that in the paper of Mr. Rouse on the Research Institute these details had to be condensed even further.

Mr. Billings' discussion, paragraph 3. In the experiments at the Walchensee Power Plant no agreement could be reached whereby the titration method could be used simultaneously with the others. The agreement included only the use of the current-meter and the Allen method for the investigations at the Walchensee.

Paragraph 7. The application of the methods of Allen and Gibson, developed and favored in America, have not yet been introduced in Europe, not because the European engineers are conservative or against these methods, but because the extensive and excellent experience with the current-meter method and the quality of the European meters have not necessitated a departure from this type of measurement.

Mr. Cole's discussion, paragraph 4. It must be admitted that it would have been possible to find a more suitable location for measurement than that chosen; unfortunately local conditions (accessibility and an already existing opening in the penstock, impossibility of rapid reconstruction, and impossibility of interrupting plant operation) made it necessary to conduct the measurements at this position. However, it must be emphasized that the current-meter measurements were conducted under identical flow conditions and that the comparison of the two methods was hence justified.

Paragraph 5. A supporting rod was used for the current meters, but not for the pitometer. The experiments showed, however, that in particular with high velocities, when the pitometer was far inside the penstock, an appreciable vibration of the tube took place, which surely did not improve the accuracy of the readings.

Paragraph 6. An undeniable advantage of the pitometer method is the fact that the velocity can also be measured very near to the walls. To offset this advantage, the difference in head in the U-tube during pitometer measurements at the Walchensee during low discharges was only about 60 mm, so that an error of 2 mm would already amount to 3 per cent. Such an error was easily possible considering the incessant fluctuation of the liquid column. During the larger discharges the difference in head in the U-tube increased to about 300 mm, and the probable error in reading decreased accordingly.

The statement of Mr. Cole that the outer ring of the penstock cross-section, in which region the proximity to the wall made current-meter measurement impossible, amounted to about 12 per cent of the total cross-sectional area, could lead to a mistaken understanding of the resulting error. This error will be comparatively small, because the velocity decreases rapidly in the neighborhood of the wall and because there has been established through comparison with pitot-tube measurements the curve of velocity in this region.

For the experiments at the Walchensee plant the flow near the boundary was determined from the equation

$$v = v_o \left[1 - \left(\frac{r}{R}\right)^n\right]^{1/7}$$

with $n = 3$ (v_o represents the velocity at the center). By using values of $n = 4$ and 5 to determine the form of the curve at the wall, the discharge would be changed by only 0.05 to 0.1 per cent.

Paragraph 7. The pitometer responds just as rapidly to variations in velocity as the current meter; the basic difference lies, however, in the fact that the current meter yields an integrative measurement of the fluctuations, while the pitometer shows instantaneous values and the fluctuations in velocity give rise to disturbing oscillations in the U-tube and a decreased accuracy in reading.

Paragraph 8. This conclusion in Dr. Kirschmer's original report is not based upon the assumption of a constant pitometer coefficient, but on the contrary was derived from Fig. 19 of his report. From this diagram it is evident that in the measurements at the Walchensee the pitometer gave too high values for small discharges.

Paragraph 11. Although the current-meter measurement No. 8 showed an appreciably higher efficiency than measurement No. 1 for a somewhat smaller load, no reason could be found to believe that an error in measurement had been made. Against such an error stands the fact that No. 8 consisted of two single measurements, which both gave exactly the same rate of discharge.

Mr. Mousson's discussion, paragraph 6. The very fact that the British Pitometer Company was entrusted with conducting the Allen method should have been a guarantee that this method would be carried out by a personnel theoretically and practically familiar with it. It is to be regretted exceedingly that this supposition was not justified to a sufficient extent. The experiments with the Gibson method were conducted at this time in order to try out a new pressure recorder that had been designed by Dipl.-Ing. Volkhardt under the direction of Professor Thoma. Since then Volkhardt has described this apparatus and further experiments with it in Publication No. 6 of the Mitteilungen des Hydraulischen Instituts der Technischen Hochschule, Munich.

Paragraph 9. In order to avoid confusion, it must be understood that the further discussion of Mr. Mousson deals not with the experiments at the Walchensee, but with the measurements at the experiment station in Obernach; the latter results were published in the article, "Die Genauigkeit einiger Wassermessverfahren."

Paragraph 13. The derivations of Mr. Mousson provide a logical and very welcome explanation for the results of the measurements at Obernach. It is necessary, however, to test these derivations very scrupulously. The Research Institute repeated the experiments in 1931 with the same Ott meters as in 1930, with results that depart appreciably from those of 1930.

Instead of a mean departure for the spoke-vane Ott meter of —0.56 per cent, this value became —0.78 per cent, and for the conical-screw propeller instead of —0.30 per cent a departure of +0.39 per cent from the results of the measuring basin, despite the fact that the measurements were again conducted with the greatest care. The investigations were extended to include the maximum discharge possible in the canal, with a velocity of 1.7 m per sec. An asymptotic approach of the variations to a constant value with increasing velocity could not be determined in these experiments with the conical-screw meter.

Unfortunately the investigations had to be discontinued because of more pressing matters. Since a further check has not as yet been possible, the results are mentioned only provisionally. These experiments will be continued this year, at which time the variation noticed in 1931 will be checked and a thorough investigation will be made along the lines discussed by Mr. Mousson.

Paragraph 15. It is not probable that the results of the Type I current meter were influenced by the presence of algae or similar material caught in the spokes, since the meter was watched constantly to avoid just this occurrence.

Paragraph 29. In order to reduce to a minimum the errors which could occur during the current-meter runs, the head at the measuring weir was recorded continuously. If the average head during the time of the current-meter measurement did not correspond exactly with that at the time of filling the measuring basin, then the results of the basin measurement were corrected according to the $^3/_2$ power of the head on the weir.

Contrary to general practice, the measurements were carried out with one meter at a time, because only one Price meter was available and all meters were to be tested under identical conditions.

Paragraph 32. Measurement of the velocity curve in the immediate neighborhood of the channel walls was omitted for the following reasons:

1 The accuracy of customary meter measurements was to be determined—not for cases in which special means for increasing the accuracy are employed.

2 The form of the curve in the neighborhood of the walls is sufficiently well known under normal conditions of flow (seventh-root law), so that plotting the transition near the wall and floor can be undertaken with all desired accuracy.

3 Through measured points the form of the curve is determined to such an extent that in drawing the transition curve to the wall not much freedom remains. Outward displacement of the velocity curve at one point usually calls for an inward displacement in the neighboring section, through which the differences in area are counterbalanced. From several trial diagrams of the velocity distribution it was found that changes in the curve near the walls would not change the result by even 0.1 per cent.

4 Even when the form of the velocity curves at the boundaries of the individual horizontal or vertical sections has been determined exactly, there still remains a certain amount of discretion and uncertainty in determining the boundary transitions for the summation curve of all the individual sections.

Paragraph 33. Again it must be emphasized that at the Walchensee as well it was not a matter of conducting the meter measurements in an especially refined manner, but in the customary way, and then of comparing these results with those of the other methods. Hence the measurements of velocity were undertaken across a single diameter.

Paragraph 36. The exceptionally good agreement of the ratings of the Bureau of Standards with those of Dr. Ott is remarkable, but should not, however, lead to discontinuing the study of rating accuracy and agreement between results of various stations. It is true that the results of different stations show discrepancies principally at low velocities and approach the same value as the velocities increase. Nevertheless, this range is rather great, so that for a velocity of about 1.5 m per sec differences up to 1.5 per cent have been determined in different rating stations—although for other types of meter than those used by the Research Institute.

Paragraph 39. The way pointed out by Mr. Mousson of increasing the accuracy of meter measurement through use of the two-type current-meter method is especially worthy of notice. There are surely many instances in which one will undertake the necessary duplication of apparatus and increase in time for measurement and computation, when it is essential to reach a particularly high degree of precision.

Dr. Ott's discussion, last paragraph. In agreement with the following discussion by Mr. Pierce, it may be added at this point that the newer Gurley-Price meter is apparently greatly improved and also more suitable for measurements over comparatively small cross-sections—as, for instance, in the canal used by the Research Institute. In the summer of 1931 measurements were conducted by the Institute with the Gurley-Price meter No. 622 mentioned by Dr. Ott, with a mean departure (based on 20 single

measurements) of —0.6 per cent from the result of the measuring basin, while with the old model Price meter in the experiments of 1930 a departure of +1.84 per cent was determined. This result was obtained in the long, straight experimental canal of the Research Institute under conceivably favorable conditions. However, this result is given only provisionally at this time, since the experiments, as already mentioned, were not completed, and in particular were not controlled through subsequent rating of the meters. How the new Price meter agrees with basin measurements in turbulent flow and under various conditions of angularity is to be investigated through measurements this summer. Even the new type of Price meter showed during rating appreciable variation when towed near the left and right walls and near the surface and channel bottom. In plotting the results for a horizontal section, the velocities near the one wall were greater and the velocities near the other wall less than those measured with the Ott meters. The accompanying differences in area as compared with the Ott curves, however, nearly counterbalanced each other.

A welcome and practically valuable problem to investigate would be a study of the characteristics of various meters in flow such that the velocity varies greatly across the width of the meter—as is the case, for instance, in the neighborhood of channel walls.

Mr. Rheingans' discussion, paragraph 6. Recording the salt-velocity curves by means of a spark was decided upon after extensive preliminary experimentation. Use of the customary recording apparatus with pen and ink showed a lag in comparison with records of an optically registering oscillograph, as well during rise as during fall, which would seem to be a result of the friction of the pen on the paper and the viscosity of the ink. Such frictional restriction is avoided through use of a sparking apparatus. The tiny punctures left by the sparks in the paper record lie with very slight scattering on either side of an easily constructed mean curve. In the case of the size of diagram used by the institute (about 100 mm in height), the error due to this scattering effect would probably be less than that due to the frictional lag of the pen.

It was impossible to explain this great negative variation (see latter part of paragraph 6) through error in measurement of the time of injection, since the time record was controlled automatically by the opening and closing of the valve on the injection pipe and also registered automatically by means of a recording amperemeter. It is certain, however, that at every injection into the open canal there resulted a disturbance of the flow, a sudden backwater effect, which was noticeable on the surface and probably gave rise to the strong negative error in time for every measurement beginning at the point of injection. In closed pipe lines flowing full this source of error becomes practically negligible, since the momentum of the injected liquid is very small in comparison with that of the flow in the pipe. Hence there is hardly room for contradiction of the statement, based upon the experiments at Obernach, that the Allen method is less suitable for open channels and despite the greatest care is subject to more errors than when used for flow in closed pipe lines.

Paragraph 9. An essential prerequisite for the titration method is the existence of a strongly turbulent flow. The manner of flow at the sampling station is of no concern. The high efficiency of modern turbines generally precludes the probability of strongly turbulent movement of the water through the turbine. Hence the titration method is confined principally to measurements in mountain streams etc., as is noted in the article "Die Genauigkeit einiger Wassermessverfahren." In just such important cases the use of current meters is far inferior to the titration method, and the latter often affords the only possibility of a practical means of measurement. The Research Institute has conducted a great number of such measurements, described in the publication of Dr. Kirschmer, "Das Salzverdünnungsverfahren für Wassermessungen" (The Salt-Dilution Method for Water Measurements), *Wasserkraft und Wasserwirtschaft*, 1931, No. 18.

For discharge measurement of turbines the use of the titration method is limited to special cases in which thorough mixing is certain to occur—for instance, in Pelton wheels, in the waste pipes of which complete mixing almost always takes place. In such cases good results have already been obtained, as is also noted in the foregoing publication of Dr. Kirschmer. If the necessary high degree of turbulence is not present to a sufficient extent at a power plant, favorable results can still be obtained through particular care to distribute the injected salt solution. Through the extensive apparatus necessary for such a case, and through the great number of samples which must be taken, the principal advantage of this method, namely, its simplicity, is lost.

Paragraph 11. This criticism concerns the company which conducted the Allen method at the Walchensee plant. As already mentioned, it was a matter of handling the current-meter measurements at the Walchensee plant as it is customarily done in penstocks in Germany, and to compare this method with the others. Hence measurements were made across a single diameter.

Paragraph 13. Through correction of the efficiency for run No. 8 (Allen) the agreement of the efficiency curves for the current-meter and the Allen methods was improved considerably. Next to the too low values of the efficiency as measured by pitometer under the smallest load, also established by Mr. Cole, the great scattering of both pitometer measurements under the greatest load is to be emphasized, which makes a difference of about 4 per cent of efficiency. Hence it could be considered more as a coincidence that the mean of these two measurements agrees with the result of the Allen method at this load. The efficiency points at three-quarters load for pitometer and current meter are within less than 0.5 per cent from each other, while at one-half load the pitometer results lie from about 1 to 1.5 per cent lower than those of the current meter, and also from about 0.3 to 0.9 per cent below those of the Allen method. The two accompanying diagrams (Figs. 23 and 24) show that the efficiency curve for the current meter up to a load of about 11,000 hp lies higher than the efficiency curve for the Allen method, and lower from there on. The value of the maximum efficiency is about the same. Under these conditions it cannot be a matter of favoring the current-meter results as expressed by Mr. Rheingans in his sentence, "Since the current meter gave . . . ."

Paragraph 14. In the article, "Vergleichswassermessungen am Walchenseewerk," further details are given concerning the manner in which pressure differences were measured for the pitometer. The transformation of the differences in head into velocities was not accomplished on the basis of pitometer coefficients, but through use of rating curves of the British Pitometer Company.

General Comment

The author wishes to express the hope that the foregoing translation will be satisfactory to the Research Institute in Munich; unfortunately both the discussion and its translation had to be undertaken in an extremely short space of time.

It is obvious that such discussion as has been submitted could continue indefinitely. The only remedy seems to lie in the following suggestion of Mr. Sigmund J. Bitterli, quoted from his article, "Making Water Measurements With Current Meters" (*Power*, January 19, 1932):

"Methods of making acceptance tests in Europe differ distinctly from those used in America. Each method possesses its particular advantages, but has peculiarities that must be

mastered before satisfactory results may be obtained. Unfortunately there exists considerable prejudice on the two continents against the use of testing methods that have apparently given satisfaction in the other. It is desirable therefore that truly comparative tests could be made.

"Various attempts to secure comparative results have been made. As each method requires the technique of those who are familiar with their own methods, such comparisons cannot be successfully made unless the respective experts can be present in the field. It is hoped, therefore, that such tests can be made either in America or Europe, with the advocates of the various methods actually present."

Progress in Iron and Steel

THE rate of pig-iron production for the year of 1932, to date, has been unprecedentedly low, if judged by twentieth-century records. One must turn back to 1898, 35 years ago, to find an annual production less than the 10,162,776 tons that will be produced in 1932 if the output for the first six months, of 5,081,388 tons, is a proper criterion of what is to follow for the remainder of the year.

By any comparison the showing for the first half of the current year is a sorry one. It is only 56.5 per cent of the 17,957,779-ton rate of 1931 and only 46 per cent of the 11,022,349 tons produced in the first half, the corresponding period, of 1931; and no one was bragging about production in 1931. Compared with the previous 10-year average of 33,955,000 tons, the rate for the first half of 1932 was 30 per cent.

Combined open-hearth and bessemer steel ingot output for the first half of 1932 is estimated at 7,567,769 tons, or at the rate of 15,135,000 tons per year, as compared with 25,193,000 tons in 1931, or 60.3 per cent of last year's rate. It is about 36 per cent of the previous 10-year average.

Both pig-iron and steel operations were at record low percentages as compared with potential capacities.

Tonnage output alone, however, is by no means a complete measure of progress in the industry, and it would be unfortunate indeed if the unusual opportunities afforded by this period of abnormally low rate of furnace and mill operation were not marked by a considerable advancement, at least in the technical and economic phases of iron and steel manufacture. It is true that the necessity for conserving cash resources has prevented large expenditure for plant improvements, and except for work started during or shortly after the prosperous period of 1927 to 1929, there has been no expansion in capacity worthy of mention. One does not look in vain, however, for evidence of successful effort along the lines of raw-material preparation and handling, equipment manufacture, plant design, and operating methods.

As a matter of fact, the exigencies of the depression have greatly stimulated the interest of steel-plant executives in the study of ways and means to increase profits by other procedures than increasing output capacity, which long-continued policy has undoubtedly had something to do with the present lopsided relationship between supply and demand and which in turn is leaving many manufacturers with an almost unsupportable burden of fixed interest and depreciation charges on idle plants. At any rate, it is now thoroughly realized and appreciated that increase of profits by reduction of costs is a wiser method, and during the past year there has been a tremendous amount of investigation, study, and planning with this end in view.

The full measure of what is being accomplished cannot be taken until a resumption of more nearly normal operations permits a fuller application of methods and equipment now being developed, but the following reports prepared by engineers who specialize in and are closely in touch with the several branches of the industry enumerate and comment on some of the more promising new developments.

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BLAST FURNACES

Some of the most interesting developments in pig-iron manufacture during the last year are to be found in operating practices that were originally considered purely emergency measures, but that are now being hailed as permanent improvements. The most important of these has been "slow blowing."

To the experienced blast-furnace operator the advantages to be derived from a careful adjustment of blast volume to the size and design of the furnace and particularly to the quality of raw materials have always been realized. The trend to large 1000-ton furnaces, as the writer pointed out in an article published in 1926, could scarcely have been justified on the basis of increased productivity alone. Careful observers at that time were fully aware of the fact that one of the chief virtues of the large units was their ability to turn out a required tonnage without being forced to take an amount of blast out of proportion to the volume capacity of the stack. There was always previously, however, a great pressure exerted upon operators to squeeze the maximum possible tonnage out of every existing unit. Recently, the pressure from the plant management has been to get as little tonnage as possible, while keeping the furnace in operation as a vital link in the chain of steel-plant manufacture, not only for making the requisite amount of hot molten metal, but as a producer of gas upon which the manufacture of coke and coke-oven gas, the generation of power, and the heating of metallurgical furnaces are largely dependent.

This situation has resulted in "slow blowing," or blowing the furnace at a rate somewhere below the point of maximum production. Blast volumes were gradually reduced to keep pace with the operation of other departments. This was not done entirely without reluctance and opposition on the part of the furnace operators, for furnace men have been trained and tuned up to the policy of large production, which means heavy blowing. Many had anticipated serious operating difficulties from building up of hearth, bosh, and stack, with resultant hanging and checking, but these were not encountered. On the other hand, surprisingly good results followed. Flue-dust losses were substantially reduced, coke consumption was decreased, and the quality of the iron became more regular and better.

With the demand for tonnage declining over a period of almost three years, it naturally followed that the advantages derived from slow blowing should be pressed to the utmost, and today we have many instances of stacks being driven at only about one-half of what was considered their normal rate. The figures of Table 1 are typical.

TABLE 1

	Normal operation	Slow blowing
Cubic feet of wind, per minute	48,000	25,000
Tons, per day	762	372
Coke, per ton	1,638	1,707
Amount of scrap used, per cent	12.6	None
Flue dust produced, pounds per ton of pig	353	111
Loss of metallic, per cent	6.50	0.33
Off-sulphur iron, per cent	0.36	None

An examination of these figures discloses the tremendous reduction of losses that can be accomplished if gas velocities through the furnace are kept at a minimum. Notwithstanding the denser burden brought about by discontinuation in the use of scrap, the flue-dust losses were cut to one-third of their previous

figure. Such results are of course only accomplished with excellent furnace supervision, but the case under consideration is one of many. At another furnace in the same plant flue-dust losses were reduced 80 per cent, and at many other plants throughout the country we now find less than 100 lb of dust being made per ton of pig. Such a small amount of flue dust can feasibly and economically be returned to the furnaces without benefit of sintering.

The thermal efficiency of the operations shown in Table 1 likewise speaks well for slow blowing, if due consideration is given to the amounts of scrap consumed. Given an equal amount of scrap in the burden, the advantage in slow blowing would have been not less than 200 lb of coke per ton of pig iron. It is interesting to notice that even with the extremely low scrap prices prevailing, many companies have found it desirable to conserve their cash by reducing stocks of ore on hand, particularly since by slow blowing such a complete recovery of the metal in the ore can be effected.

The adverse results of slow blowing are of course higher labor costs and higher fixed charges, which are directly proportional to tonnage. But even at a 50 per cent rate, the net effect in many cases has been a final cost of pig iron which is as low or even less than that obtained by so-called normal operations. From this we may rest assured that after the demand for iron returns there will be a great effort made to find the rate of blowing that will produce the maximum of profit for the plant rather than the maximum output.

Associated with slow blowing, as a means of keeping blast-furnace production in step with hot-metal requirements, is intermittent operation. Here the results, while not so productive of economies as slow blowing, have furnished some surprising upsets to former accepted theories.

Short periods of banking, especially over the week end, had been tried many times in the past, but with such unsatisfactory results as to discourage any regular procedure of this nature. Intermittent operation as now practiced was probably first suggested by the advantages of week-end banking, but in the past year a technique of performance has been developed that far surpasses anything previously employed. Instead of a complete shutdown of the furnace, a slight blowing is kept up—for characterization of which a new word, "fanning," has been coined. In the past, fanning a furnace was something that operators were prone to avoid, most of them preferring a complete shutdown, for the same reasons that have been listed against slow blowing.

Now, however, the beneficial effects of fanning, as compared with complete stoppage, are becoming apparent. Fanning keeps the hearth hot, probably has a tendency to keep the furnace walls clean, eliminates the necessity of purging the mains, and accomplishes a considerable amount of "predigestion" of the stock, which greatly facilitates easy resumption of blowing. In this way the necessity of coke blanks is avoided, and operations are carried on with a rather amazing thermal efficiency.

Almost any combination of blowing and fanning periods can be worked out, either to meet the tonnage requirements or to accommodate periodic demands for hot metal. In almost every case, however, it would seem on the basis of theory that the best results would obtain when the period of blowing is not longer than will be required to empty the furnace of stock "predigested" by fanning; and in practice this procedure is generally followed. From 4 to 12 hours' blowing, alternated with periods of fanning of from 4 to 20 hours in duration, covers most of the cases with which the writer is familiar.

In general, the lesser advantages from intermittent operation lead to the use of continuous slow blowing where conditions are such as to make the method of reducing production optional, and this is in spite of the fact that during the fanning period a complete laying off of the operating force, with the exception perhaps of one foreman or water tender, is generally followed. As with slow blowing, however, it may be stated with assurance that intermittent operation has been so successfully worked out in practice that it may be considered a permanent contribution to the art and one that will at times be utilized to advantage after the depression is behind us.

Reference has been made to the tendency of producing companies to conserve cash expenditures by the elimination of purchased scrap from the furnace mixtures. The unprecedentedly low prices of scrap materials, however, have made it economically desirable, if other conditions permit, to use large proportions of scrap, and in a few cases advantage has been taken of this unusual situation. It has been interesting to note just how much scrap could be carried without interfering with the smooth working of the furnace. One case is of record where 1130 lb of scrap per ton of pig was used for a period of several weeks. The feasibility of using large quantities of scrap has a great bearing on the relative cost of producing iron in the several large producing centers, and the low scrap prices that have prevailed during the past year have further accentuated the continually growing differential in material assembly costs in favor of the Detroit and Chicago districts.

During the past several years there has been a growing interest in the more profitable utilization of blast-furnace gas. Following the pioneer installations at the Hamilton, Ont., plants of the Steel Company of Canada, a number of American concerns have successfully applied blast-furnace gas to the heating of soaking pits and heating furnaces, and in some cases, mixed with coke-oven gas, to open-hearth practice. The year of 1932 has seen only a little extension in this practice, but at one more plant at least blast-furnace gas has been used for heating by-product coke ovens.

These uses of blast-furnace gas are not new, but they are leading to some interesting developments in the way of equipment for the more economical and thorough cleaning of blast-furnace gas and its conservation by improving the efficiency of hot-blast stoves. Up until this year the only proved dependable method of thoroughly cleaning blast-furnace gas has been by means of Theisen washers or mechanical scrubbers of the disintegrator type. Mechanical scrubbers have had several objectionable features, chief among which may be mentioned the high cost of operation and the inability to dry the gas. The use of disintegrators has also brought about an additional hazard in handling gas.

In a paper read before the A.S.M.E. in 1930, it was pointed out that the successful development of a wet electrical precipitation method would accomplish a solution of the problems that the disintegrator had failed to meet in a satisfactory manner. Subsequently there has been a very active application of attention to the perfection of the electric cleaner, stimulated no doubt by the fact that the basic patents on the precipitation principle have recently expired, and 1932 has been marked by satisfactory installations of at least two electric cleaners of distinctly different design. When it is understood that the power consumption by the electric method is only about one-tenth that of mechanical scrubbing and that dry gas, cleaner than 0.01 grain per cubic foot, is obtainable, the importance of this development may be readily appreciated.

In at least one type of electric cleaner the final cleaning apparatus may be combined with the primary washer to form a single compact unit, thus saving valuable space and simplifying the entire cleaning system which with its various units, pipes, and fittings had begun to take up more room than the furnace or stoves. We may look for a rapid application of this type of

cleaner, which promises to save about $8^1/_2$ cents per 100,000 ft of gas cleaned, or approximately $10^1/_2$ cents per ton of pig iron produced.

For the conservation of gas and the attainment of higher blast temperatures the year has been marked by the introduction or at least the placing on the market of additional types of stove checkers designed for the purpose of more rapid and efficient exchange of heat between gases and blast. In keeping with the spirit of the times the advantageous utilization of clean gas by increasing the heating surface and heating capacity of existing stoves without removal of the old checker work has received particular attention. One of the most novel and promising developments in this connection has been the design of spheroidal-shaped refractories for insertion in old checker flues, when areas are unnecessarily and wastefully large for using thoroughly clean gas. Inserts of this type will obviously solve the hitherto impractical if not impossible job of getting them into the flues without dismantling the old checkers. At any rate, 1932 has witnessed in this development another promising innovation.

There have been some new developments in the design of slag and hot-metal handling equipment, noteworthy among the latter being ladles of all-rolled and welded steel construction that have the advantages of being lighter and probably of lower first cost. Development of better refractories for blast furnaces and stoves continues, with a trend toward high-alumina brick for service where extremely hot gases have to be handled.

A new type of dust catcher utilizing a zeppelin or streamline principle for more equal distribution of gas and prevention of turbulence has been designed and built; double-compartment bosh and stack plates applying the advantages of double-compartment tuyères to the cooling of furnace walls have been brought on the market.

The rapid growth of and the demand for cheap forms of insulating material have revived the manufacture of slag wool, and considerable success has attended at least one progressive plant in the development of new methods for perfecting this process and improving the quality of the product.

All in all, blast-furnace designers and operators may take pride in their strenuous efforts to make the best of what has been, so far as output is concerned, the worst year that the industry has had in a generation, and find satisfaction in the fact that for them the period of the depression has been by no means devoid of progress along technological lines.

Lake Superior Iron Ore

During the last 20 years (1912 to 1931, inclusive) the Lake Superior district has produced approximately 84 per cent of the iron ore mined in the United States. Its record is shown in Table 2.

TABLE 2

5-Year period	Total tons shipped	Average tonnage per year
1912–1916.....	245,912,511	49,182,502
1917–1921.....	259,825,024	51,965,005
1922–1926.....	264,215,238	52,843,047
1927–1931.....	244,040,988	48,808,198
20 years.....	1,013,993,761	50,699,688

Peak year, 1916, 66,672,881 tons shipped
Peak year, 1929, 66,157,359 tons shipped
Low year, 1921, 22,851,995 tons shipped
Low year, 1931, 23,496,228 tons shipped

Districts in the United States other than Lake Superior have averaged about 10,000,000 tons' annual production over the last 15 years.

The average analysis of Lake Superior shipments does not vary much from year to year, as shown in Table 3.

TABLE 3

Year	Iron (natural), per cent	Phosphorus, per cent	Silica, per cent	Manganese, per cent	Moisture, per cent
1922....	51.87	0.099	8.23	0.76	10.78
1931....	51.53	0.087	8.60	0.80	10.84

Such uniformity is not a natural phenomenon, but is maintained only by a system of careful mixing or grading, which operation provides one of the problems constantly faced by the mine operator. To many people iron ore is simply iron ore, but the steel maker demands many varieties or grades. Most iron mines are able, by selective mining, to produce more than one grade, and in some cases the output of several mines is combined to form desired grades, each suitable for some special purpose.

There is a small but growing demand for ore of better analysis than the average, for making special quality steels. Beyond doubt the present average analysis, as shown, can be and will be maintained for a long period.

The year 1932 cannot be fitted into any previous picture. With the steel production of the country down to around 20 per cent of capacity, there is ore enough in reserve at the furnaces and Lower Lake docks, carried down in 1930 and 1931, to last for two years at the 20 per cent rate. No ore need go down the Lakes this year, and but little will. Every mine that can close down is idle.

During the last 20 years the production of steel has increased greatly, although, as shown in Table 2, iron-ore production has not. This is largely due to the increased use of scrap metal in the open-hearth process, present practice being estimated to call for about 40 per cent scrap in the charge.

Predictions made 15 to 20 years ago that iron-ore requirements from Lake Superior would rise to 75,000,000 tons or more annually have not been borne out, nor are they likely to be.

The Lake Superior region is beyond doubt equipped with mining and transportation facilities for an annual production of 75,000,000 tons or more, but as far as can now be seen the average future demand is not likely to exceed 60,000,000 tons. At this lower rate the merchantable reserves of the district promise a life of between 20 and 30 years, but back of the merchantable reserves are enormous tonnages of low-grade material, which it has already been demonstrated can be brought up to merchantable grade by various beneficiation processes, such as drying, washing, jigging, tabling, crushing, screening, sorting, flotation, magnetic separation, sintering, and direct reduction. Improvements in such methods are fairly rapid at present, as is evidenced by the fact that in 1930 about 40 per cent of all iron ore shipped from Minnesota had been beneficiated in some way.

These recent years have also seen great improvement in mining methods, outstanding among which are electrification of both open-pit and underground mines, the use of larger electrified shovels on caterpillar mounting and electric haulage in the open pits, and the use of scrapers underground, all of which have been fully described and discussed in the technical press. The importance of these changes is indicated by the fact that in Minnesota today 9000 men produce what formerly required 21,000.

In considering such statistics as those given herein, it should always be kept in mind that the output of iron mines is not free either as to tonnage or grade, but is strictly controlled by the quantity and quality of iron and steel that can be marketed. This has been a strong factor in one important change in the situation. Formerly a large part of the ore was mined and sold by independent operators. More ore was always offered than could be sold. Today by far the greater part is controlled and mined, directly or indirectly, by consuming interests, the steelmakers themselves. The independent market has practically disappeared.

The situation may perhaps be summed up by saying that Lake Superior reserves of raw material are very large and will be competitive even when beneficiation is more of a necessity than at present. Increases in output can readily be met if demanded, at any time. Direct operating costs can probably not be much reduced, but overhead items can and will be lowered, when that becomes essential to meet imported ore at Lower Lake ports.

Coke Ovens

There are at present in the United States, in operation or capable of operation, 13,727 by-product coke ovens, with an aggregate coke-producing capacity of approximately 5,000,000 net tons per month.

From a production of 3,602,000 net tons of furnace coke in by-product coke ovens in August, 1929, production in the United States declined to a low of 958,000 tons of furnace coke in May, 1932, or 27 per cent of the 1929 peak. It follows that unless finished-steel production declines at exactly the same rate as coke production, major problems in flexibility in the use of coke-oven gas for steel-plant heating purposes will arise. Variations in the ratio of coke to finished steel are intensified by the fact that many steel companies turn out plates, sheets, and other finished products on an immediate demand basis and do not average out production to any considerable extent by anticipating future orders.

A development in coke-oven technology that was inaugurated in America toward the end of the last decade is the underfiring of coke ovens with blast-furnace gas. This development coordinates the operation of coke plants with heat application in steel plants and provides flexibility in the use of gas in keeping with abnormal steel-plant operations. An excellent paper on this subject was prepared by Charles R. Meissner, superintendent of the by-product coke plant of the Weirton Steel Company, and Julius H. Strassburger, assistant chief engineer of the Weirton Steel Company, and was presented before the American Iron and Steel Institute at New York on May 19, 1932. The fine results from the use of blast-furnace gas in coke ovens are fully set forth in that paper.

Open-Hearth Plant

In common with all other plants of American industry, the open-hearth plant has been called upon during the current year to supply fractional capacity at costs far below any figures approached during the last 20 years. When it is considered that the ingot capacity of the United States is about 67,500,000 tons and that but 25,000,000 tons was produced in 1931, representing 37 per cent, and but 7,500,000 tons in the first half of 1932, indicating a rate of probably less than 22 per cent for the entire year, it is simple to picture the condition of this division of industry, which, from the enlarged units of the past decade, depends on high production rates for its profitable operation.

The newer plants with furnaces of larger size, other than in one or two exceptional instances, have been badly handicapped in producing the small tonnages ordered to the competitive analyses specified at a cost that would show a profit while the older plants with smaller furnaces have had a considerable advantage from the operating and cost standpoints.

A small plant, a component of one of the larger producers, with 50-ton furnaces, operating without hot metal, has been able to consistently show the lowest ingot cost of any of the plants of this group during the last 20 months. Further, this plant is operating intermittently by crude methods and without any of the more modern appurtenances of operation.

For several months following the beginning of the current depression, most of the steel companies went forward with substantial apparatus and practice research programs that promised much. Since the first of the present year, these programs have been severely curtailed or entirely omitted, with the unfortunate result that little or no research in these directions is now being carried on.

However, several furnaces have been equipped for their automatic reversal from temperature difference, and a few installations of combustion or draft control have been made.

One of the larger companies has been experimenting for a number of months with a method of firing the furnace through the roof, the flame being directed generally downward and toward the backwall. This experiment is being conducted with considerable secrecy, but it is understood that the time of heat is materially shortened with some fuel, and with a considerable metal savings as a result of using the method.

Another method involving the firing of the furnace from four points is being introduced, but it will probably be some time before definite results will be available for comment.

The successful conclusion of either of these methods would mark the first radical change of open-hearth practice since the days of Siemen and Marten.

Research in the metallurgy of the process has been continued in part, particularly in deoxidation, and progress appears to have been made. Progress is of necessity in detail, and no startling developments in the metallurgy are to be expected. The manufacturers of refractories have made marked improvements in these products, and as furnaces are in time operated to the end of their life and rebuilt, it is expected a material lengthening of furnace life will result.

The meetings of the two larger groups of open-hearth superintendents for the exchange of experience and ideas have been well attended during the past year, but in one of these groups at least much time is wasted in a discussion of smaller details that, while usable in a plant here or there, have no general application. However, considerable good comes from these meetings, and the appreciable increase in the number of younger, technically trained men who are working into the superintendencies of these plants augurs well for future progress.

Rolling Mills

While progress and development in the art of rolling steel have necessarily been seriously curtailed due to the extreme inactivity in operations during the past year, the period has nevertheless been marked by the completion of a number of large mill installations and several distinctly new developments.

Notable among the mills completed during the year are the wide-flange beam mill at the Illinois Steel Company, the continuous plate and sheet strip mill at the Inland Steel Company, and the continuous sheet strip mill at the Otis Steel Company. These installations in the aggregate have an additional annual producing capacity of approximately 1,750,000 tons of finished and semi-finished products.

The wide-flange beam mill was designed to produce wide-flange structural shapes from 8 to 36 in. in depth, with flanges from 6 to 15 in. in width, and weighing from 21 to 425 lb per ft. The mill was installed at the South Chicago Works of the Illinois Steel Company. The facilities consist of a 54-in. reversing blooming-mill stand, a 52-in. reversing roughing-mill stand, a 52-in. intermediate reversing stand, and a 52-in. single-pass finishing stand, all motor driven, together with a full complement of soaking pits, mill tables, cranes, reheating furnaces, bloom shear, hot saws, and cooling beds.

The continuous plate and sheet strip mill installed by the Inland Steel Company at Indiana Harbor is designed to roll universal plate from $^3/_{16}$ in. to $^1/_2$ in. in thickness up to 69 in. in width and the usual gages and grades of sheet, hot- and cold-rolled strip in widths from 6 to 66 in. The mill comprises three

slab reheating furnaces and ten four-high mill stands, of which the first stand has 36 × 49 × 96-in. rolls, the next three are universal stands with 25 × 49 × 77-in. rolls, and six finishing stands having 24½ × 49 × 77-in. rolls, together with two scale breakers all separately motor driven. The mill further includes complete finishing equipment consisting of cooling beds, transfers, tables, coilers, levelers, shears, etc., as required for the various classes of product rolled.

The continuous strip mill built and placed in operation by the Otis Steel Company at its Riverside Works in Cleveland, Ohio, is capable of producing plate, strip and strip sheets from 18 gage and heavier in thickness and 24 to 65 in. in width. The mill is placed in direct line of production with an existing 40-in. blooming mill, from which slabs can be rolled direct or reheated and rolled after passing through either of two reheating furnaces, and so arranged as to permit independent operation of either mill when desired. The mill stands are 10 in number, of which the roughing stands are two-high, having 32 × 72-in. rolls, and five are four-high finishing stands, with 20½ × 45 × 72-in. rolls. There are four edging stands in the roughing mill line. All roll stands are individually motor driven. The mill is further equipped with cooling beds, coilers, flying shears, and plate-piling machinery to handle both finished and semi-finished products, either in coils or flat plate. Two stands of cold rolls were also installed, each with 20½ × 52 × 72-in. rolls to cold-finish product from the continuous mills.

The latest development in the hot rolling of flat material and strip has been successfully demonstrated this year by the Cold Metal Process Company on an experimentally equipped Steckel hot mill located at the Brier Hill Plant of the Youngstown Sheet and Tube Company at Youngstown, Ohio.

One mill stand of four-high rolls, having 20-in.-diameter work rolls and 36-in.-diameter backing-up rolls, all 72 in. in length, is used. This stand is served by two elevated hot-coiling furnaces, one spanning the approach table and the other the delivery table, each approximately 14 ft from the center line of the mill stand. Each hot-coiling furnace has a 48-in.-diameter reel made of heat-resisting alloy steel. The purpose of these furnaces is to maintain a rolling temperature in the strip after the slab reaches the thinner gages.

During the demonstrations that took place at various times this year, slabs 3 and 4 in. thick, 9 ft long, and up to 36 in. wide were used as raw material. Each slab, heated to 1600 to 1800 F, was brought from a slab-heating furnace by crane, deposited on the approach table leading to the mill, and run through a scale breaker, equipped with steel brushes to loosen the scale on the top and bottom surfaces. High-pressure water sprays removed the loosened scale. After leaving the scale breaker, the slab enters a set of pinch rolls, which are separately motor driven and which direct it into the four-high mill stand. The 3-in. slabs are given approximately five flatting passes and the 4-in. slabs approximately eleven flatting passes back and forth through the mill, while resting on the roller tables, which are spanned by the hot coiling furnaces. During subsequent passes a set of dogs guides the strip into either of the hot coiling furnaces. The hot coiling furnaces are gas-fired and maintain the strip at rolling temperature throughout its length.

When the rear end of the strip emerges from the mill and is still in the grip of either of the pinch rolls, which are located between each of the hot coiling furnaces and the mill stand, the mill is reversed and the strip returns through the work rolls and the pinch rolls and into the opposite hot coil furnace, where it is again coiled and reheated to the rolling temperature. Reduction continues with the piece being coiled and uncoiled until it is rolled nearly to the desired gage. Following this, the strip, instead of being entered into the coiling furnace after the final pass, is conveyed by a roller table to a cold coiler, where the strip is wound into a finished coil.

Strips have been rolled by this method to approximately 12 and 14 gage and up to 36 in. in width. The maximum rolling speed possible on the experimental mill is about 300 fpm, at which speed 4-in.-thick slabs 9 ft long were converted into strips of about 12 gage in 6 to 6½ min.

The experimental rollings have given good indications as to the commercial possibilities of this type of mill. Increased rolling speeds are possible that, if applied to an actual mill installation, would enable one stand to produce an estimated tonnage of approximately 12,000 tons per month of 36-in.-wide 12-gage strip. While no edging rolls are used on the experimental mill, they are contemplated in the design of any future mill in order to roll more exactly the desired width of product and assist in breaking up the initial furnace scale. The demonstrations have further indicated that a very satisfactory hot-rolled product is possible.

Further development in the modernization of hand-operated sheet and tin mills has continued during the year. Many hand mills have been equipped with automatic mechanical feeders and catchers in arrangements known as the "combination system" or "combination method." These have resulted in increasing the production of the old type of tin and sheet mills to approximately one and one-half and three times their respective original outputs, with savings in labor up to about $5 per ton, together with better quality of product, a higher percentage of prime material, lower scrap loss, and the elimination of hard manual labor.

Electric-Weld Pressure Pipe

The use of electric current for welding purposes has extended into a variety of fields previously dominated by other methods. However, the following remarks are confined almost exclusively to the use of electric current for the manufacture of electric-welded pressure pipe. While mechanical, bedstead, and other small-diameter light wall tubing of a similar nature has been successfully produced by the electric-weld process for a number of years, it is only within the last few years that commercial pressure pipe has been manufactured by this method. The pressure-pipe demand up to the time of the entrance of the electric-weld process had been served by the butt-and-lap-weld and the seamless-hot-process systems, with the lap-weld method gradually disappearing from the picture. The invasion of the pressure-pipe field by the electric-weld process was first directed against line pipe and kindred products, and a number of large and successful installations had been made in various parts of the country of line pipe manufactured by the electric-weld process. Quite recently successful attempts have been made to enter the casing and similar pipe fields, which in recent years had been almost exclusively supplied by the seamless method of pressure-pipe manufacture. There are several different methods employed today in the manufacture of electric-welded pressure pipe, and there probably will be other methods developed, as the process, in the main, is in its infancy, and a great amount of thought is being given the problem, as it apparently offers an opportunity for reducing the costs of manufacture of pressure pipe in a rather wide range of pipe sizes, which is naturally attractive to the pipe manufacturer. Prominent among the methods employed today in the manufacture of electric-welded pressure pipe are, first, the arc method, in which a foreign metal is deposited between the edges of the shaped skelp to form the weld; second, the momentary flash method, in which the slightly separated edges of the formed blank are heated by establishing an arc between them that is maintained for a short period and then forcing the edges together to form the weld, the current being supplied to the work either by induction or by means of elec-

trodes; third, the continuous induction flash method, in which the formed blank is fed continuously forward, with the edges separated, to the welding zone, where the edges are gradually brought together and a continuous short arc is maintained by means of an induced electric current in the pipe blank; fourth, the resistance method, in which the formed blank is fed continuously forward, with the edges separated, to the welding zone, where the edges are gradually brought together and current is allowed to flow across the joint, the current being supplied to the work by means of electrodes and not of sufficient voltage to establish an arc. Line pipe within a certain range of sizes can be produced by any one of these methods at present. The resistance method of electric welding is demonstrating that it is also suitable for the manufacture of casing and similar products, and it is quite possible that the field of some of the other methods may be extended in the future.

The successful manufacture of electric-welded pressure pipe is not dependent alone upon the welding process, as there are a number of other operations that the skelp must go through before being delivered to the welder that are of great importance, such as the preparation of the edges of the skelp, the cleaning of the surfaces of the skelp, and the forming of the blank. The forming of the blank is a particularly difficult operation, and the success obtained at the welder is largely dependent upon this operation. The process of producing acceptable electric-welded pressure pipe is more of a mechanical than an electrical problem, and the actual welding is materially simplified if a properly prepared formed blank is supplied the welder.

Some of the advantages of electric-weld pressure pipe over seamless in particular are, first, lower production costs above $3^1/_2$-in.-outer diameter; second, larger sizes can be produced; third, uniform wall thickness; fourth, ability to produce longer lengths; fifth, more adaptable to the higher carbon and thinner wall piping that appears to be the trend of the trade today.

It is possible with the resistance electric-welded methods, by the employment of only one welder, to produce more than 10 miles of pipe in a 24-hour day. Due to stagnant business conditions, less advancement has been made in the manufacture of electric-weld pressure pipe than would otherwise have been accomplished had conditions been normal, but sufficient progress has been made so that it can be definitely stated that the electric-weld method of pressure-pipe manufacture is here to stay and that in all probability it will decidedly improve its position in the near future.

Refractories

In considering the recent progress made in the field of refractories, attention is drawn to a number of types of brick that have been developed and to modifications that have been made to existing types in order to meet certain definite service requirements in the operation of industrial furnaces.

Included among these are high-alumina brick, those in which the mineral mullite has been developed to a considerably greater extent than in first-quality fire-clay brick, modifications in chrome and magnesite brick, kaolin brick, and special refractories—for example, silicon carbide and fused alumina.

The high-alumina brick type of refractories includes commercial brands containing approximately 50, 60, 70, and 80 per cent alumina. For certain applications they possess very definite advantages over first-quality fire-clay brick that may be stated as follows:

(*a*) They are more refractory, and the refractoriness increases with increased alumina content

(*b*) They offer greater resistance to the erosive action of certain types of slag

(*c*) They do not vitrify at as low temperatures as do first-quality fire-clay brick; this is important, since it results in a lower spalling tendency over high temperature ranges

(*d*) They possess greater strength at high temperature ranges.

High-alumina refractories find some important applications in the steel industry in blast-furnace stoves, in open-hearth checkers, and in water-cooled ports. In many plants in the steam-power field they are standard for the side walls and bridge walls of boiler settings where certain types of slag are to be resisted that more readily react chemically with first-quality fire-clay brick. The alkaline ash from oil fuel is very destructive to fire-clay brick at the extremely high operating temperatures that are commonly employed. High-alumina brick are therefore also used in the settings of high-temperature cracking units in the oil-refining industry. Another economical application for high-alumina brick is found in the glass industry, where they are used in the ports and checkers of glass tanks. Still another application is in the field of non-ferrous metallurgy—for example, in lead softening and refining furnaces. To a very large extent high-alumina brick have supplanted fire-clay brick in the burning zones of rotary kilns used for burning cement, lime, and dolomite.

The mineral mullite has a definite crystalline form. In the pure state it consists of 71.8 per cent alumina and 28.2 per cent silica. A certain amount of mullite is present in hard-burned fire-clay refractories, but the amount is more pronounced in refractories of the high-alumina type. Refractories with an especially high amount of developed mullite are particularly suited for glass tank flux blocks.

Progress has been made in the development of chrome brick of low spalling tendency and greater strength at high temperature. This is of particular value in connection with their use in the open-hearth steel furnace and in various furnaces used in non-ferrous metallurgy.

Magnesite brick of low iron content are finding favor for special applications, particularly where they are exposed to direct heat action. They are becoming used in furnaces employed in the metallurgy of copper and other non-ferrous metals.

Brick made from the high-grade kaolin may be considered as an intermediate between first-quality fire-clay brick and high-alumina brick. They must be burned very hard in order to reduce the residual shrinkage to a minimum.

There has been a wider use of special refractories, such as silicon carbide and fused alumina.

There has been a more general use of high-temperature bonding mortars in all types of refractory construction. Research has helped considerably in the selection of these materials for specific service conditions.

Plastic ramming mixtures have been developed and satisfactorily applied—for example, plastic (thermolith) chrome batch for bottoms in heating furnaces and in slag-tap boiler settings.

In general, the power press has been applied to the manufacture of practically all the various types of refractories, resulting in greater uniformity in size and texture of the product. Continuous tunnel kilns have taken the place of periodic kilns in many plants. However, because of their greater flexibility as to production and degree of firing, periodic kilns are not likely to be entirely replaced.

Steam and Power Generation

During the last year there have been no outstanding developments in blast-furnace and steel-plant steam-power installations, but there has been a constant tendency toward applying the policies inaugurated in recent years.

The trend continues toward greater centralization of steam and power production in units of larger size and greater efficiency,

toward higher steam pressures and more efficient heat cycles, and toward more careful control of combustion.

Boilers. Water walls are becoming more and more popular, due to the immense amount of radiant heat that they are capable of absorbing and the protection that they offer to the refractories in the setting. At least one boiler company is offering a complete steam-generating unit, including boiler, water walls, economizer, and air preheater, conveniently arranged for assembly on a single foundation.

Recent changes in the boiler code now permit welded boiler drums, with the result that riveted drums will probably soon become a thing of the past. Now that suitable X-ray apparatus for testing is available and the art of welding is well understood, there should be no cause for failure of welds when the work is properly executed and thoroughly inspected. The use of welded joints properly annealed should reduce, if not eliminate, the danger from embrittlement by caustic that is always present where riveted construction is used.

The trade has offered one or two new devices for use in connection with superheaters in the past year, such as means for controlling the volume of hot gases passing through the superheater and the design of a superheater suitable for small horizontal return-tubular boilers.

The use of more thoroughly cleaned blast-furnace gas made possible through recent developments in the art of electrical precipitation has been tried out in at least one plant during the past year, with the result that more complete combustion of the gas has been obtained and the boiler outages have been reduced from one cleaning and inspection every 30 days to one every 90 days. The efficiency of the boiler, economizer, and air preheater is also improved due to cleaner heat-absorbing surfaces.

Prime Movers. Further improvement in the efficiency of turbines has been made by refinements in design and the use of heat- and erosion-resistant alloys in the blades. The turbine manufacturers have been very diligent in adapting their designs and in utilizing materials suitable for higher temperatures and pressures, with the result that the upper temperature limits have been increased to approximately 1000 F.

The Morgoil Roll-Neck Bearing

By F. P. DAHLSTROM,[1] WORCESTER, MASS.

For several years a need has existed in two-high hot rolling mills for a rugged, precision-type bearing having a capacity great enough to carry existing loads, and which could be installed in the limited space available, without reduction of neck strength. This paper reviews the experimental work with roller bearings which eventually led to the development of a unit-type oil-lubricated sleeve bearing. A brief review of pertinent laws of bearing design is included, together with their application to rolling-mill problems. Following a description of the Morgoil bearing, some results of field tests are included which show how successfully the bearing has met the conditions imposed upon it.

THE Morgoil bearing was especially designed for application to the necks of steel-mill rolls. It embodies a type of construction in which the bearing assembly is removable as a complete bearing unit. The roll neck itself is not used as the journal, but fits into a sleeve contained in the bearing unit. The outside of the sleeve is accurately ground cylindrical and forms the journal proper. Before describing the mechanical details of the bearing unit, mention will be made of some of the experiments first conducted with other types of high-quality precision bearings, which eventually led to the development of a bearing of similar precision, but having, among other advantages, greater capacity in the limited space available.

Early Roller-Bearing Applications

Roller bearings of various types had been installed in merchant mills, both in Europe and the United States. The roll speeds were comparatively slow, which favored the introduction of a roller bearing, as such a bearing has its greatest capacity at low speeds. In these installations the roll bodies were long and grooved with comparatively deep passes. The rolling pressures therefore could not be higher than the breaking strength of these relatively weak rolls. Lubrication of the roller bearing was also more simple at these lower speeds, as a grease was quite suitable for the purpose.

Problems and Experiences in Attempt to Equip Strip Mills With Roller Bearings

In modern two-high strip mills the length of the rolls is deliberately made short so as to permit carrying extremely high rolling pressures without roll breakage. The ordinary open-type plain roll-neck bearing then remained as the limiting factor in two-high strip-mill rolling. As reliable performance of the roller type of anti-friction bearing in many industrial applications was a demonstrated fact, attempts were made by our company to use roller bearings in solving the two-high strip-mill roll-neck problem, but it was soon apparent that there were several inherent disadvantages.

At the outset it was necessary to use a smaller diameter roll neck in order to provide room for a bearing of suitable capacity. The usual roll-neck diameter is two-thirds the nominal roll diameter, and in order to install a roller bearing, the neck diameter had to be decreased about 20 per cent, which reduced the strength of the standard roll neck about 50 per cent.

The speeds of the finishing stands of strip mills are high; on the smaller sizes as much as 1000 rpm. As the effective capacity of a roller bearing decreases with increase in speed, the best that could be secured, even with the reduced neck size, was a bearing having a capacity barely great enough to carry the expected rolling loads. For the finishing stands of the larger size of strip mills, no roller bearings were submitted which could be guaranteed to successfully carry the expected loads continuously, and the only apparent solution was to install a larger mill and thus provide the necessary room for roller bearings of a suitable capacity.

Experimental work with roller bearings was therefore started on the smallest size of strip mill, and the first installation was made in the field. The power savings were, as anticipated, about 50 per cent. The mill roller, having a more efficient mill at his disposal, took heavier reductions, and the bearings soon failed, possibly because of overload or because of faulty lubrication. It was a question as to whether more installations should be made with bearings or necks that did not have inherently sufficient overload capacity.

The problem of lubrication also presented itself. In the installation referred to, grease was used as a lubricant, and after the bearing failure, the surface of the individual rolls had an appearance much like carbon. It is not known whether the excessive heat was caused by too much or too little lubricant, entrance of scale (with resultant binding), or inherent friction between rollers and retainers, etc., which might have been serious at heavy loads and high speeds.

In later designs it was decided to lubricate such high-speed bearings with oil. This method had the distinct advantage of being able to connect the bearings directly to the existing mill oil-circulating systems, the reliability of which systems had been well demonstrated. Many hours of shop testing showed that an ordinary oil was suitable for the expected normal loads, but it was soon found that the designs of closures which were recommended to retain grease at slow speeds were not satisfactory for retaining oil at high speeds.

From the foregoing, as well as other experiences, it was evident to us that in order to make a successful roll-neck-bearing installation with roller bearings, the following conditions must be met:

(*a*) First-class workmanship should enter into the manufacture of bearing enclosures and assembly of bearing parts, and the maintenance be assigned to responsible workmen

(*b*) The bearing must be kept free from foreign matter

(*c*) Thorough lubrication must be had at all times, as only the combination of cylindrical, or conical, surfaces of

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the rolls in contact with the bearing races approach anti-friction performance. (The friction between rolls and retainers and the friction between the ends of tapered rollers and the roller-confining means are serious limitations, especially at heavy loads and high speeds.)

(*d*) The bearing should not be loaded above its rated capacity.

After a consideration of these conditions, it was decided to investigate the possibilities of a flood-lubricated sleeve-type bearing made with comparable precision and given the same care that was demanded by the roller bearings.

Further Shop Experiments With Roller and Sleeve Bearings

A study of the modern theory of sleeve-bearing design indicated that this type had a greater potential capacity for our conditions than the roller type. This type of bearing could also be installed in existing mills without reduction of neck strength.

A pair of sleeve-type bearings were then made for a 10-in. strip roll, and the combination was installed in a standard housing. The other roll was equipped with roller bearings. The two rolls were then run tightly together with known pressures. Experiments were hardly begun when a neck on the latter roll broke at a bearing load of 42,000 lb at 500 rpm. A new roll of steel was then substituted so as to permit carrying the pressures desired. For a number of days, test runs were made in which the pressures, powers, speeds, oil flow, bearing, and room temperatures, and rise of temperature in the circulating oil were measured. The data in Table 1 are compiled from these tests.

TABLE 1 DATA OBTAINED FROM TEST RUNS, 10-IN. STRIP HOUSING

(Upper roll fitted with sleeve bearings; lower roll fitted with roller bearings)

Rpm	Bearing pressure	Hp[a]	Oil temperatures Inlet, deg F	Oil temperatures Outlet, deg F	Oil circulation, qt. per min. per bearing Sleeve	Oil circulation, qt. per min. per bearing Roller	Remarks
400	10,000	13	...	...	..	..	Broke cast-iron neck; roller bearing mounted; replaced with steel roll
500	42,000	24					
500	12,000	11	110	130	..	..	85 hr total running time
500	20,000	13					
500	28,000	15	110	131	..	..	
500	40,000	15	120	131	..	..	
600	40,000	17	125	152	12	7	
770	33,000	22	130	158	..	..	
50	73,000	4	70	78	..	..	Broke cast iron neck; sleeve bearing mounted after 3-hr run

[a] Includes all bearing losses and rolling friction of mill rolls.

The limit of rolling pressure which could be applied was determined by the strength of the standard size neck on the sleeve bearing. Eventually this pressure was deliberately exceeded, and the neck broke at a load of 73,000 lb at 50 rpm. No damage whatsoever occurred to the bearings.[2]

Fig. 12 shows the broken roll ends. The roll at the right was carried on roller bearings. The disk on the extreme right was cut from the neck which broke at a load of 42,000 lb at 500 rpm. In order to determine the effect of roll speed, the opposite neck was broken in a testing machine, with the bearing mounted in place and with the test load applied to the same point on the bearing enclosure. The neck broke at a static load of 81,000 lb. The fractures at both ends appeared to be similar and the Brinell hardnesses were alike. The calculated neck stress was 22,300 lb per sq in. If the results of further tests were similar to this, we would infer that the effective capacity of a roll neck at 500 rpm would be one-half its static capacity.[3]

The roll at the left fitted the Morgoil bearing and broke at a static load of 158,000 lb. The corresponding neck stress was 25,500 lb per sq in. The opposite end of the roll broke at an applied load of 73,000 lb at 50 rpm, or only 46 per cent of its static capacity. (This should have been higher. The steel roll was badly spalled, however, and severe vibrations were felt in the housing during the run, which probably imposed additional loads on the bearings.)

Our experiments proved conclusively the reliability of the flood-lubricated bearing for this application. It showed that the bearing is really anti-friction, the low frictional losses being in close agreement with calculations.

Some Laws of Sleeve-Bearing Design

A—Intermittently Lubricated Sleeve Bearings. In several handbooks and publications on bearings there may be found the formula pv = a constant, in which p is the bearing pressure in pounds per square inch of projected area and v the surface speed of the journal in feet per minute. The constant was originally 60,000, and in some designs, according to the literature, was as high as 200,000.

If we trace the origin of this formula we find that it was introduced by Dr. Thurston, first president of The American Society of Mechanical Engineers, and the constant he recommended was based upon observations he had made of the "behavior of the journals of the engines of naval steamers in 1862." This formula is used in proportioning the lengths of bearings in mills constructed by our company where such bearings have their oil supply regulated by the ordinary "sight-feed" oiler.

If this formula is used to determine the length of a roll neck 7 in. in diameter on a 10-in. strip roll rotating at 1000 rpm and carrying a rolling load of 50,000 lb, it will be found that its length should be 33 in., instead of the usual dimension, 9 in. The example cited is typical, and one should therefore expect the ordinary open, plain roll-neck bearing to have the short life that it does. If water were not continuously and copiously used, the bearing would promptly melt.

B—Flood Lubricated Sleeve Bearings. When a bearing is continuously lubricated with a suitable oil, the conditions are entirely changed. The laws of hydrodynamic theory of lubrication then apply. One of the early recorded references on the subject of flood lubricated bearings is the paper by Reynolds (1).[4] In the reports and papers presented by the A.S.M.E. Special Research Committee on Lubrication, there may be found a wealth of information and references on this subject. Among those who have presented papers of great value to us in the development of the Morgoil bearing are Kingsbury, Howarth, McKee, Karelitz, Bradford, and Hersey.

Although many discussions have been presented explaining the phenomena of complete lubrication of journal and bearing, perhaps the author's interpretation of this subject will be in

[2] A special gage was developed by the author for measuring rolling pressures. It consisted essentially of a sheet manganin resistor, insulated with sheet mica and placed between steel blocks. These blocks were mounted in a suitable enclosure, and the combination was inserted between the roll-adjusting screw and bearing chock. The resistor formed one arm of a Wheatstone bridge. The resistance of manganin changes proportionally with pressure; therefore the change in resistance of the gage unit, caused by rolling pressure, was measured by a galvanometer graduated to read the rolling pressure directly in pounds.

[3] It is interesting to note in a catalog of a prominent roller-bearing company that the rated capacity of the bearings at 500 rpm is 40 per cent of the static capacity. In the author's opinion the percentages of static capacity at various speeds, as established by the roller-bearing manufacturers, might well be used as a starting point in estimating the variation of roll-neck capacity with speed.

[4] Numbers in parentheses apply to similarly numbered items in the List of References given at the end of the paper.

order at this time, after which the resulting formulas will be applied to the design of roll-neck bearings.

When a bearing is continuously supplied with a sufficient quantity of oil, it appears that the adhesion of the oil to the rotating journal causes the journal to function as a remarkably effective and efficient pump. The journal pulls some of the oil through the region of minimum clearance. The oil which is not drawn through this region must escape at the ends of the bearing. It requires a tremendous squeezing force to do this, and the resulting pressure actually lifts the journal away from the bearing, permitting the oil to separate the two by a film of definite thickness, which can now be fairly accurately calculated (3).

The more viscous the oil, the more pressure will be required to force the oil out at the ends; hence we have a first law of flood-lubricated bearings as follows:

The carrying capacity is proportional to the viscosity, Z, of the oil.[5]

As the speed of the rotating journal is increased, the journal draws in more oil (the average velocity of the oil being about one-half the surface velocity of the journal); consequently, a proportionately greater force is required to squeeze the oil out at the ends of the bearing. From this fact we have a second law for flood lubricated bearings:

The capacity is proportional to the speed of the journal.

For a given speed it is obvious that the amount of oil carried in by the journal is substantially proportional to the axial length of the bearing, thus we have a third law:

The capacity is proportional to the axial length, L, of the bearing.

We can combine these laws into an expression as follows:

$$\text{The bearing capacity in lb } P = \frac{Z \times \text{fpm} \times L}{\text{a number}} \quad \ldots\ldots [1]$$

If p = the bearing pressure in lb per sq in.; D, the journal diameter in in.; N, the rpm, then:

$$P = pDL \quad \ldots\ldots [2]$$

and

$$\text{Fpm} = \frac{ND \times 3.14}{12} \quad \ldots\ldots [3]$$

Substituting these values in Equation [1], we have:

$$pDL = \frac{Z \times ND \times 3.14 \times L}{\text{a number} \times 12} \quad \ldots\ldots [4]$$

or

$$p = \frac{Z \times N \times D \times 3.14 \times L}{12 \times DL \times \text{a number}} = \frac{ZN}{\text{a number}} \quad \ldots\ldots [5]$$

From the foregoing we derive that the load-carrying capacity of a given bearing in lb per sq in. is equal to the viscosity of the oil times the rpm of the journal divided by a number that is established by experience.

The minimum value which this number can have, without rupture of the oil film, depends upon the clearance ratio and the accuracy of finish, as well as the condition of the journal and bearing surfaces. With clearances of from 0.001 to 0.00125 in. per inch of journal diameter and with a high degree of accuracy, McKee (5) has found by experiment that this number can be as low as 5 before failure of the oil film begins.

The experimental work of McKee was conducted with bearings 2 in. in diameter. The author, in connection with the development of the Morgoil bearing, has consistently found that at low speeds and with bearings 8 in. and 11 in. in diameter, this number can be as small as $1^1/_2$ and 1. In practice a value of from 10 to 25 times this may be advisable in order to have a conservative design.

Transposing Equation [5], we obtain

$$\text{Number} = \frac{ZN}{p}$$

The use of the variable ZN/p in formulating laws of lubrication of sleeve-type bearings was first proposed by Hersey. (2) He showed that under certain assumptions the coefficient of friction and relative film thickness was a function of this variable; also that the carrying power or safety factor of a bearing might be defined from its relative film thickness.

In papers by Howarth (4) and Karelitz (3) the expression ZN/p is one of the factors which determines the eccentric position of a rotating journal in its bearing. Karelitz still further proposes that the minimum oil-film thickness (which is easily found when the eccentric position of the journal is known) might well be used in determining the safe carrying capacity of a bearing.

Earlier in this paper it was shown that by the Thurston formula, the length of a roll-neck bearing 7 in. in diameter for a 10-in. strip mill carrying a load of 25,000 lb should be 33 in., instead of the conventional length of, say, 8 in.

We will now calculate the capacity of a bearing 7 in. in diameter by 8 in. long, by the ZN/p method, in order to determine the suitability of a continuously lubricated bearing for this application.

The viscosity Z of the oil in the circulating system will be about 150 centipoises (750 sec Saybolt), and a close approximation at the temperature of the oil film will be 50 centipoises. If a value of 20 for the characteristic number of the bearing is chosen (which from our experience is conservative and would be considered as a reasonable one), we obtain a value of the permissible load in lb per sq in. of:

$$p = \frac{ZN}{20} = \frac{50 \times 1000}{20} = 2500 \text{ lb}$$

The load on each bearing being 25,000 lb, the total projected area should be 25,000/2500, or 10 sq in. The projected area of a bearing 7 in. in diameter by 8 in. long is over five times this amount, which means that the bearing has a factor of safety of over 5. The capacity of such a bearing is therefore far above the breaking strength of any chilled cast-iron or alloy cast-iron roll manufactured today.

To express the capacity of the bearing in another way: With a characteristic number of 20, the bearing could easily carry the full rolling load at speeds as low as 200 rpm.

For a condition considerably different, let us determine the safe carrying capacity of a bearing for a neck of an 18-in. roll at 50 rpm. Such a bearing would have a projected area of 200 sq in., and the viscosity of the oil film would be about 350 centipoises. The characteristic number at this slower speed can well be taken at 10. The bearing will have a safe capacity of $200 \times (ZN/10) = 200 \times (350 \times 50/10) = 350{,}000$ lb. This is equal to the safe carrying capacity of a neck having a diameter of two-thirds the roll diameter, or 12 in.

The coefficient of friction of a flood-lubricated bearing having a clearance ratio of 0.0015 is approximately $0.001 + (0.00004 \times ZN/p)$.

[5] Viscosity is usually specified in seconds Saybolt at 100 F. The seconds Saybolt viscosity is the actual number of seconds required for a standard amount of oil to drain from a calibrated vessel; hence the larger this value in seconds, the more viscous the oil. In bearing calculations the viscosity is denoted by the absolute viscosity in centipoises, which value is approximately one-fifth the Saybolt viscosity.

For the bearing of the 10-in. strip mill noted above, operating at a ZN/p value of 20, the coefficient of friction will be 0.001 + 0.0008 = 0.0018 and the frictional losses at 25,000-lb bearing load will be, per bearing:

$$\frac{25{,}000 \times 0.0018 \times 1000 \times 8 \times 3.14}{33{,}000 \times 12} = 2.85 \text{ hp}$$

or 11.4 hp per stand, which loss is very small when we realize that the losses in the ordinary type of open bearing would be over 20 times this amount.

The coefficient of friction of the bearing 14 in. in diameter for the 18-in. mill will be 0.001 + 0.0004 = 0.0014, and the frictional losses per stand will be:

$$\frac{4 \times 350{,}000 \times 0.0014 \times 50 \times 14 \times 3.14}{33{,}000 \times 12} = 10.9 \text{ hp}$$

which is also negligible, as the losses with the plain open-type bearing would be about 700 hp. In practice, Morgoil bearings run quite cool, as would be expected from these figures. No water is applied to the bearing enclosure, as the small amount of heat developed is readily carried away by the oil.

It has been mentioned that the minimum oil-film thickness might also be used as a criterion for safe bearing capacity. From the paper by Karelitz (3), the minimum oil-film thickness of the high-speed strip-mill bearing operating with a characteristic number (ZN/p) of 20 would be 0.0009 in., which is greater than his recommended conservative value of 0.0007 in. The corresponding film thickness of the 18-in.-mill roll-neck bearing would be, according to the method by Karelitz, 0.0005 in., which we find from mill experience is safe.

A Condition Not Covered by Theory

From the foregoing it is apparent that the rotating journal in a properly designed bearing is at all times separated from the bearing by an oil film of appreciable thickness. No wear takes place. The only friction involved is fluid friction, which is non-destructive, the important effect being to raise the temperature of the oil. As the speed of the rotating journal is reduced, the oil film becomes thinner, and at very slow speeds the film will not be continuous and the coefficient of friction of the bearing will suddenly increase. The actual speed at which this breakdown of the oil film starts to occur is low and will depend upon the accuracy and finish of the journal and bearing. (At a unit bearing load of 3000 lb per sq in. and with a plain mineral oil we have been able to run as low as 1 rpm without disrupting the oil film. The amount of wear which will take place when the journal operates below this critical speed or when the journal is started or stopped will likewise depend upon the accuracy of finish of the parts, also upon the degree of polish of the journal. Fortunately, in hot rolling the bearing starts and stops under practically no load, and the wear in this region is negligible.

Description of Bearing

In considering the design of a suitable sleeve bearing it seemed best not to use the roll neck for the journal, but to provide a sleeve of high-quality steel which would fit on the neck. The outside of the sleeve could then be ground and polished to obtain the most suitable journal surface. This basic arrangement was adopted, and the combination was called the Morgoil roll-neck bearing to distinguish it from any different construction that had been or might be developed.

Fig. 1 shows a cross-section of the working-side top roll bearing mounted in running position on the roll, and Fig. 2 shows the bearing at the drive side. The roll necks are made tapered to permit easy removal or replacement of the bearing units. In Fig. 1, A is the sleeve, B the thrust-bearing ring, and C a locking nut. The sleeve is prevented from creeping by a key, which key also serves to prevent relative rotation of the thrust ring. The locking nut C secures the thrust ring and sleeve in place on the roll neck, and as all of these parts therefore revolve with the roll, they are cross-sectioned in the same general direction. The casing D contains a replaceable bearing liner E which is made of either steel or bronze and has on its inner surface a high-grade babbitt centrifugally cast in place.

The sleeve, and therefore the roll, are held in a fixed endwise position by the two stationary babbitt-faced thrust rings F. A threaded ring G is used to adjust the thrust plates. One-half of the outside surface of the ring is notched, and the thrust bearing is easily adjusted by first screwing up the ring G tightly and then backing it off a few notches.

The oil for the main bearing enters the casing at hole H, after which it travels through the passages indicated by dotted lines and enters the bearing liner at holes J. Pressure relief and drain grooves are provided at the roll end of the bearing liner as shown. These grooves effectively drain the oil from the liner. Oil slingers are provided on the steel sleeve, which in combination with the grooves prevents any escape of oil without the use of auxiliary packing rings. The drained oil is free to flow through the passages at the bottom of the casing, from which it enters a drain pipe, not shown.

In order to prevent water and scale from entering the bearing, the casing is grooved for a circular ring K, which is made of bakelite impregnated material or its equivalent. The ring is backed by springs which force the ring against the end of the roll body and thus effectively prevent the entrance of water. Such a solid ring appears to be much superior to split rings or resilient packing, which had been used with more or less success in connection with our earlier experiments. By using the end of the roll for a sealing surface instead of the neck, the need for split rings, etc., was avoided.

The arrangement in Fig. 1 shows one method of lubricating the thrust bearing. Where there is no need of the neck projecting through the bearing, there is provided an entering pipe in the center of the enclosing cover. A bakelite washer, backed by a spring, prevents wastage of oil and causes it to be delivered into the space K with line pressure, from which space the oil is conveyed by drilled passages directly to the thrust bearing as shown. Where the neck must project through the bearing at the working side of the mill, the lubricant for the thrust bearing is supplied from the same passages as supply the main bearing.

In Fig. 2, which shows the bearing for the drive side of the mill, the end adjacent to the roll is exactly like the corresponding end of the bearing in Fig. 1. The bearing sleeve A is free to float endwise in the bearing liner E. In strip mills the casing D is held endwise by lugs or the equivalent which fasten to the roll housing. In mills having rolls with grooved passes the two casings are connected by means of spacer pipes and tie rods, as shown in Fig. 3. In both types of mills the endwise adjustment of the roll is made entirely at the working side of the mill by means of lugs on the casing shown in Fig. 1, which cooperate with adjusting nuts and studs on the housing.

The bearing in Fig. 1 is readily removable as a unit by first removing the end plate and next unscrewing nut C. A puller is then applied, and after tightening the puller screw the end is struck a hammer blow, which frees the sleeve from the neck. The entire unit may then be withdrawn. The adjustment of the thrust bearing is not affected.

For the bearing in Fig. 2 it is only necessary to remove nut C and the split threaded ring which engages with the nut before applying the puller. The easy removal of the bearing unit is thus seen to be a very important advantage of the Morgoil

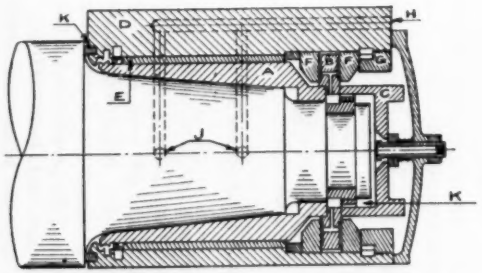


FIG. 1 - CROSS SECTION - UPPER BEARING WORKING SIDE OF MILL

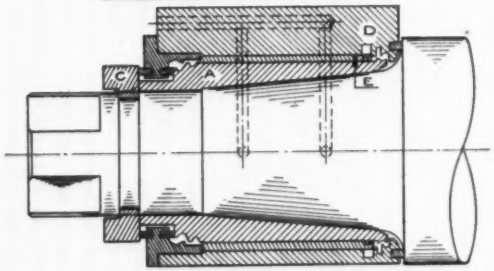


FIG. 2 - CROSS SECTION - UPPER BEARING DRIVE SIDE OF MILL

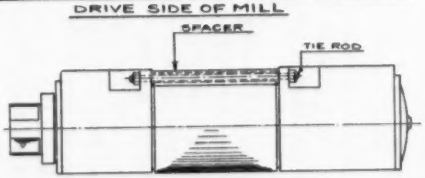


FIG 3 - TIE ROD AND SPACER BETWEEN BEARINGS

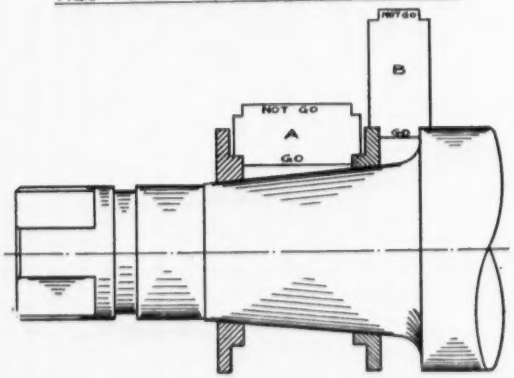


FIG. 4 - METHOD OF GAGING ROLL NECK.

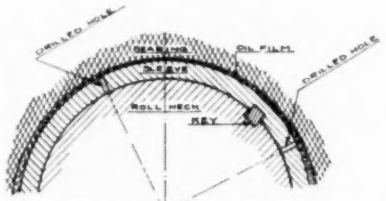


FIG 6 - ROLL NECK AND BEARING PARTS CORRESPONDING TO FIG 5

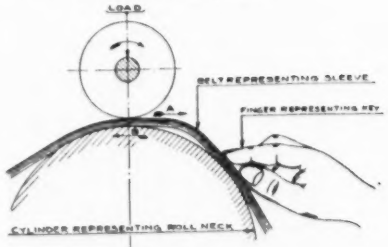


FIG 5 - EFFECT OF CREEP OF BELT

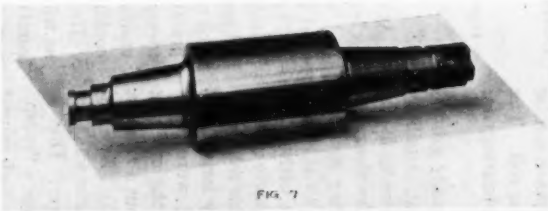


FIG. 7

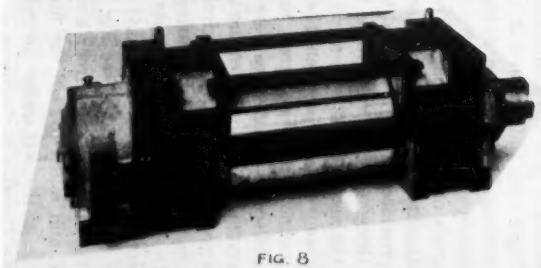


FIG. 8

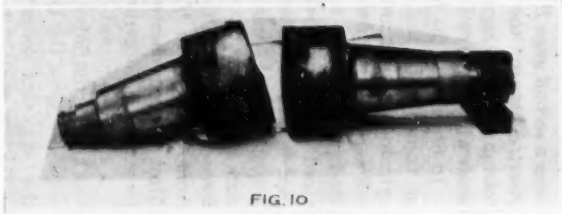


FIG. 10

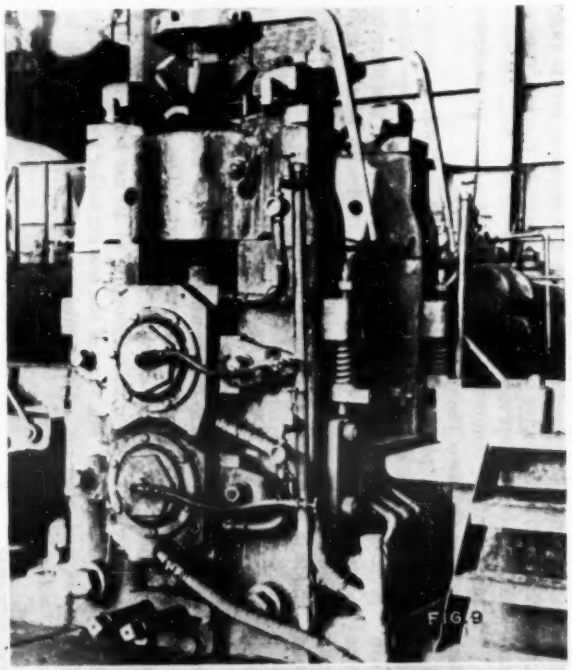


FIG. 9

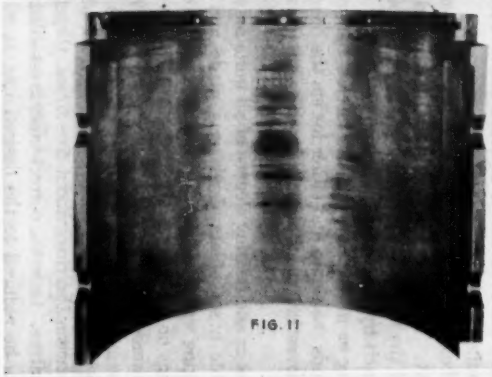


FIG. 11

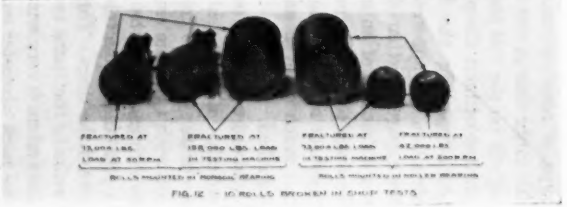


FIG. 12 - 10 ROLLS BROKEN IN SHOP TESTS

type of construction. Due to a special method of lubricating the inner side of the sleeve, it is not necessary to force the unit tightly on the roll neck, and its removal is often easily accomplished without the puller fixture.

In replacing the bearing, further advantages of this construction are apparent. If, when the bearing is partly slid on the tapered portion of the neck, it is advisable to lower the bearing a fraction of an inch by the crane, no harm whatsoever can be done to the bearing or its running surfaces in case the crane operator misunderstands the signal and either lowers the bearing too much or even moves it in a direction opposite to the one signalled.

In order that the necks and bearing sleeves of all units may be strictly interchangeable a set of standards was made and an accurate system of gaging was established. Fig. 4 shows part of a roll with two gage rings on the neck. The distance between the two rings checks the amount of taper. A feeler gage *A* having a plus and minus tolerance is used to measure the distance between the rings. The errors in taper magnify the corresponding errors in the space between the rings about five times, therefore the system is very sensitive. Once the correct taper is established, the neck diameter can be reduced to the desired size, at which time a second gage *B* having a plus and minus tolerance can be inserted between the larger ring and the end of the roll. The tolerance on this gage is comparatively large, as the bearings are made with ample end clearances.

In order that the barrels of the rolls may be readily ground or turned with the standard roll shop equipment, auxiliary cylindrical sleeves are provided having a tapered bore to fit the roll neck. It was first thought necessary to force the sleeves of the bearing unit on to the neck very tightly as no key was provided to prevent relative rotation between the two. With moderate rolling pressures no "creep" occurred, but with heavy pressures it did occur, and scores and grooves were made in both neck and sleeve. Attempts were made to measure this rotational force which caused the creep. Keys of successively larger size, from $^1/_4$ in. up to $^3/_4$ in. (in a 14-in. roll bearing) were tried. Most of them sheared. In order to demonstrate the creeping force, a simple experiment was made which pointed out a remedy.

A belt was placed on a wood cylinder (see Fig. 5). One end was held by the finger, the other end being free; a roller was then pressed against the belt and rolled toward the finger. Due to the temporary elongation of the belt under the roller a "creep" of the belt resulted in the direction of the arrow *A*, which caused the belt to bulge as shown in Fig. 5. As the belt was comparatively stiff, an appreciable resisting force was required at the finger tip in order to prevent the belt from slipping forward at that point. As the roll advanced toward the finger, the belt had to flatten; consequently, the finger would have to yield or else the belt would have to slip backward in the direction of the arrow *B*. It was evident that if a lubricant were applied between the belt and the cylinder, the belt would slip back easily; consequently, only a small force would be required at the finger tip to prevent the belt from creeping forward.

Fig. 6 shows the parts of the sleeve, key, oil film, and bearing as represented by the parts in Fig. 5. The bearing pressure is transmitted through the oil film (Fig. 6), which pressure is represented by the roller in Fig. 5. This compresses the sleeve slightly, and because of rotation there is a tendency to creep. Several holes were drilled radially in the sleeve, allowing the oil-film pressure to force oil through the holes and thus lubricate the surface of the sleeve and roll neck. The force of creep could not build up above the frictional resistance between the sleeve and neck, and as this was now greatly reduced, it was found that a $^1/_4$-in. key was not sheared even under the heaviest loads obtainable in our experiments.

It is now our practice to tighten the retaining nut *C* (Figs. 2 and 3) a moderate amount, which permits the lubricant to do its work and preserve the tapered surface, and, as before mentioned, the removal of the bearing unit is often accomplished without the use of the standard puller fixture.

Fig. 7 is a photograph of a 10-in. roll. It will be noticed that the neck diameter where it joins the roll is large (about seven-tenths the nominal roll diameter) and that the fillet has a large radius. Fig. 8 shows a 10-in. roll with bearing units mounted in place on the roll. Fig. 9 shows a bearing installation made in an existing 18-in. open-top housing.

Results of Field Experience

Pressure and power readings were taken on the finishing stand of a 14-in. strip mill in which the life of the ordinary plain open-type neck bearing was very short. The highest rolling pressures and electrical loads were obtained when finishing 6 in. by 0.038 in., with a reduction of 10 per cent, as will be seen in Table 2. The wear of the bearing under these conditions was such that roll adjustments were necessary after each bar was rolled.

TABLE 2 DATA OBTAINED FROM TEST RUNS, 14-IN. STAND, PLAIN BEARINGS

Item	Finishing size, in.	Per cent reduction	Amperes at 600 volts	Rolling pressure	Delivery speed, ft per min.	Approx. temp., deg F
A.....	$9^1/_2 \times 0.062$	6	1000	180,000	1150	1400
B.....	11×0.105	7	1000	191,000	1150	1500
C.....	6×0.038	10	1350	210,000	1750	1300

At the time the tests were made there were, to our knowledge, no bearings of any type which could be made to fit in this housing and which could be guaranteed to carry the maximum rolling load (210,000 lb) continuously at a speed of 480 rpm.

The first set of 14-in. Morgoil bearings were then made and installed in this stand for test. They were designed to carry a rolling load 50 per cent in excess of the maximum measured. The neck stress under these conditions would be as high as good practice would permit—12,000 lb per sq in. The bearings performed in every way as expected. During this test the bearing casings were only comfortably warm to the hand and the only cooling water used was on the roll barrel proper.

Heavier reductions were taken with less power consumption, as will be evident by comparing items A and B in Table 3 with similar items in Table 2. It will be seen that reductions 50 per cent higher were carried which resulted in a 35 per cent increase in rolling pressure. The power consumption, however, was from 40 to 50 per cent less. A study of several readings showed that the power saving for given reductions, etc. was 70 per cent. This is more than can be reasonably expected for an entire strip mill. A fair value would be 50 per cent.

TABLE 3 DATA FROM TESTS 14-IN. STAND, MORGOIL BEARINGS

Item	Finishing size, in.	Per cent reduction	Amperes at 600 volts	Rolling pressure	Delivery speed, ft per min	Approx. temp., deg F
A.....	11×0.062	9	425	242,000	1270	1400
B.....	11×0.105	11	600	265,000	1250	1500
C.....	....	..	700	300,000	....	....
D.....	....	..	800	[a]	....	....

[a] Roll broke at load estimated at 340,000 lb as current was 800 amp.

The data for item C were taken when a few bars, colder than normal, were rolled. As will be seen from Table 3, one of the rolls finally broke. The electrical load was 800 amp, according to a recording ammeter in the motor circuit. No pressure gages were in the stand when the breakage occurred, but as the rolling pressure and electrical load were substantially proportional, a fair estimate of the rolling pressure at the time would be 340,000 lb.

A second piece passed through the rolls before the break

was noticed, but owing to the low frictional resistance of the bearing, the two half-rolls held together and no outward sign of bearing distress was evident. The bearings were later examined and found to be in perfect condition. Fig. 10 is a photograph of the broken roll. The roll was a sound one with a normal depth of chill. Fig. 11 is an unretouched photograph of one of the bearing liners which was installed with the roll in Fig. 10. The photograph was taken after the roll broke and shows the perfect condition of the liner; in fact, the bearings were not even "run in." The dark areas in the center are high spots. This condition is typical of all the other bearings.

As a further example of the stamina of this design of bearing, an 18-in. roll (in the housing shown in Fig. 9) was broken when carrying an estimated load of 1,000,000 lb, with no damage to the bearings. The normal maximum load on this stand is approximately 800,000 lb and occurs when rolling a 22-in. slab from 2 in. to 1 in. in thickness.

Power Savings

The power savings possible with flood-lubricated bearings are very high. In the finishing stand of a strip mill the savings, as noted, have been as high as 70 per cent. This is probably the maximum. A fair average for the rolling-mill industry would be approximately 40 per cent.

The coefficients of bearing friction in billet and sheet bar mills and in the roughing stands of merchant and rod mills have been measured by us and found to vary from 5 to 20 per cent. A fair average would be 10 per cent, which is 25 times the coefficient of friction of a flood-lubricated bearing installed in mills of this type. This means that practically all (24/25) of the power lost in roll-neck friction can be saved by the use of flood-lubricated bearings. The percentage of the total power consumed in bearing friction in such mills probably varies from 30 to 50 per cent, and both the roller and flood-lubricated bearing will save practically all of this wasted power.

In high-speed stands the coefficient of friction of a flood-lubricated bearing operating at a ZN/p of, say, 100, is one-half of 1 per cent, while the average coefficient of friction with the open-type plain bearing is about 5 per cent. The flood-lubricated bearing will therefore save nine-tenths of the power lost in friction. This will be at least 50 per cent of the total in strip mills and about 30 per cent in rod mills.

Lubricants

Any lubricant used continuously in a steel mill sooner or later becomes contaminated with water from condensation, or it may at times enter directly. Iron oxide in the form of fine scale is always present, together with rust which forms on the inside of all steel enclosing parts.

A most desirable property of a steel-mill lubricant, therefore, is its ability to separate from water. So far as the author knows, the only lubricant that has this property is a highly refined mineral oil. Compounded oils and substances which may be added to such an oil may improve its lubricating properties (increase its oiliness, etc.), but the stability of the lubricant will then be destroyed and the resultant emulsions which may form cannot be tolerated. Even the best of straight mineral oils, which have the desired water-separating qualities when new, may undergo quite a decided change with use; therefore the problem of removing water is always an interesting one.

The flood-lubricated, sleeve-type bearing successfully carries heavy rolling pressures in the restricted space available, using as a lubricant straight mineral oils, which at the present time function best in circulating systems.

Conclusions

The Morgoil type of roll-neck bearing has a very high load capacity, generally far in excess of the roll-neck strength. The neck diameter at the point of maximum stress is about seven-tenths the nominal roll diameter.

The bearing can be removed easily from the roll neck as a unit without affecting any adjustments or exposing the bearing surfaces.

A straight mineral oil of proper viscosity is entirely suitable as a lubricant. Such an oil has the highest stability in the presence of water, which is not true of compounded oils or lubricants. This makes it possible to lubricate the bearings from existing circulating systems the reliability of which is well known.

Maximum power savings are obtained, as the bearing runs on an oil film of definite thickness.

There is no appreciable wear, the only roll adjustments required being due to pass wear.

The Morgoil roll-neck bearing was developed only after thorough investigation and experiments with other types of existing precision bearings had shown the need of such a construction. It has successfully met the severe conditions that are peculiar to metal rolling mills, and therefore has a definite place in this field.

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2 Hersey, M. D., "The Laws of Lubrication of Horizontal Journal Bearings," *Journal*, Washington Academy Sciences, vol. 4, 1914, pp. 542–552.

3 Karelitz, G. B., "Performance of Oil Ring Bearings," Trans. A.S.M.E., 1930, APM-52-5, pp. 57–70.

4 Howarth, H. A. S., "Journal Running Positions," Trans. A.S.M.E., APM-51-3, 1929, pp. 21–23.

5 McKee, S. A., "The Effect of Running-In on Journal Bearing Performance," *Mech. Eng.*, vol. 49, 1927, pp. 1335–1340; vol. 50, 1927, pp. 528–533.

Discussion

W. R. Clark.[6] The writer has long been interested in the development of reliable roll-neck bearings having low coefficients of friction. Pressures are encountered in the cold rolling of brass on 20-in.-diameter rolling mills with necks about 15 in. in diameter as high as 500,000 lb per roll neck at roll speeds of from 15 to 25 rpm. Some rough determinations of roll-neck friction for this type of rolling indicated that when a good grade of graphite grease was used on the roll necks, with plenty of cooling water, discharged directly on the neck itself, a coefficient of friction of 0.013 to 0.016 was obtained. Another rough test on an 18-in. roll, running at approximately 60 rpm, with a bearing diameter of approximately 13 in. and load per bearing of from 200,000 to 300,000 lb, indicated a coefficient of friction of from 0.05 to 0.06. These bearings obviously ran warmer than those having the lower speed and lower coefficient of friction.

In cold rolling thin-gage material it is well known by rollers that if the roll necks heat up, the gage of the material rolled is affected, due to the increase in diameter of the necks and the rolls near the necks, so that rolling to accurate gages made it necessary to limit the speed of rolls and the pressures between them to points where the bearing temperature could be maintained substantially constant.

Some years ago the writer went so far as to build a set of flood-oil-lubricated bearings for an 18-in. mill, but owing to inadequate methods of sealing and difficulties with taking end thrusts, these bearings were discarded. Since that time we have followed the

[6] General Works Manager, Bridgeport Brass Company, Bridgeport, Conn. Mem. A.S.M.E.

developments in anti-friction bearing design with the belief that sooner or later reliable bearings with low coefficients of friction would be available for two-high rolling mills.

Several years ago we discussed the matter with Mr. H. A. S. Howarth, and his company offered the design of a precision oil-film-lubricated bearing for a 12-in. rolling mill. Mr. Howarth convinced us at the time that it would be perfectly possible to design a satisfactory flood-lubricated bearing broadly of the type which Mr. Dahlstrom has described and that it should work successfully on a rolling mill, if proper oil were supplied and maintained at proper temperatures and if the seals were adequate to keep the bearing free from roll scale and cooling water.

There was another consideration involved, however, in the 12-in. mill, and that was the probability of starting the mill at times with the bearings preloaded. In cold rolling thin strip the pressures developed between the rolls may run anywhere from 30,000 to 100,000 lb per inch of width of the strip rolled. The elastic spring of a cold-rolling mill plus the elastic deformation of the rolls, due to bending and due to actual flattening at the line of contact with each other or with metal passing through them, may be two or three times greater than the thickness of the metal being rolled. In this case the reduction of gage in rolling is accomplished by the additional spring of the mill and rolls caused by entering the strip into the bite of the rolls, which have been already running together in elastic contact. The rolling-mill bearings therefore may be preloaded with no material in the rolls up to 75 or 80 per cent of the load which they carry while rolling.

If a mill, set up to roll thin-gage material, is shut down for any reason and started up again, it is likely that most of the oil will be squeezed out from between the bearings, and the first fraction of a revolution in starting such a mill might be made with a metal-to-metal contact of journal and bearing. This contact would continue until enough oil were drawn into the bearing to reestablish the normal oil film. Variation in the temperature of the body of a roll would have its effect on the temperature of the roll neck, which in turn would tend to change the oil-film clearance between the neck and its journal.

Roller bearings solved the particular problem of rolling in which we were interested on our 12-in. mills. These mills have been increased in speed from 60 to 120 rpm and carry loads up to 125,000 lb per bearing while rolling thin strip. The gage variation with roller bearings is less at the higher speed today than it formerly was at the lower speed with the old grease-lubricated plain bearing. The power consumption per ton of material rolled has been reduced 45 to 60 per cent.

The author, with his bearing, has gone a step beyond the field which may be served by roller bearings. He has developed a bearing which is suitable for heavy pressures at high speeds and at the same time one that can be built within the limits of a rolling-mill housing. It is well known that the carrying capacity of a roller bearing decreases as the revolutions per minute increase, and if the paper is read correctly, there are indications that the author's bearing works even better at high speeds than at low speeds.

He has described an excellently designed bearing and also has solved the problem of roll changing in a very nice way. The writer would like to have him discuss the question of starting up a mill with preloaded bearings and also the effect of changing roll temperature on the diameter of the journals and whether or not this change would adversely affect the oil-film clearance space allowed in the journal.

The permissible variation in oil temperature, which in turn affects the viscosity of the oil and the thickness of oil film, must be considered in the application of these bearings to rolling mills.

G. S. Warren.[7] This company has had a number of roller-bearing cold-roll strip mills working in a satisfactory manner for several years. On the two-high finishing cold-roll mills the roller bearings were not strong enough to make the reductions necessary, so that we had to install the plain babbitted bearings. We had also tried out roller bearings on the two-high hot strip mills and the results were a great improvement over the old type of brass and babbitted bearings, but the roller bearings were not strong enough to stand up under the heavy loads required to roll strip steel, and therefore the roller bearings had to be given up for this type of mill.

The company about four years ago installed a continuous hot strip mill with five stands of two-high roughing mills with four edging mills and six stands of two-high finishing mills. This mill was to roll strip up to 10 in. wide and from 0.035 in. thick in the narrow sizes to 0.065 in. thick in the wider sizes, and in lengths up to 1500 ft. The mill would do this on all sizes except on the light gages, where it was found impossible to hold the gage due to the brass and babbitted bearings. They tried out a greasing system on these bearings, which helped some, but did not correct the trouble.

When the Morgan Construction Company showed the design of the Morgoil bearings, these were installed on the last pass of the finishing mill, which has a delivery speed of from 1200 to 2400 fpm.

The trial proved a great success, as due to the fact that the necks of the rolls were made larger in the Morgoil bearings the rolls were much stronger and there was no wear in the bearings and the gage was uniform in cross-section and from end to end of the strip. The power required for this mill with the Morgoil bearings was from 50 to 60 per cent of the power required for the brass and babbitt bearings. The roller, who was watching the ammeter on the motor which drives this mill, finally loaded the mill up so that the ammeter was reading about the same as it would on the brass and babbitt bearings, and then one of the rolls broke. When the roll broke, he was reducing the cross-section of the steel about twice as much as he would have done with the brass and babbitt bearings.

This ended the test, as at this time the Sharon Steel Hoop Company replaced the six two-high mills with four four-high roller-bearing mills in order to roll strip up to 22 in. wide instead of a maximum width of 13 in., which was possible on the two-high mills.

The 22-in.-wide strip nearly doubled the load on the roughing mill, and so for a further trial a set of Morgoil bearings were installed in the No. 2 roughing mill, where the reduction was about 49 per cent. The bearings have been in operation for about six months. On a recent inspection of one of the bearings it was evident that there was practically no wear on the bearings, as at least 50 per cent of the surface of the babbitt still showed the tool marks.

The Morgoil bearing has made it possible to double the capacity of a rolling mill without any change to the drive; it has reduced the power required to roll steel 40 to 50 per cent. It has reduced the cost of lubrication, as the oil is in a closed system. Greasing systems on the other types of bearings make some savings, but still most of the grease is washed into the sewer.

S. M. Weckstein.[8] As to the development of the Morgoil bearing, there are a number of points on which it would be interesting to have the author's comments.

The design of the neck as shown permits of ideal construction to give maximum strength in bending. It would be interesting

[7] Chief Engineer, Sharon Steel Hoop Company, Sharon, Pa.

[8] Industrial Equipment Engineer, Timken Roller Bearing Company, Canton, Ohio. Jun. A.S.M.E.

to know the taper which is used on this neck and the diameter of the neck under the split threaded ring which is inserted in the neck. These dimensions will have an appreciable effect on the strength of the neck in torsion. In the case of a continuous mill the design would undoubtedly be satisfactory, but in the case of looping-type mills where three or more stands may be in line, the necks are apt to be weak. This has been proved conclusively on roller-bearing-equipped mills where due to the straight neck used the diameter under the split threaded ring was much greater than would be possible in the case of the tapered neck. In spite of this several necks have been twisted off when rolling was done in several of the stands at the same time. This occurred even though a saving in power of over 50 per cent was shown when the same mill was operated with plain-bearing-equipped rolls.

It appears from the design that pass adjustment is made through the plain thrust bearing. It is interesting to know just how close this adjustment can be made, since it depends on the oil film in the bearing. As regards the mechanical means of making this pass adjustment it is assumed that the methods now used with tapered roller bearings are being followed.

The type of closure used on the Morgoil bearing has been used for quite a number of years in connection with tapered roller bearings and has proved to be very successful both with grease and oil lubrication.

Very low coefficients of friction are claimed for the Morgoil bearings. It is interesting to know what arrangements are being made on both continuous and looping mills to take care of the backlash in the rolls, especially on bar mills.

The author has given no information on the capacity of the thrust bearings and on the methods of determining the thrust on various types of mills. This information appears to be quite important in the design of the bearing.

Since this bearing is designed primarily for use where roller bearings could not be applied because of lack of vertical height, it brings up the subject of two-high hot sheet mills. On these mills the roll bodies are heated and the necks assume temperatures as high as 400 to 500 F. Will it be possible to apply these bearings on mills of this type?

Author's Closure

Referring to Mr. Clark's questions, there are three conditions which might be considered when starting up a mill with preloaded bearings. Considering first the power aspect, it is a fact that the torque required to overcome the starting friction of a flood-lubricated bearing is much greater than the torque required to overcome running friction. The latter obviously is negligible; the former depends upon the elapsed time of shutdown and the smoothness of journal finish. As a matter of information, we have determined the torque required to overcome starting friction of flood-lubricated bearings under various preloaded conditions. To obtain such data it is necessary to first apply a small amount of current to the driving motor, and then gradually increase the amount until the motor starts.

The data thus secured are of interest and may be of value in some special application, but in most cases need not be considered. The reason is apparent when we realize how many mills are now easily started with the ordinary open type of plain bearing. In the usual mill construction there is lost motion in roll spindle couplings etc. Such play permits the driving motor to rotate a few degrees before all slack is taken up, and the stored energy, together with slight shock of impact, permits comparatively easy starting of a preloaded mill. We have found by tests that by applying, say, one-third full-load motor current, a preloaded mill is started without any difficulty. This is true in general for both two-high and four-high mills. In practically all cases there is enough friction between working and backing up rolls to easily start the latter rolls. In cold-rolling mills where the stock is lubricated, there is a possibility of a high polish being formed on the rolls which would cause slipping between the rolls at starting. To prevent such a condition, provision can be made in the bearing for the introduction of oil under high pressure. The necessary pump for this purpose is arranged to start automatically before current is applied to the main driving motor, and then stopped automatically as soon as the main motor starts. We have found from tests that such an arrangement reduces the starting friction to a very small value.

The second factor to be considered might be the wear on preloaded bearings when starting. Here, again, it is well to recall conditions that exist in mills with open-type bearings. The necks in these mills are none too smooth, but nevertheless if by some magic power we could have prevented all wear in such open-type bearings, while running, our problems would have been solved long ago, and there would have been no demand for a precision bearing. After studying many mill bearings, the author finds that most of the wear with the ordinary plain bearing occurs at the higher speeds. It is his firm conviction that with an enclosed type of flood-lubricated bearing, such as he has described, wear at starting will be negligible. From tests we have found that the oil film is reestablished within a fraction of a revolution after starting.

The third and perhaps the most important consideration is the effect of variations of oil-film thickness upon the gage of finished strip. As a starting point the author first investigated the variation of gage with speed in several stands of cold-strip mills equipped with roller bearings. Measurements were made of the thickness along a continuous length of strip while the mill was alternately run at high and low speed. It was conclusively determined that the material was heavier in gage when rolled at high speed than it was when rolled at low speed. The existence of a time effect was thus shown. As a further proof of this time effect it was found that the rolls actually "Brinelled" the stock when at rest. From these experiments we can state that a greater pressure is required to reduce the thickness of strip at high speed, and because of inevitable spring of the mill this results in a slightly heavier gage. The Morgoil bearing automatically tends to compensate for this added spring of the mill as the thickness of the oil film increases with speed. This naturally permits the rolling of a more uniform product.

The effect of change in roll temperature, and consequently change in neck and oil temperature, is to change the viscosity of the oil. The amount of change is a minimum, as the bearings are generously supplied with oil the temperature of which can easily be kept between desired limits. The change in minimum oil-film thickness is very small—about 0.0002 in. with the normal temperature changes. This change is negligible when compared with the changes in neck or roll diameter which now occur in mills with either plain or roller bearings.

The effect of moderate changes in total clearance, because of expansion, is also negligible as, within limits, the actual film thickness at the point of nearest approach, i.e., minimum oil-film thickness, is practically independent of the clearance. This we have determined by experiments with bearings having varying clearances.

Mr. Weckstein's questions regarding the sizes of the taper necks for the Morgoil bearing are best answered by reference to Fig. 13, which has been made accurately to scale from actual drawings. It will be seen that the strength of the neck near the roll is very much greater in the case of the Morgoil bearing than in the case of the tapered roller bearing, and even at the outer end the diameter under the split ring is still substantially larger.

The Morgoil bearing was developed primarily to meet pressing

needs which existed in strip mills. Since the introduction of this bearing the field of application has been, and is, continually broadening. The Belgian type of mill referred to by Mr. Weckstein is not built by our company for various reasons. Where

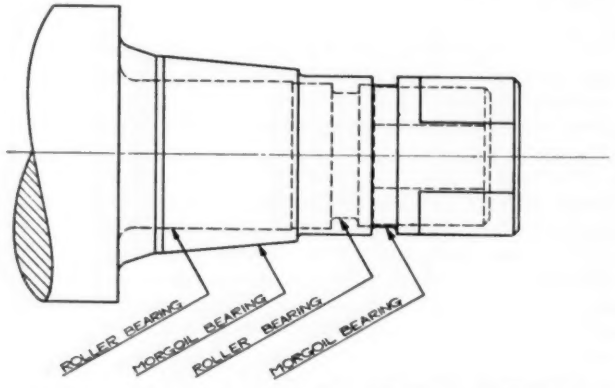


FIG. 13 COMPARISON OF NECKS FOR MORGOIL AND ROLLER BEARINGS, 14-IN. ROLL

looping is desirable, the individually driven staggered arrangement is considered much superior to the Belgian type. It is a fact that to date we have not had any inquiries asking for the adaptation of Morgoil bearings to this particular design of looping-type mill.

Regarding the accuracy of pass adjustment, the principles involved are exactly the same as those in adjustments for thickness of material, and these principles have been verified in practical experience in the field. Regarding the mechanical means of making this pass adjustment, there are several methods which have not been referred to in the paper. Some are external to the bearing; others are internal—that is, an integral part of the bearing. Where the Morgoil bearing has replaced the roller type, it is obvious that the same methods of pass adjustment have been retained for the sake of interchangeability.

Regarding back lash, we installed brakes in some of our early experiments, but removed them later as we found that they were not required. Possibly a superior coupling condition accounts for this result.

Regarding the capacity of thrust bearings, we at first attempted to secure a roller type of thrust bearing to be mounted in the same casing with the sleeve-type bearing. We soon found, however, that their capacity, at the required mill speeds, was very low, being only about 20 per cent of their static capacity. We then decided to adopt the centrifugally lubricated plain-disk thrust bearing in our initial installation. There were two things which we did not know at that time; one being the actual thrust in the mill and the other the capacity of such a bearing. It is a fact that we have not had any thrust-bearing failures to date. Where the thrust loads are known to be very severe, we have since taken extra precaution to put into practice the principles so clearly demonstrated by Professor Kingsbury and others. As the thrust-bearing surface can actually be considered a development of cylindrical bearing surfaces, the principles described in the paper relating to such bearings can be applied to the design of thrust bearings.

Since the paper was presented, our company has agreed to tackle the problem of rolling structural members without depending upon collars in the rolls for taking end thrusts. In this case these thrust loads have been quite accurately determined, and we feel that the time will soon come when the Morgoil bearing will do as much for the structural rolling mill as it has done for the strip mill.

Regarding two-high hot sheet mills, it is a question what the future will bring. Conditions are rapidly changing in this industry, and if a demand is evidenced, a suitable bearing can undoubtedly be supplied.

IS-55-3

Recent Developments in Merchant-Bar Mills

By G. L. FISK,[1] PITTSBURGH, PA.

After first discussing trade demands on merchant-mill products, the author points out the value of research and records in the making and rolling of steel. He then proceeds to discuss the factors affecting the quality of product in the merchant mill, and the items to be considered when selecting the type of merchant mill for a given range of products in a given plant. He next takes up the use and arrangement of vertical mills and the application of roller bearings to mill rolls, in case of existing and new mills, discussing also the economic and practical considerations involved. He then gives particulars of a new transfer-table design, and points out the uses and advantages of electric-driven flying shears. Finally, cooling-bed requirements to meet the needs of various merchant-mill products are then considered, as well as cooling-bed design and operation, objections to the double cooling bed, and double cold shearing from a single bed.

RAPID industrial progress in fields of mass production has been the indirect cause for corresponding developments in the art of rolling steel. Consumers are no longer satisfied with products of a more or less generally accepted standard, but demand merchant bars suited to specific and individual requirements. The steel manufacturers have found it to their advantage to provide extensive research departments not only to develop and determine the best grades of steel for various purposes, but also to observe their behavior in the subsequent manufacturing processes. This has enabled the steel manufacturers to trace to their sources such causes of troubles and rejections as can be remedied in the making and rolling of their products. These statements may be summarized as the tendency in recent years to achieve economy by giving special attention to the making and rolling of steels so as to lessen the expense of subsequent manufacturing operations.

To meet the exacting demands of the trade, merchant mills today are called upon to roll sections true to shape and size, within extremely small tolerances, which are free from injurious surface defects and are of homogeneous and definite grain structure. Although the fulfilment of these requirements depends on suitable mill arrangement and equipment and on good mill practice, it should be understood that the merchant-mill operation is no remedy for steel poorly made or which has been mistreated in prior rolling operations.

In case of unsatisfactory merchant-mill products, it is frequently a matter of contention whether the trouble originated in the steel-making department, in the bloom- or billet-rolling process, or in the finishing mill. Adequate records should be kept to show the complete history of the various steel heats. From such records the causes of defective mill products can be determined with certainty and mistakes can be avoided. Concerns which have engaged in this kind of research have been well repaid for their efforts, both in current economy and by being less dependent on the personal knowledge and experience of individuals. It is rather surprising that this has not been done to a greater extent, in view of the vast economic gain that accrues from definite knowledge as to how to make, heat, and roll in manufacturing a given product in a given plant. Steel plants differ materially in one part or another of their layouts and equipment, and often it is not safe to judge the experience in one plant by that in another, particularly in the case of merchant-mill products.

To confine ourselves to merchant mills, let us assume that billets of proper quality are available for rolling into finished bars. Good merchant-mill design and practice then demands that particular attention be given to the heating of the billets, the elimination of scale, the avoidance during the rolling process of stretching of the material and of severe twisting and scratching, the rolling to accurate shape and size, the controlling of the finishing temperature, the avoidance of scratching or other marking of the bars after leaving the mill rolls and during the cooling process, the handling of the hot bars to insure straightness of the product, and the controlling of the rate of cooling of the rolled bars. In addition the mill must be capable of rolling at high speeds and with a minimum crew and a minimum of delays and repairs in order to achieve the desired economy.

It is evident that all these requirements cannot readily be met, since some of them will react unfavorably upon others. For instance, a mill designed primarily for high-speed rolling, and consequently for large output, may, for this very reason, suffer as far as the quality of its product is concerned. A low-first-cost high-tonnage mill may prove very expensive to operate on a variety of relatively small orders. A mill somewhat higher in first cost and requiring a larger operating crew may be the more economical. In other words, each mill must be designed with particular consideration of the conditions in the plant where it is to be built and of the requirements of the trade which it is to serve.

Types of Mills and Their Selection

There are two types of merchant mills of particular interest to the modern steel plant, namely, the straightaway continuous mill and the semi-continuous mill. The selection of type of mill should be made on the bases of quality and kind of products, diversity of products, relative sizes of orders, number of mills in the plant, and ultimate cost of products.

We shall first consider the straightaway mill, of which Fig. 1 illustrates a recent installation. In the last several roll stands the pairs of rolls are alternately vertical and horizontal in order to avoid twisting of the material, and loopers are provided for vertical and horizontal looping so as to avoid stretching of the material between stands. One or more such mills may be justified in a large steel plant where the tonnage requirements for merchant bars demand several mills or where a large part of the mill products is suited for continuous rolling. In a plant of this character, the range of products assigned to each mill will be relatively limited, and the orders for the straightaway mill can be selected on the assumption that small orders, complicated sections, and sections requiring special attention can be diverted to other mills in the plant.

The straightaway mill has the advantage of low first cost and low operating cost. It is particularly well adapted for rolling of strip, skelp, flats, and angles, but also permits the rolling of other plain sections, including merchant bars. On the other

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

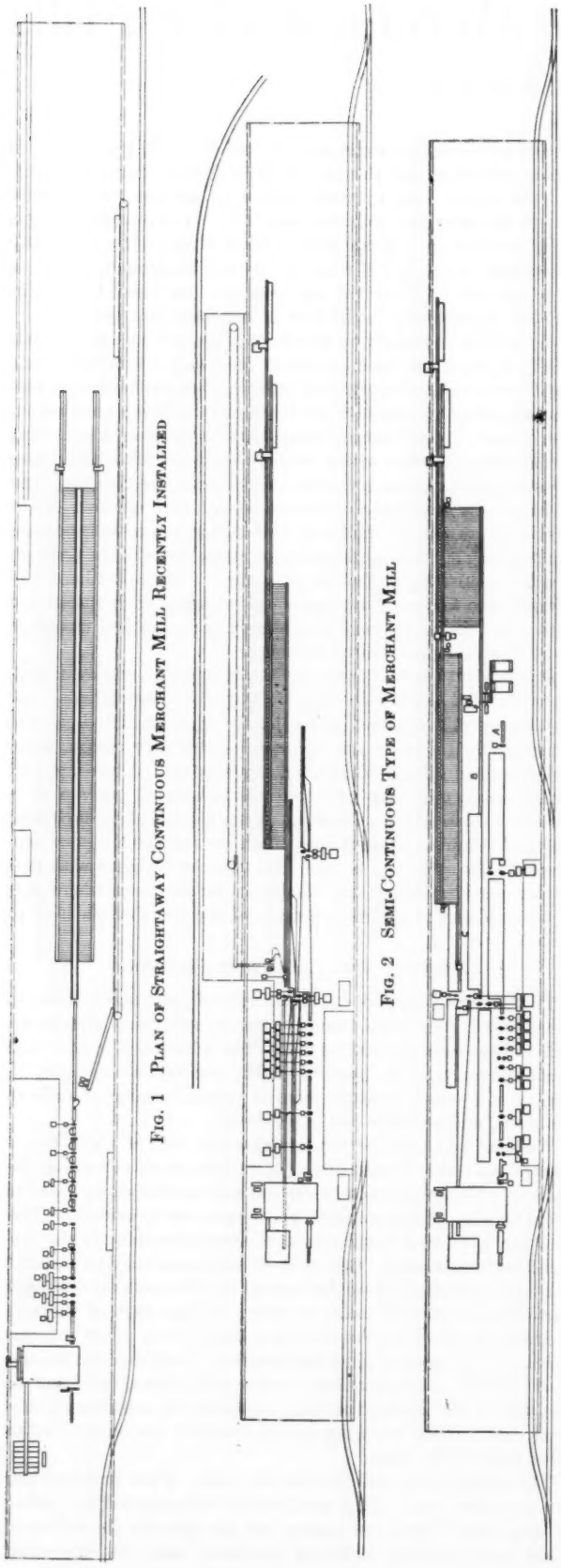


FIG. 1 PLAN OF STRAIGHTAWAY CONTINUOUS MERCHANT MILL RECENTLY INSTALLED

FIG. 2 SEMI-CONTINUOUS TYPE OF MERCHANT MILL

FIG. 3 ANOTHER RECENT INSTALLATION OF A MERCHANT MILL OF THE SEMI-CONTINUOUS TYPE

hand, it is by no means suited to, nor economical for, the conditions met with in the average merchant-bar plant.

To illustrate the shortcomings of the straightaway mill when used strictly as a merchant-bar mill, it is well to consider the proceedings in rolling a merchant round, for instance, as compared to the rolling of strip sections for which the continuous mill is so well fitted. The strip product must be accurate as to gage and width, but the thickness delivered by intermediary roll stands during the rolling process is comparatively unimportant. If the section delivered by one pair of rolls is somewhat thicker or thinner than expected, the speed of this pair of rolls can be adjusted accordingly and the discrepancy will automatically be compensated for in the succeeding pair of rolls. The mill rolls can therefore be set beforehand, and the successful progress of the bar through the mill becomes chiefly a matter of roll-speed control to maintain loops in the material between the several stands, and adjustment for gage in the last stands. The continuous mill is ideal for this condition. The width of the strip is readily controlled by edging rolls.

On the other hand, the ability to deliver a round bar true to shape and size from the finishing stand is dependent upon the delivery of an exact section from the next preceding or leading stand of the mill, and this section in turn is dependent upon the delivery of sections true to shape and section area from earlier stands. To set up a mill for the rolling of merchant bars requires numerous section adjustments involving several roll stands, and considerable skill is necessary upon the part of the operator. In a continuous mill the product of section area and delivery speed for the several roll stands must, by necessity, be a constant. Any adjustment in section area must therefore be accompanied by corresponding change in roll speed as in the case of the strip, previously referred to. But a change in section area in one stand may necessitate section adjustments of various magnitudes in several other stands, and the speeds of their respective rolls must also be changed accordingly. At the same time the roll speed must always be controlled, irrespective of any section changes, so as to maintain loops between the roll stands—this in order to avoid stretching of material, at least in the finishing end of the mill. The greater the number of continuous stands in a straightaway mill, the more these difficulties are aggravated. In other words, when using such a mill for rolling merchant bars it is subject to roll-speed control, just as in the case of the ordinary continuous mill for strip or flat stock, but the operator must also contend with additional and superimposed speed changes incidental to section adjustments.

Since it is admittedly a difficult and exacting art to produce high-grade merchant bars, even under the most favorable conditions, the additional burden imposed upon the roller in the case of the straightaway merchant mill is certainly not desirable. Although it is theoretically possible to attain as accurate a product as that of the semi-continuous mill, the chances of doing so are fewer. To set up a mill for any particular section is bound to involve considerable loss of time, as well as loss of steel in the form of scrap. In the case of often-repeated changes of section, these unfavorable factors will naturally adversely affect the economy of mill operations, and, in case of small and varied orders, the cost of rolling may become prohibitive.

Another disadvantage of the straightaway mill is the limited opportunity it affords for the making and breaking of scale to obtain satisfactory bar surface. Scale is formed on the bar by cooling in air, and in the ordinary close-coupled continuous mill the necessary time factor to produce the desired result is lacking.

Still another handicap of the continuous mill is the necessity for looping, which limits it to sections permitting such treatment.

The chief advantages of the straightaway mill, apart from those already referred to, are the feasibility of high delivery

speeds, i.e., a high rate of production, and incidentally the ability to roll to smaller section areas than in the case of the semi-continuous type of mill.

Figs. 2 and 3 illustrate recent installations of the semi-continuous-type merchant mill. Each has a straightaway roughing train with several stands arranged for continuous rolling. The intermediary and finishing stands of these mills are arranged to permit the material, in process of rolling, to run free after each roll pass, and transfer tables serve to deliver the bar from pass to pass.

Under these conditions the passes of the intermediary and finishing stands are entirely independent of one another, as far as speed is concerned, and the roller is free to adjust the passes individually as suits his requirements in obtaining an accurate product. The continuous passes of the roughing train are comparatively few in number, and for that reason are easily adjustable to the required accuracy.

With the freedom of action thus obtained the roller can quickly change from one section to another, and is generally bound to produce more accurate products with less scrap loss. This type of mill is also well adapted to the rolling of intricate sections not suited for looping. The transfer tables can be used to delay the bar in process, in order to control the scaling of the bar as well as the finishing temperature of the product. Furthermore the rolling speed of the intermediary and finishing trains is independent of that of the roughing train. Small sections, which must be looped or repeated through the intermediary or finishing roll stands to conserve heat, can for that reason be finished at the temperature best suited to each particular product. These are matters of considerable importance, as affecting the surface and the structure of the product.

To summarize: The semi-continuous mill is the better for high-quality products, also when a great diversity of products must be rolled on one mill, or when the tonnages of the individual sizes rolled are comparatively small. In other words, such mills are more generally applicable to conditions as they exist in the trade today. The straightaway mill has its place in large steel plants, where other mills are available for specialties and jobbing orders; that is to say, where the work can be selected to suit the mill.

Vertical Mills

The reduction of a square billet into a square bar of smaller section, which is the usual function of the merchant-mill roughing train, is accomplished by reducing first one axis of the bar and then the other, and by repeating this process until the square of desired size is attained. For this reason the continuous roughing trains shown in Figs. 1 and 2, with all rolls horizontal, necessitate the use of guides to twist the bar between some of their roll stands. Unless the stands are spaced sufficiently far apart to minimize the severity of the twist, the structure of the material may be harmed in the operation or the guides may scratch its surface. A fair portion of the rejected product from the ordinary merchant mill is traceable to the twist guides.

The vertical mills with overhung rolls, indicated in the roughing train of Fig. 3, are principally intended for edging work on flat material, and the first one also serves for the breaking of scale. Edging mills of this type, with one or two roll passes at or near the pass elevation of the horizontal rolls, are also usable, and have been used to advantage, in roughing trains, to avoid twisting, when the passes required in the vertical rolls are few in number.

In case of small sections rolled at high speed, the use of twist guides conduces to difficulties and scrap loss. The twisting and guiding of a small bar necessitates small guide clearances, and any slight resistance in the path of a small-sized bar traveling at high speed may cause a cobble. The finishing end of the straightaway mill shown in Fig. 1, in contrast to the finishing ends of the semi-continuous mills in Figs. 2 and 3, would require twist guides if all its rolls were horizontal. Furthermore, since looping is required, for reasons already stated, it would be necessary to loop twisted sections between some of the roll stands. These difficulties are avoided by the use of vertical mills of recently improved design.

The needs of the finishing end of the continuous merchant mill call for vertical mills with full-sized rolls, supported in bearings at both ends, and with a full complement of roll passes, just as would be provided in the case of horizontal rolls. The rolls must be adjustable up or down to bring any pass to the elevation of the pass line of the preceding and succeeding horizontal rolls. The mill must be capable of sidewise adjustment to bring any pass in the vertical rolls into alignment with any pass line of the horizontal rolls. Furthermore it must have means for accurate roll adjustment to pass opening, and for moving one of the rolls longitudinally with respect to the other. Another requisite is that roll changing should be as easy as in the case of horizontal rolls. The mill should be readily accessible to the operator and should not extend beyond the other roll stands, particularly on the working side of the mill, where it may in that way be an obstruction to operations and visibility in the mill as a whole.

A vertical mill, designed with these principles in mind, is shown in three views in Fig. 4. Roller bearings appear to be essential in this mill, in that they can be enclosed so as to avoid any possibility of scale getting to the bearings. Each roll is suspended from its top bearing and can be removed individually with its bearing boxes, merely by releasing the spring tension holding it in place, and by loosening and moving a clamp. It is also possible to remove both rolls, together with a specially designed roll housing *A*, by removing the cap *B* and loosening the clamping screws *C*. The mill proper is accessible on all sides, having its driving mechanism located in a pit below the floor level. Notwithstanding this arrangement, the pinion housing is removable as a unit, and any machinery part may be easily reached in case of repairs. One of the principles of the design, however, has been to provide against wear and replacements of parts by the use of roller bearings and high-grade materials throughout.

Roller Bearings Vs. Plain Bearings for Mill Rolls

One of the most interesting recent developments is the application of roller bearings to merchant-mill rolls. Attempts have been made, with more or less success, to substitute roller bearings on the rolls of existing mills, but such proceedings usually involve serious difficulties. Roller bearings take up more space than plain roll-neck bearings, making it necessary to reduce the diameter of the roll neck to compensate for this difference, or a good portion thereof. Since the roll neck is frequently the weakest part of the roll, it goes without saying that the substitution of roller bearings for plain bearings on existing mills is most frequently not practicable. Even when it can be done without seriously weakening the rolls, the housing design, originally intended for plain bearings, does not readily lend itself to a practical solution of the roller-bearing application. Roll changes and roll adjustments become cumbersome, and the change-over to roller bearings, principally because of the necessary and expensive alterations to existing roll equipment, is too costly to warrant serious consideration.

In the case of a new mill, where the designer has full freedom, the picture is a different one and conditions may justify the use of roller bearings on the roll necks, provided full advantage be taken of the opportunities incidental to the use of such bearings.

The roller bearing does not show any wear as compared with the plain mill bearing. This fact can and should be taken advantage of, to save time in aligning and setting the rolls to ap-

proximately correct position. Furthermore, as soon as discrepancies in longitudinal and radial roll directions have been ascertained, from a trial bar or otherwise, it is a relatively simple matter, if the roller-bearing application has been designed with this purpose in view, to make the exact adjustments to the required pass opening. The roller bearing therefore offers the opportunity of prompt and exact roll adjustments by aid of indicators, and largely without the cut-and-try process incidental to the use of plain bearings. When the rolls have been adjusted to the correct position they will stay in this position, thanks to the absence of bearing wear. Thus the operator has only to consider the wear in the pass itself during the subsequent rolling operation. This of course is of decided importance, and an advantage which will be readily appreciated.

In case of a new mill, the additional space required for roller bearings as compared with plain bearings can be provided by increasing the roll diameter correspondingly, so that what would be a 14-in. mill, for example, in the case of plain bearings, will become a mill with 18-in.-diameter rolls when using roller bearings on the roll necks. This will naturally increase the roll cost, and also the cost of the mill machinery to some extent. The necessity of grinding the roll necks to exact size is another unavoidable expense connected with the use of roller bearings.

On the other hand, the power consumption with roller bearings may be reduced from 25 to 50 per cent, depending upon the rolling speed and the shape and nature of the material rolled. To offset the high cost of rolls and mechanical equipment, the reduced motor cost with roller bearings deserves serious consideration. The tendency in modern merchant mills is toward individual motor drives and direct current.

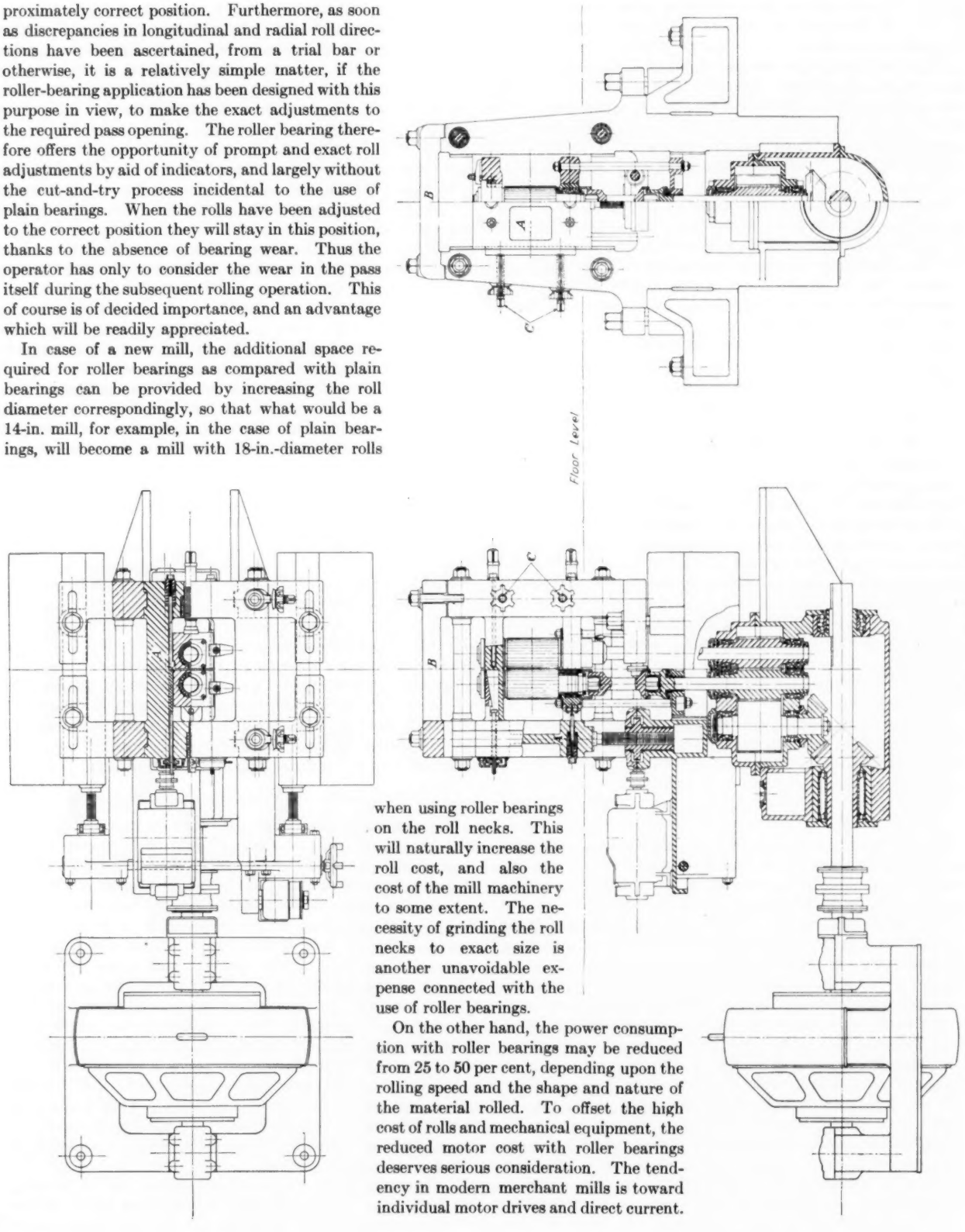


FIG. 4 VERTICAL MILL

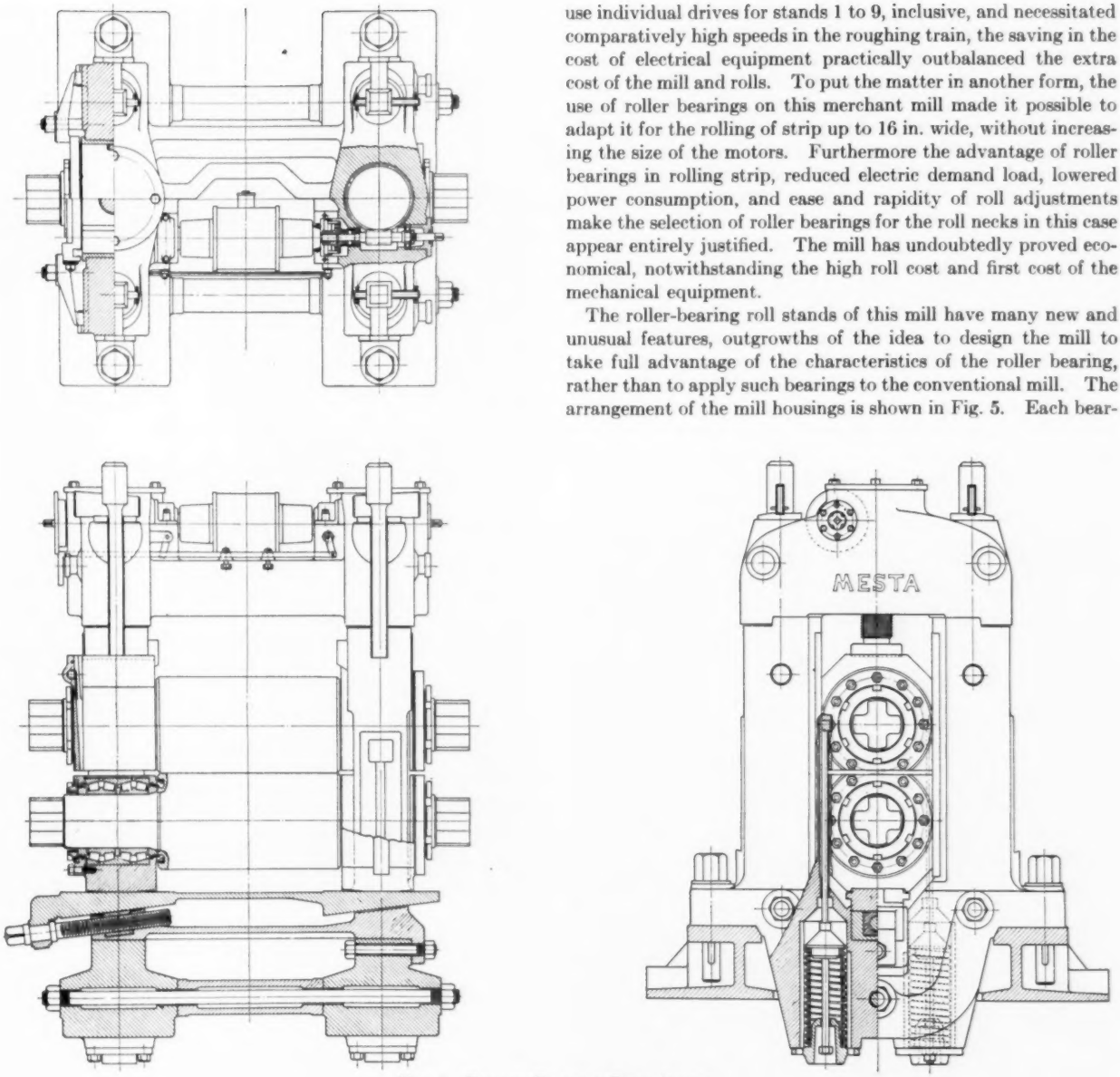


FIG. 5 ROLLER-BEARING ROLL STAND

Any saving in size of motors, therefore, may mean a substantial reduction in the first cost of the electrical equipment.

It is yet too early to predict to what extent roll-neck roller bearings will invade the merchant-mill field, but considerations may exist, apart from those already mentioned, which favor roller bearings. A notable example is the mill recently installed and shown in Fig. 3, which is equipped with roller bearings on the roll necks of stands 1 to 12, inclusive. These stands deliver to two outlets, marked *A* and *B*.

One of the chief problems here was to provide for exceptional diversity of product. Nominally a merchant mill delivering to outlet marked *C*, which covers the range from $^3/_4$-in. up to 3-in. rounds and corresponding sections, the outlet marked *B* is provided to cover rounds from $2^1/_4$ in. up to $4^1/_2$ in. diameter and corresponding sections. Should the market for such products not permit satisfactory mill operation, the outlet marked *A* is usable in rolling strip and flats up to 16 in. wide.

Thanks to these unusual demands, which made it imperative to use individual drives for stands 1 to 9, inclusive, and necessitated comparatively high speeds in the roughing train, the saving in the cost of electrical equipment practically outbalanced the extra cost of the mill and rolls. To put the matter in another form, the use of roller bearings on this merchant mill made it possible to adapt it for the rolling of strip up to 16 in. wide, without increasing the size of the motors. Furthermore the advantage of roller bearings in rolling strip, reduced electric demand load, lowered power consumption, and ease and rapidity of roll adjustments make the selection of roller bearings for the roll necks in this case appear entirely justified. The mill has undoubtedly proved economical, notwithstanding the high roll cost and first cost of the mechanical equipment.

The roller-bearing roll stands of this mill have many new and unusual features, outgrowths of the idea to design the mill to take full advantage of the characteristics of the roller bearing, rather than to apply such bearings to the conventional mill. The arrangement of the mill housings is shown in Fig. 5. Each bearing box, with roller bearings complete, is built in the form of a dirt-proof unit, which is slipped on or off the roll neck as one piece. Each roll has provisions for locking the inner bearing sleeves to its roll necks, to prevent bearing-box movement on the neck in the longitudinal direction, except to the extent of the running clearances of the roller bearings. At the same time, the roll is free to revolve, together with the inner bearing sleeves, while the bearing boxes are held against rotation in the housings of the roll stand.

The bearing box for the free end of the roll is provided with flanges by means of which it may be clamped in the ordinary manner to the roll housing; whereas the bearing box at the driven end of the roll has no flanges, but is free to move with the roll longitudinally in the housing to take care of roll expansion. The bearing boxes of the bottom roll rest on a casting with a horizontal top surface but movable up and down, by wedge adjustment, to bring the bottom roll to the desired pass position. With this arrangement the bottom roll always stays horizontal, although

adjustable to any required elevation. The top roll and its bearing boxes are balanced by means of spring cartridges located in the base of each roll housing. The housing cap is made unusually heavy, for the purpose of overcoming the spring tension of the top-roll balance, so that it can be set and wedged in place with minimum delay, and without giving consideration to the position of the screwdown screws. The adjustment of the top roll is effected against the tension of the roll balance by means of a motor-driven screwdown. The motor is permanently geared to one of the screws and to a dial indicating the screw position. A hand-operated clutch throws the other screw in or out of engagement with the same motor. It will be understood quite readily that by this arrangement either end of the top roll can be adjusted up or down to any exact dimension readable on the dial. The alignment of the rolls in the longitudinal direction for a given pass is accomplished by means of a small adjusting shaft on the top bearing box on the open side of the mill. An index on this shaft is graduated in 0.005-in. divisions and permits longitudinal adjustments of the top roll with respect to the bottom roll within a total movement of $^1/_4$ in.

It will be noted that the bearing boxes are mounted on the rolls prior to their insertion into the roll housings. The roll assemblies are lowered into the housings in the customary manner, and are then clamped into position as already described. To exchange rolls, the guides are taken away, the cap is removed, the clamp bolts are loosened, and both rolls with their bearing boxes are then free for removal, with one lift, by an overhead crane. The new rolls are stacked in pairs and also inserted into the mill with one crane lift.

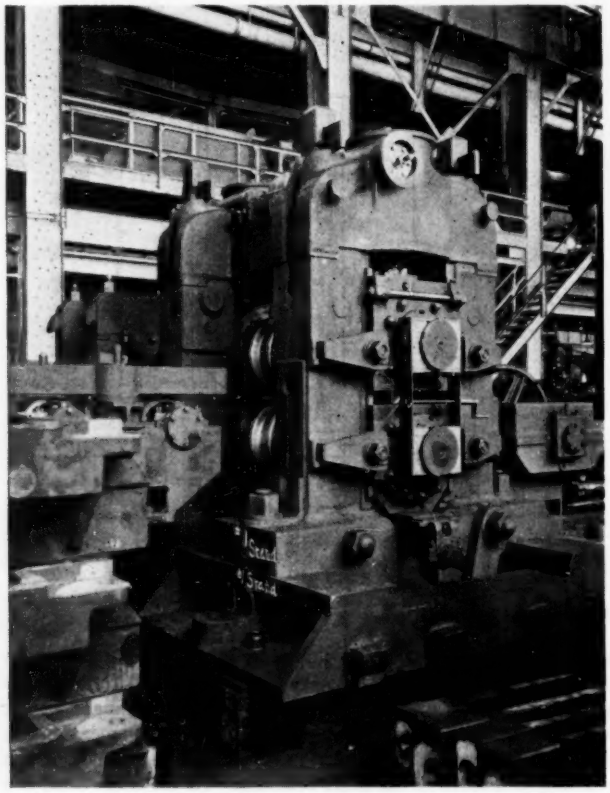


FIG. 6 VIEW OF ROLL STAND SHOWN IN FIG. 5

With the roll housings properly designed, it appears that roll changing in mills with roller bearings can be carried out as easily as in mills with plain bearings. Roll adjustments are easier and more definite with roller bearings, and a roll set-up can be maintained indefinitely, except for adjustments to compensate for pass wear.

A sufficient number of spare bearing boxes, with bearings complete, must be provided in the case of the roller-bearing mill, to allow mounting of bearings on rolls as they are scheduled for use. This work can thus be done prior to the actual roll-changing operation, unnecessary delays can be avoided, and the cost and time of roll changing can be made to compare favorably with these items for the plain-bearing mill. The additional investment in roller bearings should be offset to a large extent by wear and replacements of plain metallic bearings, since the roller bearing is extremely long-lived when it has ample capacity with which to avoid overloading. At the same time, the adoption in recent years of lignum-vitae and non-metallic compositions as materials for plain roll-neck bearings has in many instances very materially reduced the bearing wear and the cost of bearing maintenance.

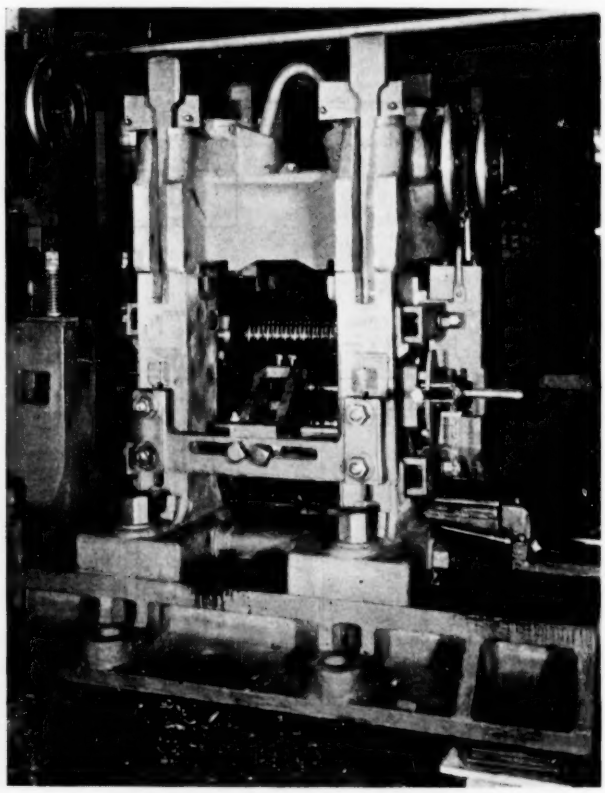


FIG. 7 ROLL STAND WITH PLAIN ROLL-NECK BEARINGS AND HAND-OPERATED SCREWDOWNS

Fig. 6 is reproduced from a photograph of the roller-bearing roll stand shown in Fig. 5. For comparison, an up-to-date roll stand with plain roll-neck bearings and hand-operated screwdowns is illustrated in Fig. 7.

TRANSFER TABLES

The Y-shaped transfer tables shown at the intermediary and finishing roll stands of the mill illustrated in Fig. 2 are of notable construction. Each of these tables serves to receive the bar as it is run out from one stand, transfer it sidewise, and deliver it in the opposite direction to the succeeding stand.

Tables for this purpose have in the past been made with skewed rollers, whereas the tables here referred to have all the rollers parallel with the axes of the mill rolls. The rollers nearest the

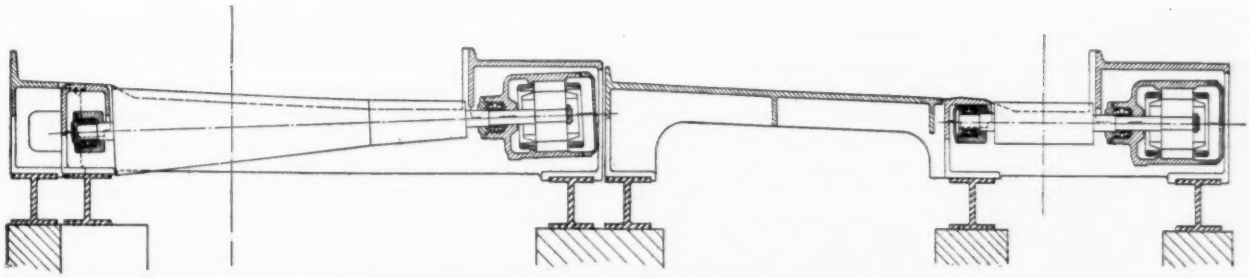


FIG. 8 CROSS-SECTION THROUGH TWO ADJOINING TRANSFER TABLES

mill are tapered and of such length as to cover the distance between the particular roll stands the table is serving. The length of the tapered rollers decreases with increasing distance from the mill rolls to form a triangular-shaped table section, which is followed by a table section of narrow, uniform width with straight rollers. A cross-section through two adjoining tables is shown in Fig. 8, the section being so taken as to show the longest tapered roller in one table and one of the straight, short rollers in the other.

Each of the tapered rollers has two tapers, a more pronounced one at the receiving end and a less pronounced one at the delivery end. The width of the less pronounced taper is uniform for all the rollers, and the diameter at the center of the slight taper is equal to the diameter of the straight rollers. The slightly conical portions of the tapered rollers are set horizontal and level with the straight rollers, thus causing the tapered rollers to be mounted with sloping axes, but all on a uniform slope. Apron plates provided between the rollers make a continuous and practically level floor, suitable for the transfer of bars, but also adapted for use as a looping floor. All the rollers of the table are driven by individual motors, operated from a single electric control. When operated in either direction, all the rollers run at the same, but variable, r.p.m.

The transfer table is controlled in the following manner: The table rollers are operated in the direction of bar delivery, with the straight rollers running at, or slightly above, the bar-delivery speed. Immediately the entire bar is delivered to the table the rollers are reversed, but a timing relay is provided in the electric control to cause the straight rollers of the table to reverse a second or so later than the tapered rollers. During this time interval the bar in transit will be partly supported on straight rollers running in one direction and partly on tapered rollers running in the opposite direction. The bar, seeking the path of least resistance, will promptly move toward the small ends of the tapered rollers. By this time both the straight and the tapered rollers will be operating in the same direction to deliver it to succeeding mill rolls.

FLYING SHEARS

Electrically driven flying shears have been developed in recent years for the cutting of bars up to 4 in. square. Such shears may be used for dividing or cropping bars during their progress through the mill. If it is desired to crop the front end, or rear end, or both ends of the bar after its passage through a given roll stand, load relays can be provided in connection with the motor for this stand to do such cropping automatically. Electric time relays furnish the means for controlling the timing of the cropping operation, i.e., the length of crop.

A recent development is the use of a flying shear following the finishing stand of a merchant mill, not only to divide the bars into cooling-bed lengths but also to cut test pieces from the ends of bars on their way to the cooling bed. Such a shear used in mill layouts, Figs. 2 and 3, is illustrated in Fig. 9. It is of the rotary type, and with a bar running through the shear can be accelerated to bar speed by the time the knives begin the cutting operation. It was formerly necessary to divide the bar into cooling-bed lengths farther back in the mill, where the bar speed was low enough to permit cutting with the flying shear equipment then available. The cutting of test pieces by a flying shear close to the finishing stand permits the roller to gage the bar without going to the cooling bed. Except for small-sized bars, which could be cut with hand-operated snapper shears, it was formerly necessary to gage the mill products at the cooling bed, often an awkward operation and one involving considerable loss of time.

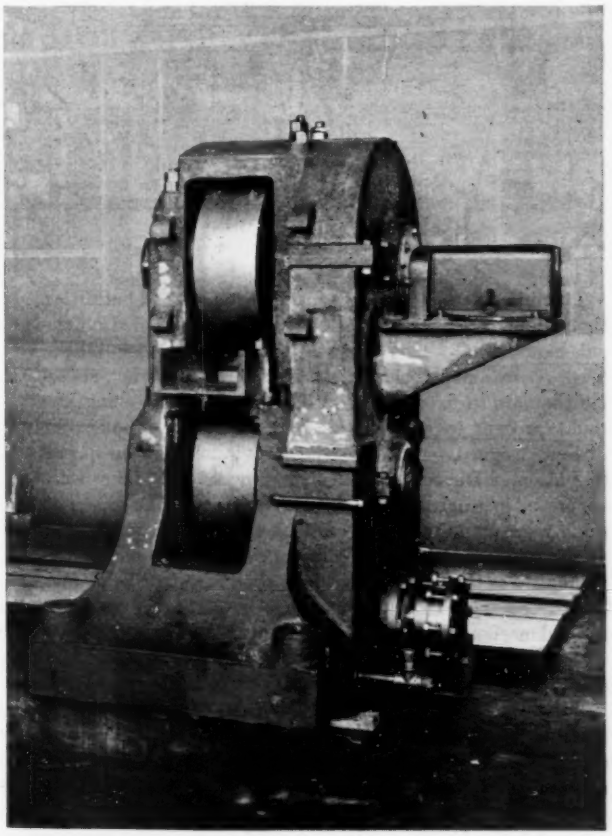


FIG. 9 FLYING SHEAR

COOLING BEDS

In days of old, the product of the merchant-bar mill was usually delivered from the finishing stand into a trough. The ends of each bar were manually inserted into the heads of an apparatus for stretching, to obtain the necessary straightness, and the bar

was then dropped upon a sloping-skid bed to cool slowly in contact with other bars. This was common practice, at least with square and rectangular sections. Rounds, straightened in the same manner, were usually permitted to roll down the sloping-skid bed onto a secondary bed of stationary horizontal skids. The first round delivered to the secondary-skid bed would be laid straight, and parallel to the mill delivery trough already referred to. Subsequent bars would roll by gravity until stopped by making contact with bars already deposited on the secondary-skid bed. The cooling of the rounds, therefore, was also effected at a slow rate by maintaining contact between the bars during the cooling process. Having collected a group of bars in this manner, a part thereof were moved on for shearing and the remainder were moved forward to make room for additional bars from the mill. In other cases, the mill was operated to collect a group, or pack of bars, and then allowed to remain idle until the bars were sufficiently cooled to permit them to be moved, making space for another pack. The use of a double bed permitted the delivery of bars to one bed while bars were cooling on the other, and vice versa.

The introduction of the mechanical cooling bed permitted the rolling of longer lengths than could be handled on the old hand beds. This meant the use of heavier billets, fewer operations in the mill, and less scrap loss when shearing the bars to fixed lengths. The mechanical cooling bed could be built to cool any desired tonnage per hour, but it was designed for cooling the individual bars in spaced formation. During the cooling process the bars were supported in grooves formed by a series of spaced skids. The skids were made of rough castings and the straightness of the grooves governed the straightness of the bars cooled therein. Not only was the straightness of the product impaired by inequalities of the skid castings, but the wear of the moving parts of the bed mechanism, or the replacement of worn parts, would cause further misalignment in the grooves and distortion of the bars subject to cooling.

For certain classes of products, such as screw stock and spring flats, for instance, the rapid cooling of the bars, when carried in spaced formation on the mechanical bed, was detrimental. The annealing effect obtained with the old hand bed, where the product was cooled in pack formation, would soften such classes of product and make them better suited to subsequent manufacturing operations.

Attempts have been made to pack-anneal spring flats on mechanical beds by collecting them in small groups, and then handle these groups in the ordinary manner in the grooves of the bed. This method of pack annealing, however, has the disadvantage that the outside bars in each group cool at a faster rate than the interior bars, causing the mill product to be non-uniform in hardness. As far as rounds are concerned, the ordinary mechanical cooling bed has no facilities for their pack-annealing, i.e., for the slow cooling of rounds in pack formation. It should also be noted that the rapid cooling of bars in spaced formation, with free access of air to each individual bar, is apt to cause more scaling of the bars than when they are cooled in contact with one another.

To overcome the difficulty of non-uniform hardness encountered with small individual packs, beds have been devised for continuous mechanical pack-annealing, but most of these designs are predicated on the use of a separate bed for the pack-annealing of spring flats only, and are not adapted for the packing of rounds or the handling of mill products which do not require pack-annealing.

With increasingly exacting demands on mill products, particularly in the case of relatively high-carbon, special, and alloy steels, it appears that the mill operator must be in a position to control the grain structure of his product. To this end he requires facilities to govern the rate of cooling, not only in the case of flats, but also in the case of squares, rounds, and other sections. A given

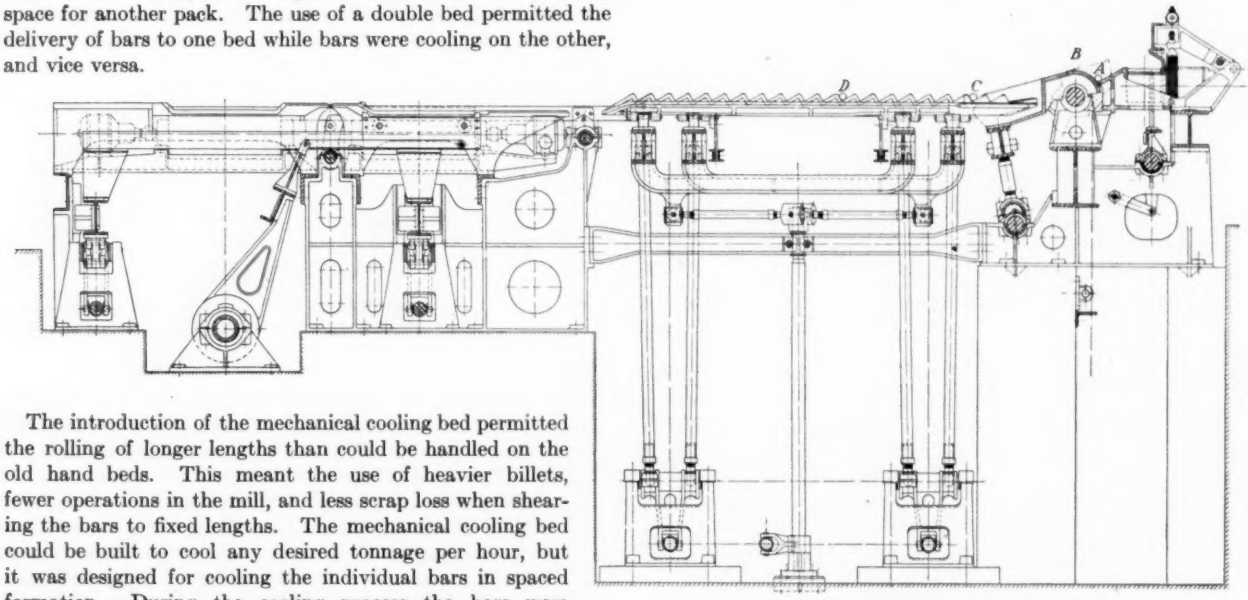


Fig. 10 Recent Design of Cooling Bed

cooling bed may therefore be called upon to handle any of these sections in spaced formation, or in pack formation, to suit the requirements of individual orders. Few steel plants can afford to build single-purpose mills or to limit the scope of a mill to the range of products for which the ordinary cooling bed is adapted. Certain grades of steel, when cooled in pack formation, will camber decidedly, due to contact with hot bars on one side and open-to-air cooling on the other. Such bars, when permissible to do so, should be cooled individually while supported in grooves in two directions. Other bars of irregular cross-section are likely to acquire a decided camber during the cooling process, or may be rolled with a camber to cause them to be straight when cooled. Such bars should be handled on a flat bed. In order not to limit the scope of the mill, therefore, a single bed should be adapted for the cooling of bars in both spaced and pack formation, and should preferably have its surface adjustable to meet the requirements already outlined. Provision must also be made to avoid scratching and marking of bars during their handling to and by the cooling bed.

A cooling bed recently installed and built to meet these requirements is shown in cross-section in Fig. 10. The roller table conveying the rolled bars from the mill to proper position alongside the cooling bed has cylindrical rollers flanged at one end and placed at a slight skew, in order to cause the bar to align itself

along the flanges of the rollers. The skewing of the rollers serves to keep the bar from wandering sidewise on the face of the rollers, which would cause it to lie crooked prior to the kick-off operation. In such event the kick-off arms would strike certain portions of the bar ahead of other portions and would be likely to cause kinks and twists in the rolled bars during the kick-off operation. As already explained, this is avoided by skewing the rollers, and their flanges serve to keep the bar out of contact with the side guard while the former is being conveyed to position alongside the cooling bed. The straight rollers are an improvement over the tapered ones frequently used in similar tables, in that the different surface speeds at various points of the taper tend to twist the hot bars, such as squares, during their travel. Any twisting of the bar around its own axis is thus avoided by the use of cylindrical rollers.

The stopping of the bar on the run-in table alongside the cooling bed is accomplished by lifting the apron plates above the table rollers. The apron plates, in their lifted position, are arranged to support the bar for its entire length, except for the necessary gaps at the table rollers. The top surfaces of the aprons are smooth-machined and apply a gentle frictional resistance for practically the entire length of the bar, to avoid scratching or marking of product.

From the cooling-bed run-in table, the individual bars are swept sidewise onto the cooling bed by means of a series of spaced kick-off arms. These kick-off arms and the lifting apron plates are operated from a common drive. They are so interconnected that the apron plates will lift and allow the bar in transit to come to rest before the kick-off arms engage the bar and sweep it onto the cooling bed. In this manner any nicking or marking of the hot bars by the kick-off arms is avoided. The motion imparted to the kick-off arms is such that their free ends sweep the bar off the run-in table and then return beneath the surface of the table to their point of starting. By virtue of this arrangement of the kick-off mechanism, a bar can be delivered from the mill onto the aprons of the cooling-bed run-in table and behind the kick-off arms while these arms are engaged in sweeping the preceding bar onto the cooling bed.

The kick-off arms deliver successive bars onto an apron of sufficient slope to cause the individual bars to slide by gravity to a stop, against which each bar is aligned by impact, to insure straightness prior to delivery to the cooling bed proper.

The cooling bed has two sections. A preliminary section serves for the pack-annealing of squares, flats, and other rectangular sections, but is also useful in the handling of rounds, as will later be set forth. A secondary section provides for the cooling of bars in spaced formation and also for the packing of rounds.

Referring to the primary-bed section, the packing of flats, which should be cooled on edge in order to insure straightness, is accomplished by the use of counterweighted dogs, or abutments, against which the first bar delivered is deposited for sidewise support against overturning. Succeeding bars from the mill are deposited in identically the same position as the first bar, causing the dogs and the bars previously deposited to yield or move a distance equal to the thickness of each bar delivered. In this manner a pack of bars is formed with the edges of the individual bar resting on the surface of the primary bed. When the pack has attained sufficient size to be self-supporting, the dogs automatically withdraw below the bed surface. Further additions of bars will cause the pack to grow until it has attained a size limited by the width of the primary-bed section. The bars first deposited will then slide off the end of the primary bed onto the secondary bed as additional bars are added to the pack. The pack-annealing operation thus becomes continuous, in that bars will then be discharged at the delivery end of the primary bed at the rate at which bars are added to the opposite end of the pack. The width of the primary section is sufficient to complete the annealing operation and allow the bars to take a permanent set before they are delivered to the secondary-bed section.

The secondary bed has two sets of carry-over bars, one set balanced against the other. By such balancing of the moving parts they can be made relatively light, in that the only work performed in operating this bed section is to overcome the friction of moving parts and to lift the material deposited upon the bed. Each carry-over bar in both sets has two working faces, one straight and the other notched. A motor-driven turnover will quickly move one or the other of these working faces into operating position, thus producing in one case a notched bed surface suited for advancing bars from notch to notch, and in the other case giving the bed a flat top surface on which bars may be advanced in pack formation or spaced, as the case may be.

To illustrate the adaptability of this bed, its operation with reference to handling of various grades and shapes of product should be noted. Let us first consider the cooling of rounds in spaced formation. Delivered by the kick-off arms onto the cooling-bed apron, the bar will receive a preliminary straightening when stopped by impact in position *A*, Fig. 10. It will then be lifted to position *B*, to roll by gravity down the slope of the primary bed section to position *C*, in a groove of the carry-over bars of the secondary-bed section. The free rolling of the bar down the primary-bed section is an excellent method for straightening rounds. Successive rounds are carried forward, groove by groove, from position *C* on the secondary-bed section, from which section they are finally discharged, one by one, onto shuffle bars, for group assembling in the customary manner prior to shearing.

The working surfaces of the entire bed are smooth-machined, including the notched and straight working faces of the carry-over bars. Marking of the product is avoided by these means, and a straight round, delivered to groove *C* of the secondary bed, can be maintained straight throughout the cooling process by avoiding the irregularities of unmachined castings. To maintain the alignment of the machined grooves of the secondary-bed section, this bed is made up of short individual units, each readily adjustable to move the carry-over bars of any unit longitudinally in either direction, or up and down, as may be required to maintain exact groove alignment. This adjustment can readily be made by the aid of a wrench without interruption to the operation of the bed.

When desiring to cool rounds slowly in pack formation, the first round delivered to position *C*, Fig. 10, is immediately carried forward, groove by groove, to position *D*. The adjusting motor of the carry-over bar is then operated, changing the secondary bed to a flat surface and leaving the first round as shown in position *D*, Fig. 11. The next round delivered from the mill, when caused to roll down the slope of the primary bed, continues its rolling until it makes contact with the first bar deposited on the secondary bed. When a small group of bars has been assembled in this manner on the secondary bed, it is moved forward sufficiently to make room for additional bars. The process is repeated until the secondary bed is covered with rounds, all straightened by rolling, and lying in contact with one another. With the secondary bed filled in this manner, the cooled rounds will be delivered to the shuffle bars during the continued operation of the process, which then is continuous, maintaining a pack as shown in Fig. 12.

The bar-lifting arms shown in dotted lines in Fig. 10, and serving to raise the rounds from position *A* to position *B*, are attached to a shaft, located as shown, beneath the surface of the primary bed. The shaft is motor driven, and when turned approximately one-half revolution the bar-lifting arms become hidden below the bed surface and another set of arms attached to the same shaft, which will be referred to as "packing arms," are moved to

operating position. The packing arms are illustrated in Fig. 13, and are used to turn flats edgewise and deposit them on the primary-bed section. The slope of the primary bed is made adjustable for reasons readily understood from the operation of the bed, yet to be described.

With the packing arms in operating position, and the slope of the primary bed adjusted to the position shown in Fig. 13, the packing dogs, or abutments, previously hidden below the bed referred to. Sections now delivered to the primary-bed section will slide on the steeper slope to be deposited in a groove as shown at *G*, Fig. 17, for subsequent forward movement, groove by groove as additional bars are delivered from the mill.

It will be apparent that angles and irregular sections can be delivered by the mechanisms described to the secondary-bed section for cooling, while supported in grooves, or while placed in close or spaced formation on a flat bed surface.

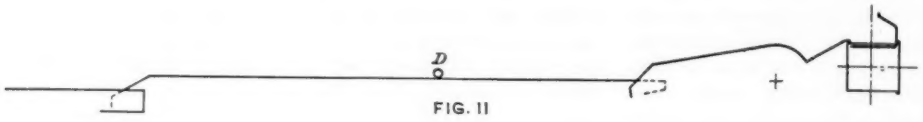


FIG. 11

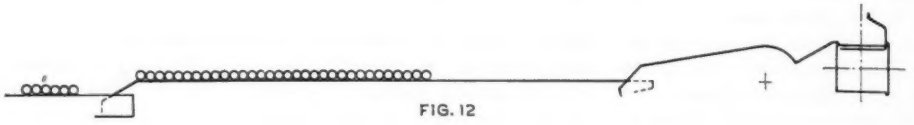


FIG. 12

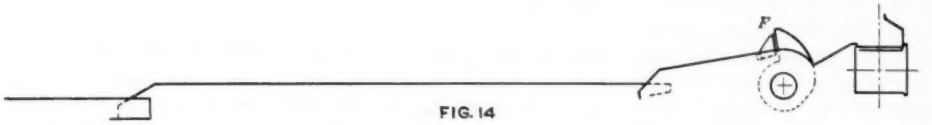


FIG. 14

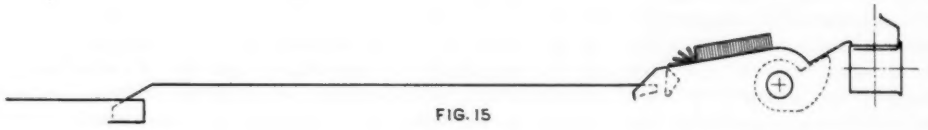


FIG. 15

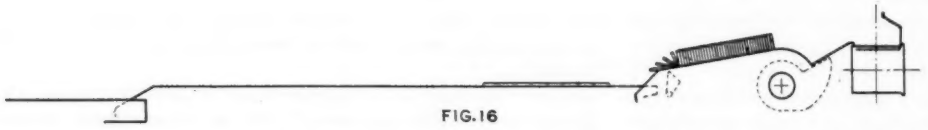


FIG. 16

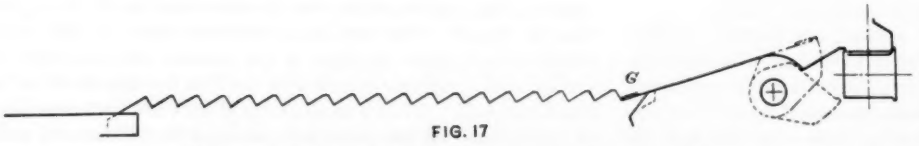


FIG. 17

FIGS. 11, 12, AND 14–17 OPERATION OF COOLING BED

surface, are moved by motor to position also shown in Fig. 13. The bed is then in order for continuous pack-annealing of flats. The first flat delivered to the cooling-bed apron slides to a stop for straightening by impact at position *E*, Fig. 13. It is then moved to position *F*, Fig. 14, and deposited against the dogs by means of the packing arms. Subsequent stages of this packing operation are illustrated in Figs. 15 and 16. Squares may be pack-annealed in a similar manner, except that the use of packing dogs is not then required.

Flats or squares not requiring pack-annealing may be cooled in spaced formation by changing the slope of the primary bed as shown in Fig. 17 and by changing back to the bar-lifting arms first

COLD SHEARING

Beyond the cooling bed is the shearing equipment for cutting the cooled mill products to lengths for shipment. It is frequently the case that one cold-bar shear is unable to keep up with the production of a mill, and merchant mills for this reason are often provided with two shears for dividing the rolled bars into the desired lengths. The conventional way to meet this situation has been to provide a double cooling bed, such as shown in Fig. 1, to divide the mill products between two beds, which in turn deliver to individual shear tables and shears. This can hardly be termed a practical and economical solution to the problem, for several reasons. A cooling bed is an expensive piece of equipment, but can be built for practically unlimited cooling capacity without very materially increasing its cost. The run-in table to the cooling bed, the bar-assembling means following the cooling bed, such as shuffle bars, and the cold run-out table, are all the same, irrespective of bed capacity, for any given length of bed. To increase the capacity of a bed, it is merely necessary to lengthen the carry-over bars of the bed proper and possibly make the operating mechanism for this bed portion somewhat heavier. The double bed must, for this reason, be considered incidental to double shearing and not as a requisite to augment the cooling capacity. Cooling beds are very reliable in service, and it would seem far-fetched to consider one side of a double cooling bed as a necessary spare in case of trouble on the opposite side.

In the layout shown in Fig. 2 there are two shears, but only one cooling bed. Still the two shears have theoretically the same shearing capacity in either case. This is accomplished by spacing the shears from the cooling bed so that the first shear is removed from the bed a distance not less than one-half the bed length. The cooling bed is provided with two cold run-out tables, one for each shear, and the first shear is built so as to permit the cold run-out table for the second shear to run through its frame. Furthermore, a transfer mechanism is provided, as shown in cross-

section in Fig. 10, which serves to lift a set of bars from the run-out table next to the cooling bed, transfer it to above the rollers of the second run-out table, and lower it onto these rollers, all in one operation. This transfer mechanism is made of the same length as the cooling bed, to handle batches of full-length bars.

Referring to Figs. 10 and 2, a group of rolled bars are assembled on the shuffle bars and transferred in the usual manner onto the first cold run-out table. This group of bars is immediately transferred, by the mechanism already referred to, onto the second cold run-out table for conveyance to the second shear. In the meantime, another group of bars is moved onto the first run-out table and conveyed by its rollers to the first shear. Assuming the two shears to be operating at approximately the same speed, it will be apparent that this process can be continued indefinitely, by alternately furnishing groups of bars to the two shears. The distance between the cooling bed and the first shear provides for the necessary timing.

The additional table and transfer equipment necessary with the layout shown in Fig. 2 are decidedly cheaper than the extra cost of a double as compared with a single bed. The shearing capacity in either case is the same. There are other advantages, however, which make the shearing and cooling bed layout shown in Fig. 2 very desirable.

A double cooling bed takes up a great deal more space, and the layout of the mill usually makes it necessary to locate the cooling bed more or less centrally in the mill building, thus dividing the remaining floor space into narrow portions, of little value for storage and handling purposes. With the double shearing arrangement from a single cooling bed, the bed, table, and shear equipment can usually be placed over to one side of the mill building. Not only is considerable floor space conserved in this manner, but the remaining portion is uninterrupted, and for that reason decidedly more useful. Compare Figs. 1 and 2, which both show actual modern installations.

Let us next consider a mill with two cooling beds serving different outlets and handling different classes of products. One bed of such a mill would be comparatively long and narrow for small-sized bars. The outer outlet, delivering heavier products, would require a shorter bed, but wider to obtain sufficient cooling capacity. Taking advantage of this situation, both beds can be made to deliver to the same cold run-out table, as is the case in the mill layout shown in Fig. 3. An additional cold run-out table and transfers, just as specified in the case of the layout shown in Fig. 2, make it possible, and entirely practical, to use two shears for shearing the products from either of the two beds. Since the two outlets are never used at the same time, this arrangement gives the advantage of two double beds with two shears for each, and with a decided saving in cost, mill floor space, and ease of operation.

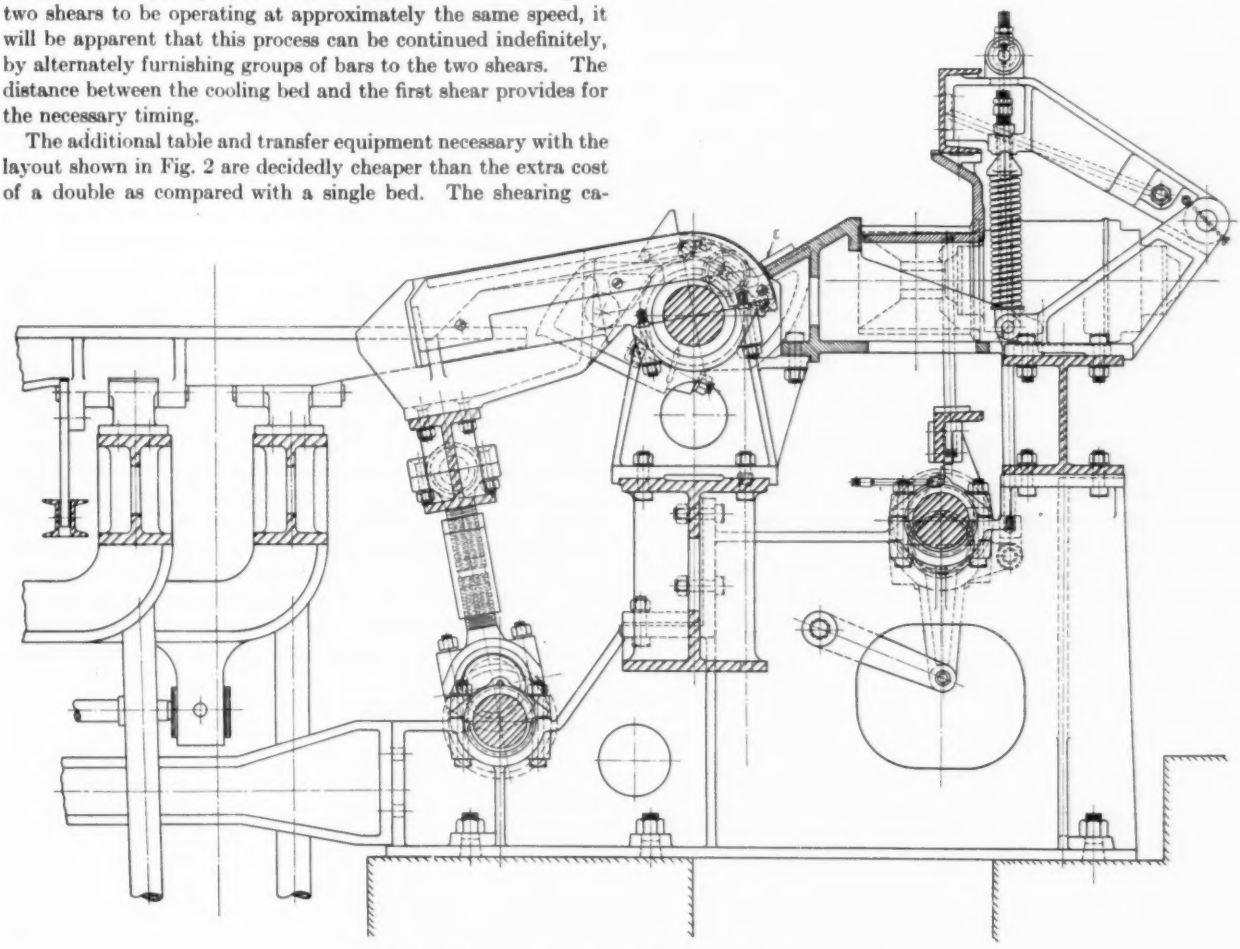


FIG. 13 SHOWING DETAILS OF COOLING-BED MECHANISM

There are no doubt many other recent improvements worthy of note, but some of these must as yet be considered in the development stage. This is the case, for example, with satisfactory equipment for the coiling of merchant bar. Developments in this line are to be expected.

Comparing the merchant mill of today with that built five or more years ago, there is undoubtedly a strong tendency toward labor-saving and high-grade equipment. Quantity of product, once the seemingly all-important factor, is now considered, only as far as consistent with quality, the paramount requirement.

Discussion

S. M. Weckstein.[2] The author has pointed out that attempts have been made with more or less success to substitute roller bearings on the rolls of existing mills. It has been our experience that the rolling requirements of the mills will govern the possibility of making the changeover.

At the plant of the Timken Steel and Tube Company two stands of 22-in. three-high bar mills designed for plain bearings were changed to tapered roller bearings. The necks for the plain bearing rolls were 14 in. in diameter and 14 in. long. For the roller bearings the necks were cut down to 12 in. diameter. The length of neck was retained, as were the 24 in. maximum and 21 in. minimum roll diameters. The housings were machined out for the bottom roll. At the present time these stands can easily be changed over from plain to roller bearings by the use of proper liners. This was demonstrated recently when comparative tests were made on the same rolling practice on plain and roller bearings. These tests showed power savings of $40^1/_2$ to $48^1/_2$ per cent in favor of the roller bearings. The bearings on the pinion stand also were changed over to roller bearings, and the total savings obtained have long since paid for the cost of the changeover.

The same satisfactory results have been obtained on a number of changeovers of 10-in., 12-in., and 16-in. two- and three-high mills.

We have, however, run into a number of cases where it has been impossible to make the changeover due to the rolling loads encountered. In some cases it was impossible, due to the outside diameter of the bearing, to retain the present minimum roll diameter. In other cases it was necessary to change to alloy rolls in order to retain proper neck strength. In still a few other cases the housings were not strong enough to permit of opening up the windows the required amount.

It has been pointed out that in new mills proper provision can be made for the incorporation of roller bearings by an increase in roll diameters. This naturally increases the roll cost and to some extent the mill machinery cost. To offset this initial increase in cost it should be borne in mind that it is safe to figure an average saving of about 30 to 35 per cent in power, and consequently a reduced first cost of the electrical equipment required. In addition the reduced cost of lubrication and maintenance and the reduction in scrap due to positive control of the accuracy of the product must be credited to the roller bearings.

Another important factor which as yet cannot be fully accounted for is the life of the bearings and the resultant cost of bearings per ton of product rolled. At the Timken plant there are two stands of three-high 28-in. mills, two stands of three-high 22-in. mills, three stands of 18-in. two-high mills, one stand of 16-in. three-high, and six stands of 16-in. two-high mills, all equipped with tapered roller bearings. On all of these mills an accurate record is being kept of the number of hours of operation and the number of tons rolled on each individual bearing. For example, in the 28-in. mills where bearings of 16-in. bore 24 in. outside diameter by $12^1/_2$ in. long are used, the maximum tonnage rolled on any one bearing to date is approximately 200,000 tons, equivalent to approximately 5000 hours in service. At the last inspection these bearings were in perfect condition. Similar tonnages have been rolled on the other mills with the bearings in perfect condition.

The question of spares is also important. In 1928, when the Timken company mills were installed, the experience with roller-bearing-equipped merchant mills was limited. Consequently, as a safety measure it was decided to carry on hand a large number of spares. The set-up was to have a complete set of bearings in the mills, a complete set of bearings assembled on the spare rolls ready to go in the mill, and a third set mounted in chucks and ready for assembly on any desired rolls. A study of the record charts giving the tonnage rolled on the bearings shows that only a few of the third set of bearings have ever been used. This is due to the trouble-free operation of the bearings and to the ease of changing bearings on rolls. It is therefore safe to figure on one set of bearings in the mill and one set of spares.

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Carburizing With Butane Gas

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The hydrocarbon gases of the paraffin series which are being used for commercial carburizing include methane (natural gas), propane, and butane. These hydrocarbons used without the admixture of air or other gases apparently give up carbon to the steel during certain cracking processes which occur at certain rather sharply defined temperatures and other conditions. Operating conditions which give satisfactory results with one hydrocarbon may not produce satisfactory results with other hydrocarbons. These conditions are not difficult to meet when fully understood.

Well-controlled butane offers: (1) smaller temperature range than commercial solid carburizers; (2) carburizing material at low cost; (3) maximum carburizing rate at temperatures metallurgically and economically advantageous; (4) chemically constant carburizing material; (5) carburizing material requiring no complications, such as admixture of air or other gases; (6) no labor or space within the plant is required for handling or storage of solid carburizer; (7) carburizing process so free from dust and dirt that with proper furnace equipment it may be placed in the regular factory line of manufacture; (8) absolute control of depth and character of case.

The paper gives some details of the author's experiments, certain general observations regarding the physical chemistry and a table of physical constants of the paraffin hydrocarbons, a description of a flowmeter for measuring the flow of carburizing gas to the furnace.



IN GENERAL it may be said that all commercial steel harder than the softest grades is dependent upon its carbon content for the additional hardness, although of course it may have its hardness increased, to some extent, by cold work; also the hardness as well as the strength is effected to some extent by various alloying elements, notably, manganese, chromium, tungsten, nickel, molybdenum, vanadium, etc.

Generally speaking, in spite of the marvelous progress that has been made in steel composition during the last few decades, we still do not know how to make steel commercially that is extremely hard and resistant to abrasion and at the same time tough and ductile. That is to say, steel that is extremely hard is also brittle, and therefore usually not well suited to withstand shocks or flexure.

Steel that is low in carbon is usually comparatively tough and ductile, and is therefore suitable for use where it is subject to shocks. Low-carbon steel may be given a skin of higher carbon which, with suitable heat treatment, will afford a hard and abrasion-resistant surface while the inner part remains tough and shock resistant. This combination furnishes the nearest approximation which we have to a steel which is hard and abrasion resistant on the surface, and at the same time sufficiently tough internally to withstand shock and some flexure without failure, and it therefore finds extensive use in steel machine parts.

The process of adding carbon to the surface portions only of a piece of steel, is known as "carburizing" or "cementizing," or in the vernacular, "case-hardening." It is also well known that the thickness of the hard skin or "case" may be varied at will by the treatment given. In fact, the piece of steel may, if desired, be carburized all the way through, thus changing it from a piece of low-carbon steel to one of high-carbon steel. This method is a very old one. The famous swords of Damascus are believed by some to have owed their superior cutting qualities and sturdy flexibility to steel which had been carburized, although the word "carburized" is of very much more recent origin.

The process was formerly carried out by heating the parts packed in carboniferous material in some kind of heat-resistant boxes. Previous to perhaps twenty years ago the better grades of work were obtained by using charred leather, granulated bone, and similar substances of animal origin. These materials, especially granulated bone, are still used to some extent, but most carburizing is now done by packing in granulated materials consisting mainly of various kinds of charcoal and coke with which are mixed small percentages of chemicals such as barium, calcium, and sodium carbonates. These chemicals are known as "energizers" or "activators," because their presence accelerates the absorption by the steel of carbon from the charcoal-coke source of supply, probably through the intermediary of carbonaceous gases.

During recent years various gases have grown in favor as the source of carbon and gaseous carburizing materials, and methods of application have been the subject of many patents. There are two more or less distinct methods of using gaseous carburizers:

1 Indirect, or converting the carbon into carbon monoxide, which carries the carbon in available form. This method is available wherever the carbon can conveniently be converted into carbon monoxide and is probably more generally available than the following method.

2 Direct, or breaking down the carboniferous compound in the presence of the steel to be carburized, into other compounds containing a less proportion of carbon, thereby releasing nascent carbon directly to the steel without the intermediary carbon monoxide.

The present discussion will be confined to the latter method.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Mechanics of Carburization

In dealing with the subject as a whole, it will be advantageous at this point to consider the mechanics of carburization—that is, the method or manner in which the carbon is transferred from the carburizing material to the steel. The most commonly accepted theory regarding solid carburizers seems to be that carbon monoxide (CO) is the intermediary between the carburizer and the steel. Experiments indicate that steel will take up little or no carbon directly from solid carbon in a vacuum—that is, in the absence of gas. Experience seems to prove conclusively that no carbon monoxide, cyanogen (CN), or any other compound of oxygen or nitrogen is essential to satisfactory carburizing with some carburizing materials including butane gas.

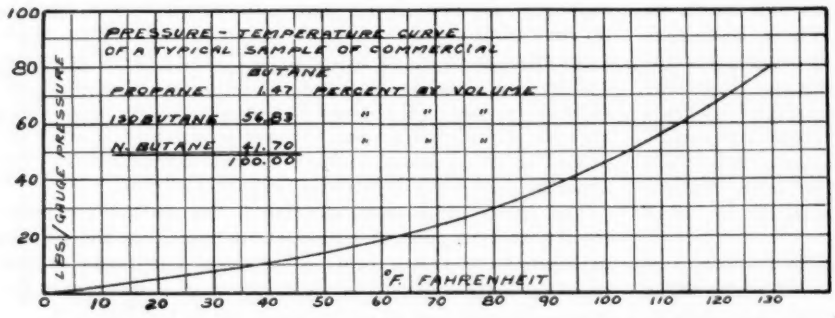


Fig. 1 Pressure-Temperature Curve of a Typical Sample of Commercial Butane

As a theory of the mechanics of carburization of steel, the author submits that the essentials consist of steel in the presence of some substance containing carbon in chemical combination, which combination shall change while in intimate contact with the steel to be carburized and at a temperature or temperatures metallurgically suitable to the steel, into other chemical combination or combinations containing a lesser proportion of carbon and thereby releasing nascent carbon to be absorbed by the steel; provided, that no other element is present which has a greater affinity for the nascent carbon than the steel has at the particular temperature at which the change in combination takes place. Further, that any substance, solid, liquid, or gaseous, that meets the foregoing conditions may be used as a carburizer with greater or less efficiency, but that obviously a vapor or gaseous compound will be the most efficient because of the greater ease with which that portion of vapor or gas in intimate contact with the steel and which has given up its quota of carbon to the steel, and has thereby lost its potency, may be replaced with potent material.

Properties of Butane and Other Hydrocarbons

Butane has long been known as a small-percentage component of natural gas and of crude petroleum, but has not been developed as a generally available commercial fuel until within the last very few years. It is obtained commercially from at least three more or less distinct sources:

a It is a by-product of compression plants at the origin of long-distance natural-gas transmission lines

b It constitutes one of the numerous "light ends" at crude-petroleum distillation plants

c It is a by-product of the petroleum "cracking" still due to the overhydrogenation of small portions of the heavier hydrocarbons.

Butane is transported as a liquid under pressure in suitably constructed tank cars or in smaller pressure containers. The pressure varies with the temperature. Fig. 1 shows this relation for a typical sample of commercial butane. Since butane gas is rapidly assuming a place of commercial importance and since information regarding it is not generally available in the mechanical handbooks, data on butane and closely related hydrocarbons are given in the accompanying Table 1.

Propane is the next lighter hydrocarbon of the paraffin series and is obtained from the same sources as butane. It has for some years been marketed under various trade names, as a gas for kitchen ranges and other heating purposes, where commercial gas mains are not available. Propane may be readily obtained commercially in 100-lb drums as well as in tank cars and may be used as a carburizer quite similar to butane, although to obtain the best results the temperature and rate of gas feed will be somewhat higher than when butane is employed.

These hydrocarbons begin to change or "crack" when the temperature is increased above certain limits, which vary with the hydrocarbon under consideration. The temperatures at which these changes begin in the presence of steel only are approximately as follows:

Methane	1200 F	Propane	860 F
Ethane	840 F	Butane	815 F

While the temperatures given are those at which changes begin to occur, temperature differences continue on up through the working ranges; that is, if methane cracks into certain other hydrocarbons at, say, 1700 F, butane may be expected to produce similar hydrocarbons at some lower temperature, perhaps 1500 to 1600 F. Further, it happens that most of the hydrocarbons obtained by cracking at these temperatures will crack again into other compounds at still higher temperatures, but it also appears that most of the hydrocarbons which are formed within the temperature range metallurgically suitable to the steel, namely, 1500 to 1800 F, do not in general change again under carburizing-furnace conditions until a temperature of 2000 F or more has been reached, which of course is much too high for the steel from a metallurgical point of view.

Further, it appears that in carburizing, as in many chemical operations, the desired action is much more effective when certain of the hydrocarbon's constituents are in a nascent condition. This, considered with previous statements and some which are to follow, suggests the reason why each hydrocarbon has its own particular temperature range and rate of gas feed at which it is most effective, and outside of which the results may be far from satisfactory.

It may be noted here also that these changes may produce hydrocarbons which are much more complicated in their atomic structure than the original gas. For instance, imagine the author's surprise during his carburizing experiments to find that while he had butane entering one end of the retort, he had coal-tar vapors mixed with naphthalene (from which moth balls are made) and probably other less easily detected substances issuing in substantial quantities from the other end of the retort in which carburizing was supposed to be taking place and not the manufacture of drugs and road materials. Incidentally it may also be noted that, under the conditions where tar was given off, it formed a coating on the work being carburized which, when the temperature was further increased, seemed to dry out, leaving a coke-like coating which effectively prevented carbon penetration at any temperature. This coke-like substance adhered so tenaciously to the surface that nothing less drastic than a severe sand blast would remove it.

TABLE 1 PROPERTIES OF PARAFFIN HYDROCARBONS

	Methane CH_4	Ethane C_2H_6	Propane C_3H_8	Butane C_4H_{10}	Isobutane C_4H_{10}	Pentane C_5H_{12}	Isopentane C_5H_{12}	Hexane C_6H_{14}
Melting point, deg F	—299.2	—277.6	—309.8	—211.0	—229.0	—204.7	—225.5	—137.7
Sp gr of liquid, 60/60 F	.....	0.3779	0.5090	0.5845	0.5647	0.6312	.....	0.6641
Deg A.P.I. of liquid	.....	242.9	146.5	110.6	119.1	92.7	.....	81.6
Lb per gal of liquid at 60 F	.....	3.141	4.234	4.864	4.699	5.253	.....	5.527
Gal per lb of liquid at 60 F	.....	0.3184	0.2362	0.2056	0.2128	0.1904	.....	0.1809
Vapor pressure of liquid, lb per sq in. gage at 60 F	.....	483	92.4	11.6	24.0	16.0*	11.6*	26.1*
Vapor pressure of liquid, lb per sq in. gage at 80 F	.....	617	128.1	22.9	39.2	8.6*	1.1*	23.6*
Vapor pressure of liquid, lb per sq in. gage at 100 F	.....	.....	172.0	37.5	59.0	0.9	5.8	20.0*
Vapor pressure of liquid, lb per sq in. gage at 120 F	.....	.....	225.0	56.1	83.3	7.8	13.9	14.8*
Mean coefficient of expansion:								
0– 50 F	.....	0.0028	0.00143	0.00096	0.00113	0.00081	.....	0.00070
50–100 F	.....	.....	0.00180	0.00113	0.00128	0.00089	.....	0.00078
Boiling point, deg F	—258.5	—127.5	—44.0	+30.9	10.0	97.0	82.3	156.1
Heat of vaporization at boiling point:								
Btu per lb	222.7	210.3	183.1	165.6	158.8	153.1	.....	143.3
Btu per gal of liquid at 60 F	.....	660.6	775.2	805.4	746.2	804.2	.....	792.0
Critical temperature, deg F	—115.8	+90.0	207.5	272.7	307.8	387.0	370.0	454.6
Critical pressure, lb per sq in.	673	708	661	533	524	485	483	435
Specific gravity of gas (air = 1)	0.554	1.049	1.562	2.085	2.067	2.490	2.490	2.974
Density of gas, lb per cu ft	0.0424	0.0803	0.1196	0.1596	0.1583	0.1907	0.1907	0.2277
Cu ft of gas per lb of liquid at 60 F	.....	12.49	8.379	6.277	6.330	5.254	.....	4.399
Cu ft of gas per gal of liquid at 60 F	.....	39.21	35.48	32.5	29.74	27.60	.....	24.31
Specific heat of gas at constant pressure:								
Btu per lb	0.593	0.505	0.473	0.456	0.462	0.448	0.448	0.442
Btu per cu ft	0.0251	0.0406	0.0566	0.0728	0.0731	0.0854	.....	0.1006
Limits of inflammability, per cent of gas:								
Lower limit	5.3	3.2	2.4	1.9	.....	1.4	.....	.....
Upper limit	14	12.5	9.5	8.5	.....	8.0	.....	.....
Maximum rate of flame propagation, in 1-in. tube, ft per sec	2.17	2.80	2.69	2.71	.....	2.72	.....	.....
Composition at maximum rate of flame propagation, per cent of gas	9.96	6.53	4.71	3.66	.....	2.92	.....	.....
Cu ft of air to burn 1 cu ft of gas	9.57	16.72	23.92	31.10	31.10	38.28	38.28	45.41
Btu in gas per lb of liquid at 60 F	.....	21882	20973	21308	20579	20948	.....	20583
Btu in gas per gal of liquid at 60 F	.....	68695	88806	106500	96685	110041	.....	113746
Btu per lb of gas	23628	21807	20925	21320	20538	20911	21043	20549
Btu per cu ft of gas	1003	1752	2503	3274	3251	3987	4012	4679

All weights of liquid are in air.
All gas volumes corrected to 60 F and 30 in. of mercury, dry.
Specific gravity of gaseous pentanes and hexane, specific heats of gases, and volume of air required for combustion computed.
Limits of inflammability from U. S. Bureau of Mines Bulletin No. 279.
All other data based on figures from International Critical Tables and from paper by Dana, Jenkins, Burdick, and Timm, *Refrig. Eng.*, vol. 12 (1936), p. 387.
* Inches of mercury vacuum.

The various hydrocarbon gases in Table 1 may in general be transformed into other similar classes of hydrocarbons by suitable manipulation of temperature, pressure, rate at which the gas is fed through the retort, length of time that the gas is exposed to any specific temperature, the material of which the retort is composed (steel, copper, quartz, glass, etc.), and probably some other conditions.

In view of all the possibilities, it is not strange that experimenters with butane for carburizing have experienced much difficulty when trying to use it according to methods previously developed for methane (natural gas) or solid carburizers.

Experiments With Butane in Carburizing

In beginning his butane carburizing experiments the author acted on the advice of a well-known builder of carburizing furnaces, namely, that the proper carburizing mixture to use would be 18 per cent butane and 82 per cent air. This recommendation was evidently made in the belief that the presence of a gas containing oxygen is essential to successful carburizing. Considerable care was taken to obtain exactly the specified mixture of air with the 48 cu ft of butane gas per hour which the rotary-type furnace to be used in the experiments was supposed to require. Again imagine the author's surprise after taking extreme care to get the proper quantity of the correct mixture, to find after only half an hour at heat that the end of the heavy nichrome tube, through which the mixture was introduced into the retort, as well as the control thermocouple, were completely melted away! And worse still, the batch of material to be carburized, consisting of about 8000 small steel bushings, was welded together and rolling around in chunks the size of a man's head! If one's foresight were only as good as one's hindsight, much trouble would be saved in experimental work. An 18 per cent butane mixture is still twice too rich to be a combustible one. Obviously, however, when entering a hot furnace the oxygen promptly combined by partial combustion with the carbon and hydrogen present to form, among other things, the carbon monoxide (CO) which the author's adviser apparently desired. Both the author and his adviser, however, failed to recognize that even the partial combustion of 48 cu ft per hr of butane gas containing approximately 3200 Btu per cu ft would liberate enough heat to ruin the inside of an already hot furnace, including its contents.

Naturally the author promptly decided that, if butane was to be used, some other method than the one advised would be necessary. After a little further study he decided to see what would happen if he used butane without any admixture of air. By blind luck (as he now knows) the first batch of work tried with the raw gas turned out an excellent eutectoid case approximately 0.040 in. thick within four hours at heat.

Several experimental heats were run at that time, all of which were very successful, and at that stage carburizing with butane, as it seemed to the author, was extremely simple. These experiments were conducted using both butane and propane from 100-lb drums.

Some months later an installation for using butane gas as a principal heating fuel was completed, and that gas was available for production carburizing. What a surprise was in store when the use of butane was started in regular production! Instead of being 99 per cent perfect as anticipated, the results were very unsatisfactory because the time required to obtain a suitable depth of case was much too long, and even when the depth was sufficient, the character of the case was not satisfactory.

Several impromptu modifications were tried without satisfactory results, and the entire carburizing department then went back to the use of solid carburizer until the discrepancies between experimental and commercial practice could be ironed out. It was at this point that the author found it necessary to add substantially to his knowledge of hydrocarbons and their action under heat. A study of some late literature apply-

ing primarily to the cracking of the heavier hydrocarbons to produce gasoline brought out many facts, some of which have been previously noted, especially the fact that different hydrocarbons may crack into similar products at suitably different temperatures. It was known that methane (natural gas) was being successfully used at the same temperatures that are commonly used for solid carburizing material, namely, 1700 to 1750 F, and it seemed that butane might be successful at some lower temperature.

The fact also developed at this time that the original experiments had been conducted at temperatures in the vicinity of 1600 F for metallurgical reasons and not to favor the carburizer. In production the gas was put into service at about 1700 to 1750 F. Dropping the temperature to 1600 F immediately gave better results, but before again attempting the use of butane in regular production, it was decided to explore the entire possible range, because 1700 F seemed too high for the butane and about 1500 F is well established as a minimum limit at which steel will take up carbon at a commercially feasible rate. It also seemed obvious at this time that the rate of gas flow would probably have an important influence upon results.

Results Obtained in Experiments

Briefly, the results are as follows:

At 1450 F large quantities of tar, naphthalene, etc. were obtained with no case visible after 4 hr at heat

At 1500 F there was less tar and other undesired products and a trace of low-carbon case after 4 hr at heat

At 1550 F there was very little of tar and undesired products

At 1575 F a satisfactory case on moderately heavy work was obtained in 7½ hr with absence of undesirables

At 1600 F the time was cut to 5 hr for 0.035-in. to 0.040-in. case as indicated by the color test

At 1625 F to 1670 F the corresponding case was obtained in 4 hr

At 1700 F the time was increased to 5 hr, and as the temperature increased above that point the carburizing time increased rapidly with a similar supply of carburizing gas.

The field above 1700 F was not exhaustively investigated with larger gas flow for three reasons:

1 Grain structure of the material is less satisfactory
2 Larger gas flow would be less economical
3 Depreciation of the furnace and equipment would be increased.

For a rotary type of furnace which has a capacity from 500 to 800 lb of work per charge, it has been found that a butane-gas flow of 15 cu ft per hr for the first half of the carburizing period and 8 to 10 cu ft per hr for the last half of the period, gives satisfactory results as to quality and speed instead of the 48 cu ft per hr previously mentioned.[2]

[2] A convenient and inexpensive flowmeter may easily be made for each furnace in the manner illustrated in Fig. 2. A nozzle with reduced area is placed in the gas-flow line and a standard U-gage from any instrument maker with a 10-in. range is attached to the flow line on either side of the nozzle so that the U-gage measures directly in inches of water the pressure drop through the nozzle. Suitable calibration makes it easy to read in cubic feet per hour for a given size of nozzle. Exact calibration will vary not only with the diameter of the nozzle aperture but with its length, smoothness of finish, angle and freedom from burrs at the ends, etc. A nozzle of the dimensions shown in the sketch, with plain drilled hole and no especial care given to finish, will pass approximately the following quantities at atmospheric pressure from an initial pressure of 8 in. of water, viz.:

6 in. water pressure drop as shown by U-gage, 23 cu ft per hr; 4 in. drop, 19 cu ft; 2½ in. drop, 15 cu ft; 1 in. drop, 9 cu ft; ½ in. drop, 6 cu ft.

It appears that the rate of increase of thickness of the case, when butane or equally potent carburizer is used, is principally dependent upon the ability of the steel to absorb and diffuse the carbon at the operating temperature. A higher carbon content than is usually desired is easily and quickly obtained near the surface of the steel, but it seems necessary to allow a rather specific length of time for the carbon to penetrate into the steel and for the high carbon content near the surface to properly diffuse.

This is the reason for the reduced rate of gas flow during the latter part of the process time; i.e., for practical reasons of speeding up the operation, the gas during the first part of the operation is fed as fast as the steel will take up the carbon, resulting in too high a carbon content near the surface, but more rapid penetration.

When a sufficient total amount of carbon has been added to finally give the desired concentration and depth of case, the supply of carburizing gas must be reduced to an amount which will just maintain a satisfactory carbon concentration near the surface of the steel during the time which is necessary for the surplus carbon near the surface to become properly diffused. Under some conditions, if the carburizing retort is gas tight, the supply of carburizing gas may be entirely cut off during the latter part of the diffusion period.

The author is frequently asked about the amount of free carbon or lampblack present. There is only a small amount of this material present if the proper gas flow is used. In the furnace just mentioned about a pint of loose, flocculent lampblack may be present at the completion of the run, which for this quantity of charge means that the work is comparatively clean and that there is much less dust and dirt than from a corresponding batch using solid carburizing material.

The current prices of liquid butane in carload lots in the industrial district between Chicago and New York inclusive, range from 6 to 9 cents per gallon, delivered on the customer's siding. The price varies with the freight rate from shipping point to customer's siding, and to some extent with a sliding price scale depending upon the customer's monthly requirements. One gallon of liquid butane is equal to about 32 cu ft of butane gas at atmospheric pressure; thus for a 4-hr heat of the size described above about a gallon and a quarter or not more than 10 cents worth of butane will be used. For corresponding results with solid commercial carburizers at the present market price of approximately $50 per ton, about 50 to 55 cents' worth of new solid carburizing material will be required. Thus the cost of butane as a carburizing material is only about one-fifth that of solid carburizer at present market prices. Labor costs are also much in favor of the gas. Where a deep case is required, solid carburizer loses its effectiveness so that the work may have to be repacked with fresh carburizer one or more times to obtain the required depth. With gas carburizers conditions are at all times under positive control, and the process may be continued without interruption until the desired depth of case is obtained.

Conclusion

From the foregoing it appears that:

A quarter-inch needle valve adds greatly to the convenience and accuracy of regulation.

It is further suggested that unless an unusually sensitive and accurate pressure control has been placed in the main gas-supply line, the carburizing-gas feed be taken from the high-pressure side of the main reducing valve as the pressure variations due to large units going on and off the line will probably be less pronounced at that point and the proposed quarter-inch needle valve will easily handle the higher pressure, and a steadier flow of carburizing gas will result.

1 The possible temperature range with butane is smaller than with the usual commercial solid carburizers.

2 The rate of flow of carburizing gas should be controlled within suitable limits for best results (see Fig. 2).

3 Both temperatures and rate of feed have sufficiently wide ranges to be easily met commercially as well as experimentally.

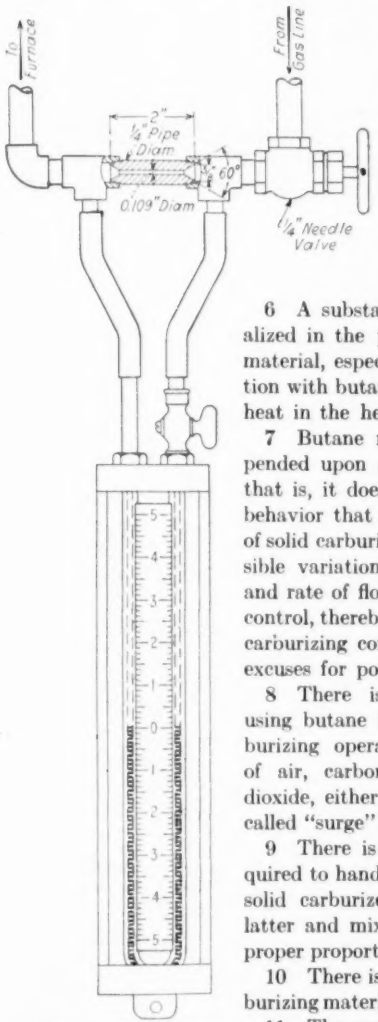


FIG. 2 FLOWMETER

4 When so controlled, butane gas offers an exceedingly satisfactory carburizing material at low cost.

5 At the maximum carburizing rate, the temperatures are advantageous both metallurgically and economically.

6 A substantial saving may be realized in the purchase of carburizing material, especially if used in connection with butane as a general source of heat in the heat-treating department.

7 Butane may be practically depended upon as chemically constant, that is, it does not exhibit the erratic behavior that always annoys the user of solid carburizer. The principal possible variations, namely, temperature and rate of flow, are within the user's control, thereby permanently removing carburizing compound from the list of excuses for poor results.

8 There is no advantage when using butane in complicating the carburizing operation by the admixture of air, carbon monoxide, or carbon dioxide, either continuously or by so-called "surge" methods.[3]

9 There is no labor or space required to handle and store new or old solid carburizer, nor to reclaim the latter and mix it with new material in proper proportions.

10 There is no deterioration of carburizing material in storage.

11 The carburizing department, in general, is much cleaner when a gas carburizer is used. In fact, the carburizing, quenching, and rehardening machines may now be placed in the regular line of manufacture if desired instead of being segregated as usual, thus saving interplant transportation.

12 By suitable control of temperature and gas flow the depth, type, and character of case are in absolute control at all times. A properly graduated case is easily obtained, and the surface may be hypereutectoid, eutectoid, or hypoeutectoid.

13 Under proper conditions the penetration of carbon is as rapid as by any other commercial process known to the author, and because there is no exhaustion of the carburizing material, the rate of penetration may be maintained at maximum.

[3] The carburizing gas is simply flowed through the work at the proper rate after the work has reached the operating temperature. The formation of slight scale caused by oxygen present in the furnace while heating up may be avoided by flushing the air out of the furnace immediately after loading. This may be done by allowing the butane to flow into the retort until it may be lighted and burns with a bright yellow flame at the vent, and then closing the retort until the operating heat has been attained.

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Appendix

The facts brought out in this paper raise some interesting questions which may be definitely answered only by further research. As noted in the paper, methane begins to change or crack at a temperature of about 1200 F, whereas ethane, propane, and butane begin to crack at temperatures roughly 350 deg lower. Methane apparently is more stable at all temperatures below 2000 F than are the other hydrocarbons mentioned. Furthermore, methane is one of the important products resulting from the changes which occur in the other hydrocarbons mentioned at temperatures around 1600 to 1700 F. It is claimed by some authorities that the methane does not radically change to a great extent until temperatures upward of 2000 F are reached. It therefore seems probable that the change from ethane, propane, or butane to methane, which contains a much smaller percentage of carbon, is the real source of the nascent carbon required for carburization of the steel with these gases when no oxygen is used. Apparently unless the supply of carburizing gas is greatly increased so that the time during which the gas is exposed to the heat is reduced, the change to methane is so nearly complete that only a comparatively small amount of nascent carbon is available by the time the gas has attained temperatures upward of 1700 F.

These inferences are given weight by the fact that although artificial coal gas may contain as high as 50 per cent of methane, it is not always a satisfactory carburizing medium unless the carbon combination has been changed from a hydride to an oxide, carbon monoxide apparently being sufficiently unstable within the carburizing temperature range that it gives up its carbon to the steel, thereby changing to CO_2.

Another question which naturally arises relates to the successful use of natural gas as a carburizing means without the addition of oxygen. In the case of at least one large user of natural gas for carburizing without oxygen, it is known that the natural gas from the West Virginia field from which that company receives its supply contains from 17 to 33 per cent of ethane, which in its behavior under heat appears to be much nearer the butane than the methane range; also that this gas is successful as a carburizing medium at temperatures from 75 to 100 deg higher than butane under other similar conditions. This temperature difference might be expected theoretically between butane and ethane (the author has had no experience with the use of ethane only). Furthermore, the Btu value has apparently little or no relation to the carburizing value.

In view of the foregoing, the question naturally arises as to whether the methane content of the natural gas is in fact useful as a carburizing material when used without the admixture of oxygen or whether the carburizing is the result of the other portions of the gas, principally ethane in the case mentioned, breaking down into methane and thereby liberating nascent carbon. If this is so, the methane might add to the efficiency of the ethane by increasing the volume of gas passing the work and adding to the turbulence of the mixture and thereby causing more rapid change of those portions of the gas in immediate contact with the steel.

These questions are offered as being possible logical inferences from the known facts and as suggestions for further interesting and valuable research work.

Discussion

R. J. Cowan.[4] The paper describes an interesting series of practical experiments and records some important results. It would have been well to have mentioned that in addition to carburization by means of carbon monoxide (CO), carburization, by means of methane (CH_4) or other hydrocarbon, is commonly recognized, that each of these gases produce a characteristic type of case, and that by combining the action of the two a wide variety of case types can be produced when the process is under accurate control. As it stands, the paper is an excellent contribution to one phase of the subject and calls attention to certain limitations that may well be considered.

The fact that the case depth dropped off when the temperature was raised beyond a certain point is in line with some work we have done using propane only for carburizing. We found that we had less case at 1750 F than at 1650 F, although the same time at the higher temperature should have produced a much deeper case.

We have had no experience with the addition of air to these reactions, for, in line with the experiment described by the author, we too were suspicious of partial combustion effects. For this reason in the process of continuous gas carburizing, referred to by the author, we use flue gas instead of air, and since this flue gas contains about 10 per cent carbon dioxide (CO_2), we are able to bring about a reaction whereby nascent carbon monoxide (CO) is produced, which makes it possible to regulate both the methane and the carbon-monoxide carburizing by properly proportioning the hydrocarbon and flue gases used.

It has been our experience that carbon-monoxide carburizing as described produces a case that is characterized by rapid penetration and excellent diffusion between case and core, and that when properly coupled with hydrocarbon, carburization will overcome the tendency toward slow reaction rates at the higher temperatures that is encountered when using propane or butane alone.

The author calls attention in a very interesting way to the many decomposition products formed when a hydrocarbon gas is heated. Our work has confirmed the importance of this observation and emphasized the effect of alloys on these reactions. Various oils and tars are produced in the presence of certain metals, but not in the presence of others. In the presence of nickel, methane begins to break down at a temperature of about 725 F, and the effect is marked in iron-nickel alloys.

[4] Surface Combustion Corporation, Toledo, Ohio.

European Supplement to Progress Report of the Iron and Steel Division, 1931

By W. TRINKS,[1] PITTSBURGH, PA.

EVEN though a man might visit the European steel works regularly every summer, he would not be able to see all that is new, because it is physically impossible to cover the whole territory in one summer. And if each plant is visited only once every three years, what is noted as an innovation on the occasion of a visit may be almost three years old—and still be new to the visitor. For that reason the following report is to be accepted with caution, but it is based upon the observations of the writer.

About two years ago a large tube mill in Germany adopted a phenol-formaldehyde condensation product (similar to Bakelite) for rustproofing pipes. The material had the disadvantage of having to be baked, which was very inconvenient for welded or screwed joints. The material in question has been superseded by chlorinated rubber, which is dissolved in benzol. It is brushed (by a machine) on the pipe, both inside and outside. The coating protects the pipe against acids and alkalis. The process will be open for licensing in the United States after the patent situation has been cleared. United States patents on a similar process are owned by an American company.

The same tube company has perfected its process for extruding seamless tubes. The process is a few years old, but troubles with the material of the dies and plungers have prevented its commercial use. These troubles have now been largely overcome, and a die now makes 500 tubes. The plunger (internal die) lasts through the extrusion of 900 tubes. A tube $2^1/_4$ in. o.d., 20 ft. long, is extruded in 2 sec. At present the tubes are reheated before they are rolled down continuously (in reducing or "sinking" mills) to $^3/_8$-in. gas pipe. A new press is being installed, however, to extrude 4-in. tubes, which will be rolled down without reheating.

The same condition (with regard to relative newness) applies to the expanding mill which uses a tension mandrel. It is several years old, but high-class tubes of absolutely uniform wall thickness have been made only recently. Tubes are expanded up to 24 in. in diameter, beginning with a 13-in. tube blank. The first pass expands from 13 in. to 18 in., and the second pass from 18 in. to 24 in.

In the production of seamless tubes, the quality of the steel heretofore has had to be very good. As a matter of fact, people bought seamless tubes because the very existence of a seamless tube was a guarantee of good material. Apparently, the piercing mill accentuates initially existing flaws in the steel. To circumvent this troublesome dependence upon the material, the tube mill at Anzin (near Valencienes, France) has built a large press in which the ingot is pierced by a plunger before it is cross-rolled. What used to be the piercing mill has now become an expanding mill.

The United Steel Works has improved the Roeckner process of rolling seamless drums up to 22 ft. in length and 44 in. in diameter, in thicknesses up to 3 in. or even more. The smallest possible wall thickness is at present $1^3/_{16}$ in., because of the difficulty of keeping thin tubes hot while rolling. Heretofore the drums had a deep grooving (from the cutting-in rolls) on the inside. This grooving is now removed by a second pass. An innovation in the making of superior strength drums consists in boring out a solid ingot, reheating, and rolling on the Roeckner mill. For such drums prices up to $170 per ton have been paid.

One of the most interesting developments is the automatic welding of wide-flanged beams by automatic machinery. The flanges have nosepieces between which the web is inserted. Carbon electrodes melt part of the nosepieces, fastening the flanges to the web partly by resistance welding and partly by fusion welding. The plan is to roll medium-sized girders (up to 12 in. by 12 in., for instance) and to weld the larger ones.

While these developments are outstanding and spectacular, much detail progress has been made and previously mentioned processes have been placed on a commercial basis.

For example, the method of blowing blast-furnace dust back into the furnace at a good distance below the stock line, by means of compressed gas, may be cited. This process is over two years old, but has proved out very well in practice and is now commercially perfect.

Operation of the five valves of a blast furnace is slow. These valves are the valves for gas, for air, the snort valve, the cold blast, and the hot blast. A pneumatic control has been introduced in which a fraction of the turn of a handwheel causes these valves to move automatically in the right sequence and without loss of time. This invention, as most of those previously mentioned, is a few years old, but the original designs were a bewildering maze of pipes; and it is only within the last year that simplifications have been effected which make the control useful.

In blooming-mill practice, attention is being paid to the possibility of bringing the ingot to the mill, either little end first (as we always do) or big end first. While the reasons are purely metallurgical, the means for accomplishing the desired end is purely mechanical. The ingot tilter can tilt forward, as is usual with us, or backward, away from the mill. In the latter case, the tilter is turned through 180 deg. and presents an unbroken roller table all the way to the mill. The thought is good, but the scheme is not perfect, because we should always enter the ingot into the mill small end first, and turn it after the first two squaring passes.

A process in which the mechanical engineer is interested only by furnishing the equipment is a method of regaining sulphuric acid from pickling tanks by crystallizing out the iron sulphate. This desirable separation is accomplished by two-stage cooling and stirring. In the low-temperature stage, brine is circulated (which fact necessitates the use of a refrigeration machine). The iron sulphate crystallizes like granulated sugar. The finished mass is emptied on a chromium-iron sieve through which the acid percolates and upon which the sulphate is retained. This method should help to prevent the dumping of sulphuric acid and iron sulphate into our rivers.

While we pride ourselves about multi-pass continuous mills with horizontal and vertical rolls, we must admit that a mill has been developed in Germany which, to use a slang expression, goes us one better. The housings which carry the vertical rolls rest on slides which are inclined at 45 deg. with the horizontal. A screw adjustment slides the housings on these inclines. In consequence, corresponding passes in the vertical and horizontal rolls are brought into correct alignment by a single adjustment. The

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motors for driving the vertical rolls are located on top of the housings, which consist of four columns. Motor and drive can be swung out of the way very quickly for changing rolls.

The foregoing instances of European progress in the iron and steel industry have been mentioned because they are representative or outstanding. They do not by any means include all of the developments that have taken place in recent years. If there are any questions regarding other developments or in regard to the details of the processes that have been mentioned, the writer will be glad to answer them if possible.

Discussion

A. G. Witting.[2] Professor Trinks has given some extremely interesting information about the progress in rolling-mill design that he found during his visit to steel works in Europe. One could not excape noticing an acknowledgment of superiority for the engineers on the other side. Consequently, when mention was made of the automatic reversal, in a set sequence of the various operating valves, on a hot-blast valve which here in America invariably is reversed by hand, contrasting it with our automatic control for open-hearth furnaces, which according to German opinion just as well can be attended to by the furnace operator, one necessarily got the impression that we, in both instances, were on the wrong track.

The writer takes the liberty of disagreeing with Professor Trinks. Nowadays the temperature of the hot blast for a blast furnace is automatically maintained, with practically no variation after the control has once been set. How this temperature in the air delivered to the furnace is maintained or how it is derived does not matter to the furnace. Furthermore, with a complete automatic control, even if reversals were done automatically, taking the impulse from the stove-stack temperature, it would always be necessary to maintain a stove tender, if for nothing else than to watch that the automatic control is functioning. If this man occasionally is occupied in reversing the stove valves or is just looking on while they are reversed automatically, the result will be practically the same, and there is very little justification for the expense involved by an automatic contraption performing this work.

Entirely different conditions prevail, in the writer's opinion, for an open-hearth furnace. The adoption of automatic control for regulating combustion is a relatively new development and has not yet passed through the stage of experimentation. It is believed, nevertheless, that what has been tried has given good results, particularly balanced draft control. We completed a new open-hearth plant last spring. There has been obtained a remarkably low fuel consumption, using producer gas as exclusive fuel—50 to 75 lb. of coal per ton of steel less than previous record figures. While several factors contribute to this result, it may be said that the balanced draft control has an important bearing on it. Combustion is carried on under continuously changing conditions. We start with a cooler furnace filled with cold scrap; we want to bring it back to the proper temperature and melt the charge as quickly as possible, giving the furnace all the fuel that efficiently can be burned. But at the end of the heat the furnace cannot receive more fuel than is necessary to replace radiation losses. Another factor contributing to the irregularity of waste-gas production is the furnace boil—i.e., the development of CO_2 and CO gases in the bath itself.

If there is not maintained an equilibrium in the furnace between the gases (fuel and air) brought to the furnace and the gases removed from the furnace by stack draft or, better, by an exhaust fan, there is bound to be a loss of fuel. If there is a plus pressure in the furnace room, combustible gases will leak out around the doors; if there is a minus pressure, cold air will leak in. In either case a loss in fuel efficiency results. To judge these conditions, to constantly watch them and to constantly regulate the draft, is too much to be expected of the furnace operator, who has other duties to perform. Hence, the actual value of automatic control that at all times tries to maintain a neutral pressure in the furnace room.

Professor Trinks indicated that in Germany the furnaces have ports and checker chambers encased in plate work to prevent leakage. We do the same; we also insulate the checkers and flue to reduce radiation losses. But with the necessity of large furnace doors that have to be moved often and quickly it is not possible to make them tight.

It also may be pointed out that while flow meters that tell the operator the volume of gas and air going to the furnace may be of a distinct help to the operator in regulating and maintaining the proper combustion ratio, such meters are applicable only to coke-oven, blast-furnace, or natural gas, but not to producer gas. Furthermore, while they will undoubtedly be of great assistance for an alert operator to prevent waste of fuel, they will not tell anything about the leakage from or into the furnace room.

Much more could be added on the advantage of automatic open-hearth control, but it is believed that enough has been advanced to warrant the contention that automatic regulation is more important for open-hearth furnaces than for hot-blast stoves and that in that respect we apparently are more up-to-date than the European steel-works engineers.

John H. Hitchcock.[3] Professor Trinks has mentioned that, in the Roeckner process of rolling seamless drums, rolls are used both inside and outside the drum. The writer would like to ask how such rolls are shaped, and whether they are placed with their axes parallel to the axis of the drum.

H. W. Smith.[4] Professor Trinks states that a tube of $2^1/_4$ in. o.d. and 20 ft. long is extruded in 2 sec. The writer would like to know how many tubes could be made in eight hours.

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[4] Chief Engineer, Aetna-Standard Engineering Company, Youngstown, Ohio. Mem. A.S.M.E.

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Effect of Reradiation on Heat Transmission in Furnaces and Through Openings

By H. C. HOTTEL,[1] CAMBRIDGE, MASS., and J. D. KELLER,[2] PITTSBURGH, PA.

In heat transmission by radiation, the reradiation of heat from adjacent surfaces has been a perplexing question. Radiation of heat through openings in furnace walls, radiation in electric furnaces having heating elements under the roof only, and radiation in some types of oil stills from muffles on the floor to the tubes under the roof are all cases in point. In this paper accurate methods of calculation are developed, and curves are presented from which the reradiation and total radiation factors can be read at a glance, for various shapes and proportions of radiating surfaces or openings. Incidental to the chief results, two methods of solving integral equations are presented. The theoretically developed values are checked and substantiated by actual test measurements.

MUCH excellent work has been done on heat transmission by radiation between surfaces of various shapes. This appears to have been limited, however, to direct radiation between two surfaces, each assumed to be at constant temperature throughout. Two distinct problems have called for attention, which involve not only direct radiation but also reradiation of heat from adjacent surfaces the temperature of which is not constant throughout. These two problems are (1) the radiation of heat from electric resistors under the roof of a furnace to the cold charge on the hearth or from a silicon-carbide muffle at the bottom of an oil still to the oil-containing tubes at the top and (2) the radiation of heat through openings in furnace walls. The first problem has to do with heat transfer within the furnace or still, and hence with heating capacity. Radiation through openings, on the other hand, is usually thought of as involving fuel economy only, as affected by loss of heat through doors and burner openings and also through cracks in the furnace walls. A more important form of this second problem, however, relating to furnace capacity and uniformity of heating, also has to do with useful heat transfer within the furnace itself, by radiation through heat ports or perforations in arches or muffles.

The second problem was solved about five years ago (1)[3] by approximate methods which gave results in most cases within 1 per cent of those obtained by the accurate methods of the present paper, for round openings, but which could not well be applied to other shapes of opening.

Mathematically, the two problems are identical; for the resistors or muffle, in the first problem, or the interior of the furnace, in the second, as the radiator, and the cold charge or tubes, in the first problem, or the surfaces of the exterior surroundings, in the second problem, as the receiver, can both be replaced by flat plane surfaces, as will be shown. The problem then involves finding the radiation between two opposite parallel plane ends of a hollow geometrical figure, such as a cylinder or a rectangular prism, taking account, however, not only of the direct radiation between the two end surfaces but also of the heat which is radiated from one end surface to the connecting sidewalls, absorbed, and reradiated by them to the other end (see Figs. 1 and 3).

The exact solution of the problem requires far more time than engineers can afford to spend on it. The object of this paper is, first, to make available for engineering use the results of accurate methods of solution, in the form of charts from which the radiation factors applying to various shapes and proportions of furnace chambers or of openings can be read at a glance, and at the same time to illustrate methods of solving integral equations, such as appear in this problem, in the hope that they may be more widely utilized for the solution of other more important engineering problems which involve such equations and which are as yet unsolved.

The factors derived mathematically have been checked experimentally.

1 Results and Methods of Using Factors

Radiation out of a hollow space such as a furnace interior, when the opening is small compared to the interior dimensions, is "black body" radiation; that is to say, it is equivalent to heat transfer between surfaces which absorb all of the heat that is radiated to them, and reflect none whatsoever, and which, conversely, radiate the greatest amount of heat which it is possible for any surface at that temperature to radiate. For radiation between solid surfaces within the furnace, the radiation is somewhat less than that of the black body; usually about 90 per cent of the latter, for most surfaces found in practice.

As far as concerns radiation through an opening, for reasons explained later the furnace interior can be regarded as equivalent to a flat diaphragm having the same temperature, and located at the inner edge of the opening, as 1–2–3 in Fig. 1; and for radiation between solid surfaces this is self-evidently the case. If this equivalent diaphragm were fully exposed—that is, if all points on the surface could radiate unobstructedly through a full hemisphere, or solid angle of 2π—the amount of heat radiated, according to the well-known "fourth power" or Stefan-Boltzmann law, would be:

$$Q_{\text{Btu per hr}} = 0.172A\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \quad [1]$$

for radiation from an area of A sq ft at T_1 deg F abs temp to surroundings at T_2 deg F abs.

The side walls, however, intercept part of the radiation, and although they reradiate a fraction of it, the amount of heat radiated is always less than that from freely exposed surfaces. Equation [1] must therefore be multiplied by a factor which is always less than unity and which depends on the shape of the surfaces

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[2] Engineer with Prof. W. Trinks at Carnegie Institute of Technology. Mr. Keller was graduated from Carnegie Institute of Technology in 1914 with the degree of S.B., and received his M.E. degree from that institution in 1917.

[3] Numbers in parentheses occurring in the text refer to bibliography at end of paper.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

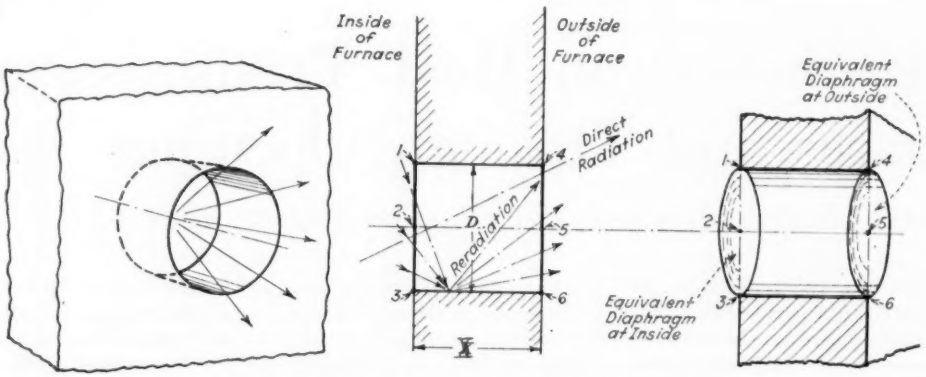


FIG. 1 OPENING IN WALL AND EQUIVALENT DIAPHRAGMS

and on the ratio of their width or diameter to the distance between them. The curves of Fig. 2, embodying results obtained by the methods explained in sections 2 to 6, give values of the total radiation factor for hollow spaces of uniform cross-section and surfaces of four different shapes—namely, the circular, the square, the rectangle of 2-to-1 ratio of sides, and the very long strip (corresponding to a long slot, in the second problem). The abscissa is the ratio of the diameter or least width of the end surfaces (D, Fig. 1) to the distance between them (X, Fig. 1).

The values shown by Fig. 2 are based on heat transfer by radiation only. In any hollow space (except in a vacuum) there is additional heat transfer to the sides of the space by convection of hot gases or air, and this additional heat is also reradiated to the colder end. Tests described in section 8 show that in round openings with axis horizontal the increase due to convection varies from 4 per cent in openings of which the diameter is one-half the wall thickness to 7 per cent in openings of diameter equal to the wall thickness, while in rectangular openings the increase in the former case is about 9 per cent (see section 8).

The method of using these factors will become clear from the following examples.

Example 1. The charging door of a skelp-heating furnace is 7 ft 0 in. wide, with an average height of 9 in., and is open at all times. The end wall is 9 in. thick. The temperature of the furnace interior is 2600 F. How much heat is lost through the opening?

The ratio of width of opening to thickness of wall is

$$D/X = 9 \text{ in.}/9 \text{ in.} = 1.00$$

Considering the opening as rectangular, the ratio of the sides is 84 in./9 in., or 9.33. Reading from the uppermost curve of Fig. 2, for very long slots and for $D/X = 1.00$, the radiation factor is found to be 0.68. This is increased about 7 per cent by convection heat transfer. The amount of heat radiated is

$$Q = 0.68 \times 0.172 \times \frac{84 \times 9}{144} \times \left[(30.60)^4 - (5.60)^4 \right] \times 1.07$$

$$= 575{,}700 \text{ Btu per hr.}$$

Example 2. In a double-chamber furnace, the combustion chamber is maintained at 2500 F, and it is required to transmit 500,000 Btu per hr to the heating chamber at 2000 F, and to distribute the heat uniformly therein. For this purpose, perforations or heat ports are to be provided in a curtain wall between the chambers. The area of the wall is 48 sq ft, its thickness is 9 in., and the material is firebrick. The ports are to be $2^1/_2$ in. high by $4^1/_2$ in. wide. How many such ports are required?

Let the number of ports be n. The ratio of the sides of the rectangular openings is $4^1/_2/2^1/_2$, or 1.80. The ratio of least width of opening to wall thickness is $2^1/_2/9$, or 0.278. By interpolating in Fig. 2 between the curve for the 2-to-1 rectangle and the square, the total-radiation factor is found to be 0.29. For such long, narrow openings the increase by convective heat transfer is negligible (unless there is actual flow of hot gases through the ports from one chamber to the other). The equation then becomes

$$500{,}000 = n \times \frac{2\frac{1}{2} \times 4\frac{1}{2}}{144} \times 0.172 \times 0.29 \times \left[(29.6)^4 - (24.6)^4 \right] \text{heat radiated} + \left[48 \text{ sq ft} - \frac{n \times 2\frac{1}{2} \times 4\frac{1}{2}}{144} \right] \times \frac{10 \text{ Btu/sq ft, hr., °F/in.}(2500 - 2000)}{9 \text{ in.}} \text{ heat conducted}$$

from which $n = 312$. This means that 312 heat ports are re-

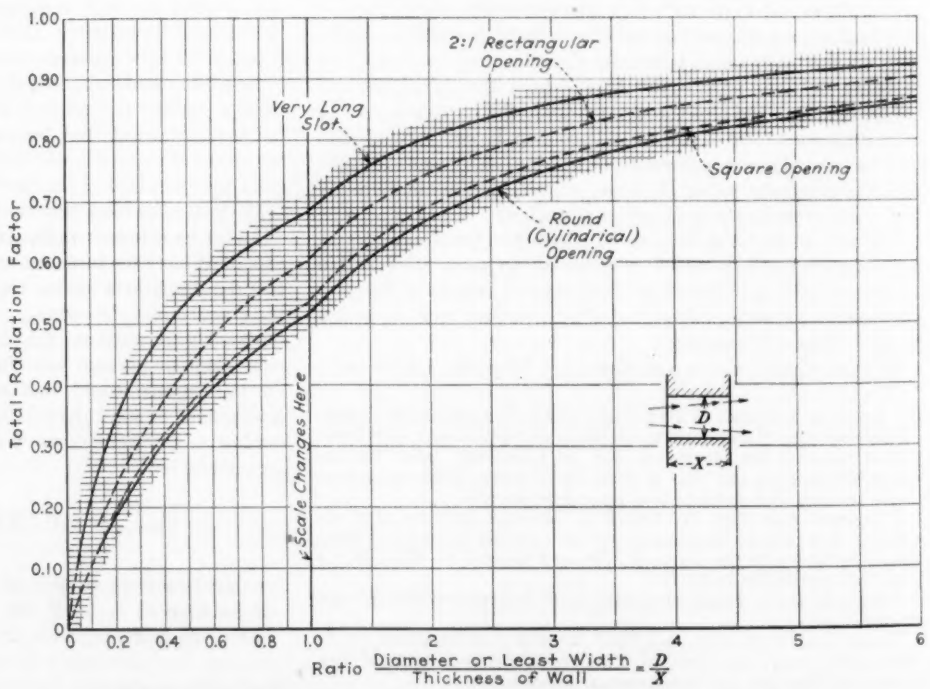


FIG. 2 FACTORS FOR TOTAL RADIATION THROUGH OPENINGS OF VARIOUS SHAPES

quired, having a combined area equal to about 50 per cent of the total area of the curtain wall.

When doors are open only part of the time, a modification of the radiation factor is required, as shown elsewhere.

2 Derivation of Fundamental Equations

The fundamental principles involved in the calculation of radiant-heat interchange between finitely bounded surfaces, and the application of those principles to some of the problems involving direct radiation between various surface arrangements, have been presented by several investigators (2–7). For the present purpose it is sufficient to say that the result, in every case of direct radiant interchange between two perfect radiators, is presentable in the form:

$$q = (\mathbf{E}_1 - \mathbf{E}_2) \cdot A \cdot F \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

in which q is the net direct heat interchange between the two surfaces, $\mathbf{E}_1$ and $\mathbf{E}_2$ their respective emissive powers,[4] A is the area of one of the two surfaces, and F is a factor which has variously been called the interchange factor or angle factor or area factor, and is equal to that fraction of the radiation, leaving A in all directions, which is intercepted within the bounds of the other surface. The factor F is plainly a geometrical one, dependent not on the shape of the radiating and receiving surfaces, but on the solid angle filled by the rays. Whether the radiators are plane surfaces, curved surfaces, or apertures is immaterial to the mathematical problem involved; but for convenience in the derivations both the interior of the furnace and the exterior surroundings (in the case of radiation through an opening in the furnace wall) or the muffle wall or electric resistor and the tube bank or cold stock (in the case of radiation within an oil still or an electric heating furnace) are considered to be replaced by equivalent flat diaphragms, indicated by 1–2–3 and 4–5–6 in Fig. 1. The problem then resolves itself into finding the radiant-heat interchange between two plane parallel black bodies or perfect radiators, maintained at uniform but different temperatures and separated by non-conducting but reradiating walls. The connecting walls will vary in temperature, consequently in emissive power, from one end to the other of the system. The direct radiation is readily calculable, but the reradiation cannot be evaluated without first determining how the emissive power varies along the walls.

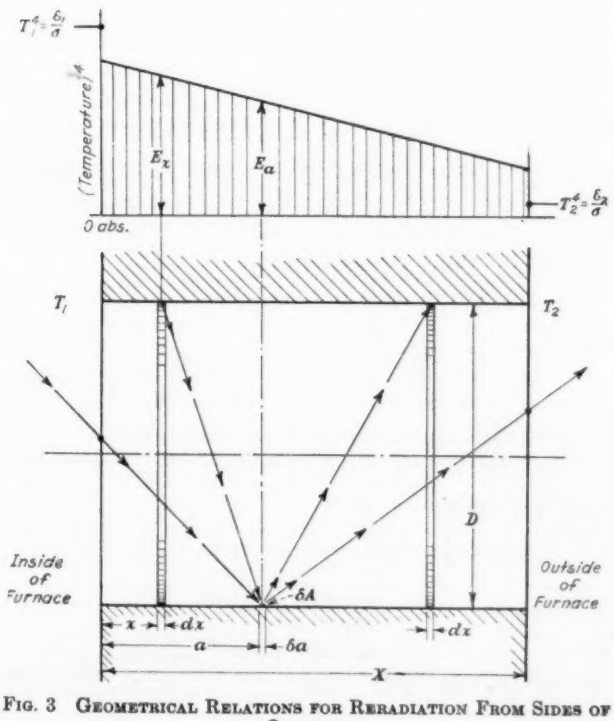


Fig. 3 Geometrical Relations for Reradiation From Sides of Opening

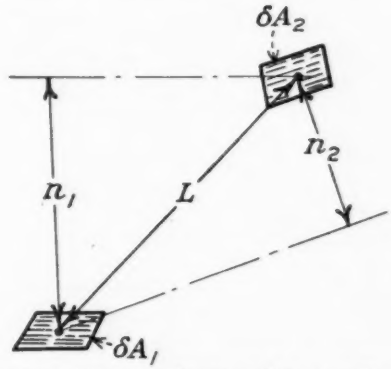


Fig. 4 Radiation Between Surfaces Small in Comparison to Their Distance Apart

The fundamental principles involved are two: namely, that the intensity of radiation is the same in all directions and that the radiation from a given area in a direction at an angle to the normal to the surface is proportional to the cosine of the angle (Lambert's principle). From these it can be shown that for any two infinitesimal black-body elements δA_1, δA_2, at any distance and relative position (Fig. 4) and having emissive powers $\mathbf{E}_1$, $\mathbf{E}_2$, the net heat interchange is

$$\delta q = (\mathbf{E}_1 - \mathbf{E}_2) \cdot \delta A_1 \cdot \delta A_2 \cdot \frac{n_1 \cdot n_2}{L^4} \quad \ldots\ldots\ldots [3]$$

where L is the actual distance between δA_1 and δA_2, n_1 the perpendicular distance from δA_1 to a plane through the center of δA_2 parallel to δA_1, and n_2 the perpendicular distance from δA_2 to a plane through the center of δA_1 parallel to δA_2.

3 Calculation of Direct Radiation

Finding the factor F for direct radiation between any two equal finite surfaces, each at constant temperature over its whole area, such as the equivalent diaphragms 1-2-3 and 4-5-6, requires quadruple integration over the areas. Evaluations of the factor for all of the four shapes considered here have been reported in the literature (2, 5, 7) and are presented in the second column of Table 1. Here, as in all of the following discussion, the smallest dimension D of the areas is taken as unity, and x represents the distance between the radiating areas as a *ratio* to their smallest dimension. The variation of the factor F with the distance ratio x is shown graphically in Fig. 5, for the various shapes.

4 Reradiation From the Side Walls

Let S_1 and S_2 represent opposed plane areas or radiators of equal area S, separated by a distance X, and at absolute temperatures T_1 and T_2, respectively. Part of the heat radiated from area S_1 strikes the side walls, is absorbed by them, and is reradiated from these walls to area S_2. Assuming for the present that the side walls (or sides of the opening) are non-conducting (or at least that the heat conducted by them is negligible compared to the heat radiated), it is evident that the total heat radi-

[4] The emissive power of a surface is its rate of emission of radiant energy per unit of area. For a black or perfect radiator it is equal to σT^4, in which T is its absolute temperature and σ is a constant, the Stefan-Boltzmann constant.

ated *to* any small area δA on the side walls must be exactly equal to the heat radiated *from* δA. Points on the sides near surface S_1 receive heat from it more readily than they radiate heat to surface S_2, while for points on the sides near S_2 the reverse is true.

Referring now to Fig. 3, a heat balance for an element δA, of

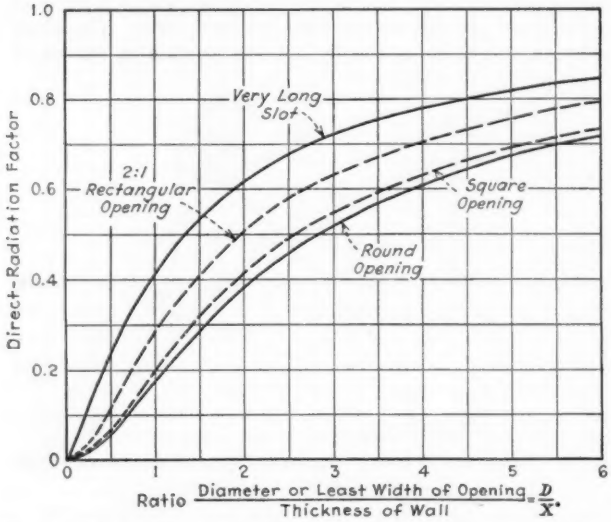


FIG. 5 DIRECT-RADIATION FACTORS FOR OPENINGS OF VARIOUS SHAPES

width δA, located at a distance a from one end of the system, yields the following equation:

$$\underset{\substack{\text{Radiation from}\\ \delta A \text{ in all}\\ \text{directions}}}{E_a \cdot \delta A} = \underset{\substack{\text{Radiation to}\\ \delta A \text{ from end}\\ \text{at } x = 0}}{\mathbf{E}_1 \cdot \phi(a) \cdot \delta A} + \underset{\substack{\text{Radiation to } \delta A\\ \text{from sides between}\\ x = 0 \text{ and } x = a}}{\int_0^a E_x \cdot f(a - x) \cdot \delta A \, . \, dx} + \underset{\substack{\text{Radiation to } \delta A\\ \text{from sides between}\\ x = a \text{ and } x = X}}{\int_a^x E_x \cdot f(x - a) \cdot \delta A \cdot dx} + \underset{\substack{\text{Radiation to } \delta A\\ \text{from end at } x = X}}{\mathbf{E}_2 \cdot \phi(X - a) \cdot \delta A}$$

where

$\mathbf{E}_1$ and $\mathbf{E}_2$ are the fixed emissive powers of the end surfaces or apertures S_1 and S_2

E_a (or E_x) is the variable emissive power of the connecting wall, at $x = a$ (or x)

$\phi(x)$ is a fraction, of direct radiation from an element of area δA of width δx, running around the walls connecting the planes S_1 and S_2, which is intercepted by an area S at distance x from δA

$f(x) \cdot \delta x'$ is a fraction, of radiation from an element of area $\delta A (= P \cdot \delta x)$ around the walls, which is intercepted by another element of area $\delta A' (= P \cdot \delta x')$ around the walls, and separated from the first by a normal distance x

P is the perimeter of radiator S_1 or S_2.

Dropping out the δA, the equation becomes:

$$E_a = \mathbf{E}_1 \cdot \phi(a) + \mathbf{E}_2 \cdot \phi(X - a) + \int_0^a E_x \cdot f(a - x) \cdot dx + \int_a^X E_x \cdot f(x - a) \cdot dx \quad \ldots\ldots [4]$$

The functions $\phi(x)$ and $f(x)$ are related to each other and to the function $F(x)$. If the fraction $F(x)$ of the radiation from S_1 is intercepted by S_2, the fraction $-\dfrac{dF(x)}{dx} \cdot \delta x$ will be intercepted by the side-wall strip of width δx, lying between the two plane areas S_2 located at distances x and $x - \delta x$, respectively, from the radiator S_1. By the principle of reciprocity this latter fraction, multiplied by the area S, must equal the area δA multiplied by the fraction $\phi(x)$ of its radiation which is intercepted by S. Therefore:

$$-S \cdot \frac{dF(x)}{dx} \cdot \delta x = \phi(x) \cdot \delta A$$

Since area δA equals $P \cdot \delta x$, it follows that

$$\phi(x) = -\frac{S}{P} \cdot \frac{dF(x)}{dx} = -\frac{S}{P} \cdot F'(x) \quad \ldots\ldots [5]$$

Similarly, if $\phi(x)$ represents the fraction of the radiation from δA which is intercepted by S at a distance x from δA, then $-\dfrac{d\phi(x) \cdot \delta x'}{dx}$ represents the fraction intercepted by the strip of area $\delta A' (= P \cdot \delta x')$, between two surfaces S located at distances x and $x - \delta x$, respectively, from radiator δA. This fraction, by definition, is equal to $f(x) \cdot \delta x'$. Then

$$f(x) \cdot \delta x' = -\frac{d\phi(x)}{dx} \cdot \delta x'$$

$$f(x) = \frac{-d\phi(x)}{dx} = -\phi'(x) = \frac{n \cdot D}{2(n+1)} \cdot F''(x) \quad \ldots\ldots [6]$$

where n is the ratio of the long to the short side of the area S.

While the factors $\phi(x)$ and $f(x)$ can be obtained by integration

TABLE 1 INTERCHANGE FACTORS

Shape	$F(x)$	$\phi(x)\left[= -\frac{n}{2(n+1)} \cdot F'(x)\right]$	$f(x)\,[= -\phi'(x)]$
Disks.....	$1 + 2x^2 - 2x\sqrt{x^2+1}$	$\left(x^2 + \frac{1}{2}\right) \Big/ \sqrt{x^2+1} - x$	$1 - x\left(x^2 + \frac{3}{2}\right)(x^2+1)^{3/2}$
Squares.....	$\frac{1}{\pi}\left[x^2 \log_e \frac{(1/x^2+1)^2}{2/x^2+1} - 4x \cdot \tan^{-1}x + 4\sqrt{x^2+1} \cdot \tan^{-1}\frac{1}{\sqrt{x^2+1}}\right]$	$\frac{1}{\pi}\left[\frac{x}{2}\log_e \frac{x^2(x^2+2)}{(x^2+1)^2} - \frac{x}{\sqrt{x^2+1}}\tan^{-1}\frac{1}{\sqrt{x^2+1}} + \tan^{-1}\frac{1}{x}\right]$	$\frac{1}{\pi}\left[\frac{1}{(x^2+1)^{3/2}} \cdot \tan^{-1}\frac{1}{\sqrt{x^2+1}} + \frac{1}{2}\log_e \frac{(x^2+1)^2}{x^2(x^2+2)}\right]$
Rectangles, $1 \times n$....	$\frac{1}{\pi}\left[\frac{x^2}{n}\log_e \frac{(1/x^2+1)(n^2/x^2+1)}{(n^2+1)/x^2+1} - 2\frac{x}{n}\tan^{-1}\frac{1}{x} - 2x \cdot \tan^{-1}\frac{n}{x} + 2\sqrt{1+x^2/n^2}\tan^{-1}\frac{1}{\sqrt{n^2+x^2}} + 2\sqrt{x^2+1}\tan^{-1}\frac{n}{\sqrt{x^2+1}}\right]$	(for $n = 2$) $\frac{1}{3\pi}\left[x \cdot \log_e \frac{x^2(x^2+5)}{(x^2+1)(x^2+4)} + \tan^{-1}\frac{1}{x} - \frac{2x}{\sqrt{x^2+1}}\tan^{-1}\frac{2}{\sqrt{x^2+1}} + 2 \cdot \tan^{-1}\frac{2}{x} - \frac{x}{\sqrt{x^2+4}}\tan^{-1}\frac{1}{\sqrt{x^2+4}}\right]$	(for $n = 2$) $\frac{1}{3\pi}\left[\frac{4}{(x^2+4)^{3/2}}\tan^{-1}\frac{1}{\sqrt{x^2+4}} + \frac{2}{(x^2+1)^{3/2}}\tan^{-1}\frac{2}{\sqrt{x^2+1}} + \log_e \frac{(x^2+1)(x^2+4)}{x^2(x^2+5)}\right]$
Infinite strips or slots...	$\sqrt{x^2+1} - x$	$\frac{1}{2} - \frac{x}{2\sqrt{x^2+1}}$	$\frac{1}{2(x^2+1)^{3/2}}$

of the fundamental Equation [3], they can also be arrived at, in a more simple manner, by successive differentiations of the expression for $F(x)$. The resulting expressions for $\phi(x)$ and $f(x)$ applying to the various shapes which are to be considered—circular disks, squares, 2-to-1 rectangles, and 1-to-∞ rectangles or infinite strips (slots, if the radiators are apertures instead of surfaces)—are presented in Table 1. The quantity x is the ratio of the distance between the surfaces to their smallest dimension D, which is assumed as unity.

If, as stated above, n is the ratio of long to short side of the area S, then S/P equals $nD/2(n + 1)$ for all four shapes if n is considered unity for circles.

If Equation [4] can be solved to give E_a as a function of a, the relationship thus determined can be used to find the desired total net interchange between the two ends of the system, which will be:

$$q = (\mathbf{E}_1 - \mathbf{E}_2) \cdot F(X) \cdot S + \int_0^X (\mathbf{E}_1 - E_a) \cdot \phi(a) P dA \quad \ldots [7]$$

Equation [4] may be simplified somewhat, without loss of generality, by taking the fixed emissive powers $\mathbf{E}_1$ and $\mathbf{E}_2$ of the two ends as 1 and 0, respectively. The final result will then be the net heat transmission between the two ends, per unit difference of their emissive powers, and can readily be converted to any other form desired. Equation [4] becomes:

$$E_a = \phi(a) + \int_0^a E_x \cdot f(a - x) \cdot dx + \int_a^X E_x \cdot f(x - a) \cdot dx \ldots [8]$$

This is an integral equation[5] of the second, or non-homogeneous, or Poisson-Volterra-Fredholm type, and is to be solved for E_a, after which the net heat interchange may be evaluated from Equation [7].

5 Solution of Equations

Of the methods available for solving Equation [8], one which dodges some complications inherent in an exact solution and which can be extended to any desired degree of precision follows from the assumption that E_a does not vary continuously from $a = 0$ to $a = X$. Instead, the walls from 0 to X are pictured as divided into m zones of equal width, each running completely around the wall and having a temperature (and consequently an emissive power) which is uniform over the zone, and in equilibrium with the rest of the system.

By such a procedure one obtains m equations linear in m unknown values of E, E_1 ..... E_m. Obviously, this procedure yields a solution which approaches the exact truth as m approaches infinity. The number of equations and unknown E's can be halved by including the condition that the emissive power is skew-symmetrical with respect to the center of the system $X/2$, at which point it has a value equal to the arithmetic mean of the emissive powers of the ends.

To try this method of solution the special case was considered of disks with diameters equal to twice the distance separating them. The walls were divided into six zones (three equations). The emissive power was found to vary linearly with the position of the midpoint of each segment, to three significant figures (see Table 2, left half). However, when the condition of linear variation of E was set into the integral Equation [8] in an attempt to solve it, the result indicated that E_x cannot be truly linear with x. To straighten out this point, the walls were divided into eight zones and the calculations carried out sufficiently far to guarantee five significant figures in the final E's obtained. The results, presented in Table 2, show a slight but definite departure of the $E - x$ relationship from a linear one. A comparison of the total change in emissive power along the wall from $x = 0$ to $x = X$ for the two cases, six zones and eight zones, shows agreement to three figures, indicating that six is an adequate number of zones into which to divide the walls.

TABLE 2 RESULTS OF SOLUTION BY DIVISION OF SIDE WALLS INTO ZONES

6 zones		Diff.	8 zones		Diff.
E_1...	0.645	0.058	E_1...	0.65239	0.04302
E_2...	0.587	0.058	E_2...	0.60937	0.04354
E_3...	0.529	0.058	E_3...	0.56583	0.04379
E_4...	0.471	0.058	E_4...	0.52204	0.04408
E_5...	0.413	0.058	E_5...	0.47796	0.04379
E_6...	0.355		E_6...	0.43417	0.04354
			E_7...	0.39063	0.04302
$E_0 - E_X$, by extrapolation, = 0.348			E_8...	0.34761	
			$E_0 - E_X$, by extrapolation, = 0.3473		

With the variation in emissive power known, it is possible to determine the total interchange factor, i.e., the factor by which the free radiation from an unobstructed disk must be multiplied to give the direct radiation plus wall reradiation when the two disks are connected by non-conducting but reradiating walls. This factor is found to be 0.681, 0.677, 0.6725, or 0.6723, depending on whether the connecting walls are divided into one, two, six, or eight zones. From the result it is obvious that a division into six zones is entirely adequate for accuracy to three significant figures when $X = 1/2$.

The above-described method of calculating total interchange factors, although time-consuming, is interesting in its proof of how slightly the $E - x$ relationship varies from a linear one. This fact may be utilized to obtain a relatively simple solution of the general problem which, although inexact from a strict mathematical viewpoint, proves to be highly accurate. Let the total change in E along the connecting walls from $x = 0$ to $x = X$ be represented by Δ. The condition that E varies linearly with x and has a value of $1/2$ at $X/2$ is then:

$$E_x = \frac{1 + \Delta}{2} - \frac{\Delta}{X} \cdot x \ldots [9]$$

If this substitution for E_x is made in Equation [8], the latter becomes, after partial integration:

$$\frac{1 + \Delta}{2} \left\{ \phi(a) + \phi(X - a) \right\} - \frac{\Delta}{X} a - \phi(a) = -\frac{\Delta}{X} \int_0^a x \cdot f(a - x) \cdot dx - \frac{\Delta}{X} \int_a^X x \cdot f(x - a) \, dx \ldots [10]$$

The interrelationship of the functions f, ϕ, and F as given in Equations [5] and [6] permits an evaluation of the integrals appearing in Equation [10] and a solution of the latter for Δ. It is

$$\Delta = \frac{\phi(a) - \phi(X - a)}{\phi(a) - \phi(X - a) + \frac{n}{(n+1)X} [F(a) - F(x - a)]} \ldots [11]$$

If the assumption involved in obtaining Equation [11] were correct, i.e., that E_a varies linearly with a, one would expect

[5] "Integral equations" are not at all the same as ordinary equations containing integrals, differing from them in that the unknown function which is to be determined appears both in the integral and outside it. Integral equations are characteristic of problems involving *action at a distance*, in contradistinction to differential equations which apply only when *non-mediate action* occurs between adjacent points or objects in contact. The first-named equations are therefore especially likely to appear in radiation problems, since radiation is action at a distance. Nevertheless, from the purely mathematical standpoint, integral equations and differential equations often lead to an identical result, and certain problems can be solved equally well by the use of either type of equations. An attempt to reconcile the opposing philosophical concepts of action at a distance and direct or non-mediate action, in the case of radiation, has been made by G. N. Lewis in "The Anatomy of Science," using the relativity theory.

Equation [11] to yield a value of Δ dependent on X alone, with a not involved. It does not, as expected, because E_a is very closely, but not exactly, linearly proportional to a. However, that value of Δ may be determined from Equation [11] which satisfies any particular value of a, such as $a = 0$. Physically, the meaning of the result so obtained would be this: An emissive power gradient would have been determined which would permit the wall strip adjacent to one end to exist in perfect thermal equilibrium, i.e., it would be receiving exactly as much heat from the end at $x = 0$ as it was reradiating to the strips beyond it and to the other end. But no other strip, except at $X/2$ and X, would be quite in equilibrium. If a is set equal to zero, Equation [11] becomes:

$$\Delta = \frac{1 - 2\phi(X)}{1 - 2\phi(X) + \dfrac{2n}{(n+1)X} \cdot (1 - F(X))} \quad \ldots\ldots [12]$$

The same result would be obtained by letting $a = X$. Equation [9], with a value for Δ obtained from Equation [12], although not mathematically exact, shows the variation of emissive power along the walls with an error affecting only the fourth or fifth significant figure.

Of more interest than the emissive power gradient along the walls, however, is the net interchange of radiation between the ends of the system, which will not be sensitive to any slight error in emissive power gradient. Equation [7] gives the heat interchange. If $\mathbf{E}_1$ and $\mathbf{E}_2$ are 1 and 0, respectively, [7] becomes:

$$q = F(X) \cdot S + \int_0^X (1 - E_a) \cdot \phi(a) \cdot P \cdot da \ldots\ldots [7']$$

The above may be compared to the interchange between a radiator of unit emissive power, free to radiate in all directions, and its surroundings of zero emissive power. For a radiator of area S, the free radiation would be $1 \times S$. The ratio q/S may therefore be considered the total interchange factor, and will be called $\mathbf{F}$. From Equation [7'],

$$\mathbf{F} = F(X) + \frac{P}{S} \int_0^X (1 - E_a) \cdot \phi(a) \cdot da \ldots\ldots [13]$$

The total interchange factor $\mathbf{F}$ is seen to consist of the sum of two quantities, the direct radiation factor F and the second term, which may be spoken of as the reradiation factor.

Returning to the approximation method and substituting $(1 + \Delta)/2 - \Delta a/X$ for E_a, and $2(n+1)/n$ for P/S, and partially integrating, one obtains finally:

$$\mathbf{F} = F(X) + \left(\frac{\Delta a}{X} + \frac{1 - \Delta}{2}\right) \cdot (1 - F(X)) + \frac{\Delta}{X} \int_0^X F(a) \cdot da \quad [14]$$

Equations [12] and [14] constitute the desired solution of the general problem, and may now be applied to the solution of the special cases of interest.

Disks. Equation [12], on substitution of $\phi(X)$ and $F(X)$ from Table 1, yields:

$$\Delta = \frac{\sqrt{X^2+1} + 2X\sqrt{X^2+1} - 2X^2 - 1}{\sqrt{X^2+1} + 1} \quad \ldots\ldots [15]$$

Equation [14], on substitution of Δ from [15], $F(X)$ from Table 1, and integration, yields:

$$\mathbf{F} = F(X) + \frac{1 - \Delta}{2}(1 - F(X)) + \frac{4\Delta}{3X}\left(\sqrt{X^2+1}\left(X^2 - \frac{1}{2}\right) - X^3 + \frac{1}{2}\right) \ldots\ldots [16]$$

Strips (or Infinite Slot)

$$\Delta = \frac{X^2}{2(1+X)\sqrt{X^2+1} - X^2 - 2} \quad \ldots\ldots [17]$$

$$\mathbf{F} = F(X) + \frac{1 - \Delta}{2}(1 - F(X)) + \frac{\Delta X}{2} - \frac{\Delta\sqrt{X^2+1}}{2} + \frac{\Delta}{2X} \log_e \left(X + \sqrt{X^2+1}\right) \ldots\ldots [18]$$

The results of the calculations based on Equations [15] to [18] have already been discussed and presented graphically in Fig. 2.

6 More Generally Applicable Method of Solving Integral Equation [4]

Although Equations [12] and [14] constitute a solution which is wholly adequate from an engineering viewpoint, the method used is in a sense a dodge, and would not have been applicable but for the almost perfectly linear variation of emissive power along the walls. A straightforward solution of general applicability is desirable as an indication both of the precision of the above method and of how other engineering problems leading to integral equations may be attacked.

This method, as originally developed by E. T. Whittaker (8), and as used by H. Buckley(9) to find the radiation from the interior of a hollow cylinder at constant temperature, is a logical development from the considerations that the integral of an exponential (e^{kx}) is equal to a constant times the exponential itself, and that the "nucleus" of the equation—that is, the $f(-)$ function—can be approximated to any desired degree of accuracy by a series of exponentials if a sufficient number of terms be used. The method consists of first finding expressions for $f(b)$ and $\phi(a)$ in the form of a series of exponential terms, of known coefficients and exponents; writing E_a also in the form of a series of exponentials, but with unknown exponents and coefficients; substituting these in Equation [4] or [8], integrating, and solving for the unknown constants. The form of E_a is thereby determined. As stated, any desired degree of accuracy can be attained by using a sufficient number of exponential terms. For the present purpose, two terms each for $f(b)$ and $\phi(a)$, i.e.:

$$f(b) = P \cdot e^{pb} + Q \cdot e^{qb} \ldots\ldots [19]$$

($b = (x - a)$ or $(a - x)$, always positive)

$$\phi(a) = R \cdot e^{ra} + S.e^{sa} \ldots\ldots [20]$$

were found to be quite sufficiently accurate for values of D/X less than 5.

The closeness with which the actual values of $f(b)$ and $\phi(a)$ can be approximated by two-term exponential expressions is evident from Table 3, which applies to round openings when $X/D = a = 1$. The exponential expressions for this particular shape and proportion are:

$$f(b) = 1.282e^{-2.405b} - 0.282e^{-6.355b} \ldots\ldots [19a]$$

$$\phi(a) = 0.5305e^{-2.286a} - 0.0305e^{-6.86a} \ldots\ldots [20a]$$

TABLE 3

a or b	Actual $f(b)$, from expressions of Table 1	$f(b)$ from exponential Eq. [19a]	Actual $\phi(a)$ from expression of Table 1	$\phi(a)$ from exponential Eq. [20a]
0.0.....	1.0000	1.0000	0.5000	0.5000
0.1.....	0.8513	0.8586	0.4075	0.4068
0.2.....	0.7097	0.7132	0.3295	0.3283
0.3.....	0.5810	0.5810	0.2651	0.2637
0.5.....	9.3727	0.3629	0.1708	0.1686
0.7.....	0.234	0.2346	0.1111	0.1073
1.0.....	0.116	0.1152	0.0607	0.0543

By Whittaker's method, it was found that when the two-term exponential expressions, Equations [19] and [20], are used for $f(b)$ and $\phi(a)$, the solution of integral Equation [4] has the form:

$$E_a/\sigma = A \cdot e^{\alpha a} + B \cdot e^{\alpha(X-a)} + C \cdot e^{\beta a} + D \cdot e^{\beta(X-a)} + E \cdot e^{\gamma a} + F \cdot e^{\gamma(X-a)} + G \cdot e^{\delta a} + H \cdot e^{\delta(X-a)} \quad [21]$$

the coefficients A to H and the exponents α and β being unknown. E_x is the same except that x is substituted for a in all terms.

Substituting these expressions for E_a and E_x in the integral Equation [4] and integrating, furnishes:

$$\left(1 - \frac{P}{\alpha - p} - \frac{Q}{\alpha - q} + \frac{P}{\alpha + p} + \frac{Q}{\alpha + q}\right) \cdot (Ae^{\alpha a} + Be^{\alpha(X-a)}) + \left(1 - \frac{P}{\beta - p} - \frac{Q}{\beta - q} + \frac{P}{\beta + p} + \frac{Q}{\beta + q}\right) \cdot (Ce^{\beta a} + De^{\beta(X-a)}) = \left(\mathbf{E}_1 R/\sigma - E + \frac{EP}{r - p} + \frac{EQ}{r - q} - \frac{EP}{r + p} - \frac{EQ}{r + q}\right) \cdot e^{\gamma a} + \left(\mathbf{E}_2 R/\sigma - F + \frac{FP}{-r - p} + \frac{FQ}{-r - q} - \frac{FP}{-r + p} - \frac{FQ}{-r + q}\right) \cdot e^{\gamma(X-a)} + \left(\mathbf{E}_1 S/\sigma - G + \frac{GP}{s - p} + \frac{GQ}{s - q} - \frac{GP}{s + p} - \frac{GQ}{s + q}\right) \cdot e^{\delta a} + \left(\mathbf{E}_2 S/\sigma - H - \frac{HP}{s + p} - \frac{HQ}{s + q} + \frac{HP}{s - p} + \frac{HQ}{s - q}\right) \cdot e^{\delta(X-a)} + \left(\frac{-AP}{\alpha - p} + \frac{Bp \cdot e^{\alpha X}}{\alpha + p} - \frac{CP}{\beta - p} + \frac{DP \cdot e^{\beta X}}{\beta + p} - \frac{EP}{r - p} + \frac{FP \cdot e^{\gamma X}}{r + p} - \frac{GP}{s - p} + \frac{HP \cdot e^{\delta X}}{s + p}\right) \cdot e^{\beta a} + \left(\frac{AP \cdot e^{\alpha X}}{\alpha + p} - \frac{BP}{\alpha - p} + \frac{CP \cdot e^{\beta X}}{\beta + p} - \frac{DP}{\beta - p} + \frac{EP \cdot e^{\gamma X}}{r + p} - \frac{FP}{r - p} + \frac{GP \cdot e^{\delta X}}{s + p} - \frac{HP}{s - p}\right) \cdot e^{\beta(X-a)} + \left(\frac{-AQ}{\alpha - a} + \frac{BQ \cdot e^{\alpha X}}{\alpha + q} - \frac{CQ}{\beta - q} + \frac{DQ \cdot e^{\beta X}}{\beta + q} - \frac{EQ}{r - q} + \frac{FQ \cdot e^{\gamma X}}{r + q} - \frac{GQ}{s - q} + \frac{HQ \cdot e^{\delta X}}{s + q}\right) \cdot e^{qa} + \left(\frac{AQ \cdot e^{\alpha X}}{\alpha + q} - \frac{BQ}{\alpha - q} + \frac{CQ \cdot e^{\beta X}}{\beta + q} - \frac{DQ}{\beta - q} + \frac{EQ \cdot e^{\gamma X}}{r + q} - \frac{FQ}{r - q} + \frac{GQ \cdot e^{\delta X}}{s + q} - \frac{HQ}{s - q}\right) \cdot e^{q(X-a)} \quad [22]$$

The values of α and β are determined by equating the coefficients of $e^{\alpha a}$ and $e^{\beta a}$ to zero:

$$1 - \frac{P}{\alpha - p} - \frac{Q}{\alpha - q} + \frac{P}{\alpha + p} + \frac{Q}{\alpha + q} = 0$$

$$1 - \frac{P}{\beta - p} - \frac{Q}{\beta - q} + \frac{P}{\beta + p} + \frac{Q}{\beta + q} = 0$$

Adding and clearing of fractions:

$$(\alpha^2 - p^2) \cdot (\alpha^2 - q^2) - 2P \cdot p \cdot (\alpha^2 - q^2) - 2Q \cdot q \cdot (\alpha^2 - p^2) = 0$$

This expression is solved for each case by substituting the proper values of P, Q, p, and q, and solving the quadratic. For the round opening of $X = D$, $P = 1.282$, $Q = -0.282$, $p = -2.405$, $q = -6.345$, and the resulting values are $\alpha = -0.3563$ and $\beta = -6.58$.

Next, the coefficients of $e^{\gamma a}$ and $e^{\gamma(X-a)}$ in Equation [22] are equated to zero to find E and F; coefficients of $e^{\delta a}$ and $e^{\delta(X-a)}$ are equated to zero to find G and H. Finally, the coefficients of $e^{\beta a}$, $e^{\beta(X-a)}$, e^{qa}, and $e^{q(X-a)}$ in Equation [22] are equated to zero, using the values of α, β, E, F, G, and H as found above; this results in four simultaneous equations, which are then solved for the last four unknowns A, B, C, and D. It was found that each two successive coefficients, such as A and B or G and H, are identical except that $\mathbf{E}_1$ and $\mathbf{E}_2$ are interchanged.

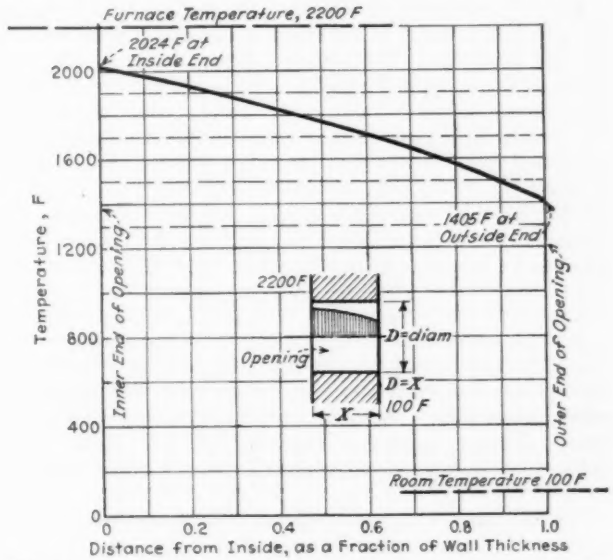


FIG. 6 TEMPERATURE VARIATION ALONG THE SIDE OF ROUND OPENING FROM INNER TO OUTER END

(For diameter = thickness of wall; 2200 F furnace temperature, 100 F outside temperature.)

For the round opening of $X = D$, the final solution is:

$$E_a = (1.261\mathbf{E}_1 - 0.6500\mathbf{E}_2) \cdot e^{-0.3563a} + (1.261\mathbf{E}_2 - 0.6500\mathbf{E}_1) \cdot e^{-0.3563(X-a)} + (0.0587\mathbf{E}_1 - 0.0053\mathbf{E}_2) \cdot e^{-6.58a} + (0.0587\mathbf{E}_2 - 0.0053\mathbf{E}_1) \cdot e^{-6.58(X-a)} - 0.05623\mathbf{E}_1 \cdot e^{-2.28a} - 0.05623\mathbf{E}_2 \cdot e^{-2.28(X-a)} - 0.0551\mathbf{E}_1 \cdot e^{-6.80a} - 0.0551\mathbf{E}_2 \cdot e^{-6.80(X-a)}$$

As previously stated, the equations can be simplified somewhat by taking $\mathbf{E}_1$ as unity and $\mathbf{E}_2$ as zero; then the above expression reduces to six terms.

On calculating and plotting numerical values for E_a, the graph was found to be an almost exactly straight line through $E_a = 1/2$ at $a = 1/2$. In agreement with the results obtained by the previously described method of dividing the sidewalls into a number of equal sections, however, the line was not absolutely straight, but had a very slight curvature. When the value of X (ratio of distance between end planes to diameter of ends) exceeded 5, the curvature became noticeable and was not quite symmetrical, indicating that for such high values of X, more than two exponential terms should be used for the $f(-)$ and $\phi(-)$ functions.

The straight-line relationships between E_a and a can be represented by Equation [9] given in the discussion of the first method of solution. If the actual emissive powers $\mathbf{E}_1$ and $\mathbf{E}_2$ of the two ends of the system are retained instead of using values of unity and zero to simplify the calculations, Equation [9] would be of the form:

$$E_a = \frac{\mathbf{E}_1 + \mathbf{E}_2}{2} + \Delta(\mathbf{E}_1 - \mathbf{E}_2) \cdot \left(\frac{1}{2} - \frac{a}{X}\right) \quad [23]$$

Δ being, as before, the fractional change in emissive power along the wall, i.e., now $(E_{a=0} - E_{a=X})/(\mathbf{E}_1 - \mathbf{E}_2)$. The approximate method of solution has already yielded Equation [11] for predicting Δ. The temperature of the wall at a is, of course, $\sqrt[4]{E_a/\sigma}$; and in Fig. 6 the emissive power has been converted to temperature for a special case, to show how the temperature along the walls may be expected to vary. Fig. 7 also shows the variation of T_a^4, for various shapes.

Given the variation of E_a with a, the method of determining the reradiation factor is the same as outlined in the preceding method of solution. When the reradiation factor is calculated by using the solution of Equation [22] for E_a and exact expressions for $\phi(a)$ given in Table 1, and then by using the two-term exponential expressions given in Equation [20], the agreement is close, and in agreement with the values obtained by using the first method of solution, Equations [12] and [14]. In Fig. 8 are plotted the reradiation factors for the various shapes. By adding to these factors the direct radiation factors (section 3 and Fig. 5), the total-radiation factors of Fig. 2 were obtained.

7 Modifying Factors

In the mathematical work it has been necessary to assume that the sides of the opening are non-conducting and have black-body absorptivity and emissivity (i.e., reflect none of the radiation which strikes them). Actually, neither of these two conditions is exactly fulfilled. Heat conduction through the material of the sides of the opening would tend to increase the reradiation by

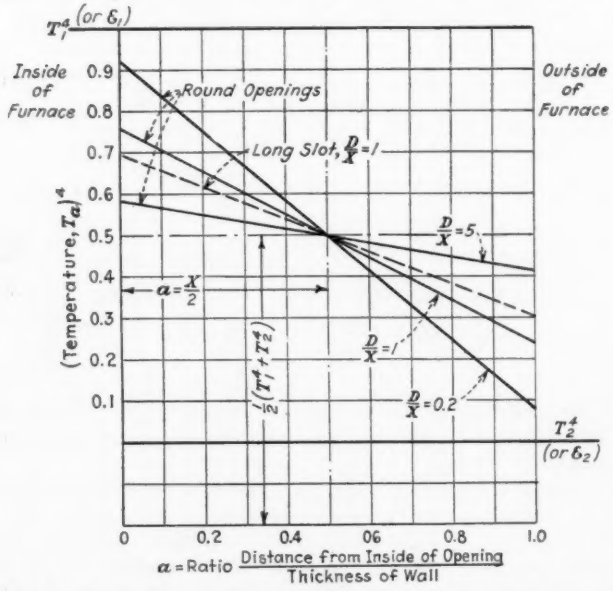


Fig. 7 Variation of Emissive Power or Fourth Power of Absolute Temperature of Sides of Opening, From Inner to Outer End

transferring heat from the inner to the outer part of the sides, whence it can be radiated more freely. In openings in walls of refractory material such as firebrick, conduction is so small compared to radiation that the error from this source is negligible in industrial furnaces.

The effect of partial reflection of heat rays from the sides is to increase the total amount of heat radiated through a given opening. As an extreme case, if the walls were perfectly reflecting (or absorbed none of the radiation), the total-radiation factor would be increased to unity for all openings,[6] and the opening would be equivalent to a freely exposed diaphragm. For rough surfaces such as firebrick, however, reflection plays only a minor part. Data on emissivity of brick surfaces are scarce and uncertain, but the value apparently lies between 80 per cent and 92 per cent of that of the black body. The proportion of heat reflected is therefore small, and since the reflection from such surfaces is almost all diffuse rather than specular, most of the rays strike the walls several times before escaping, and undergo 80

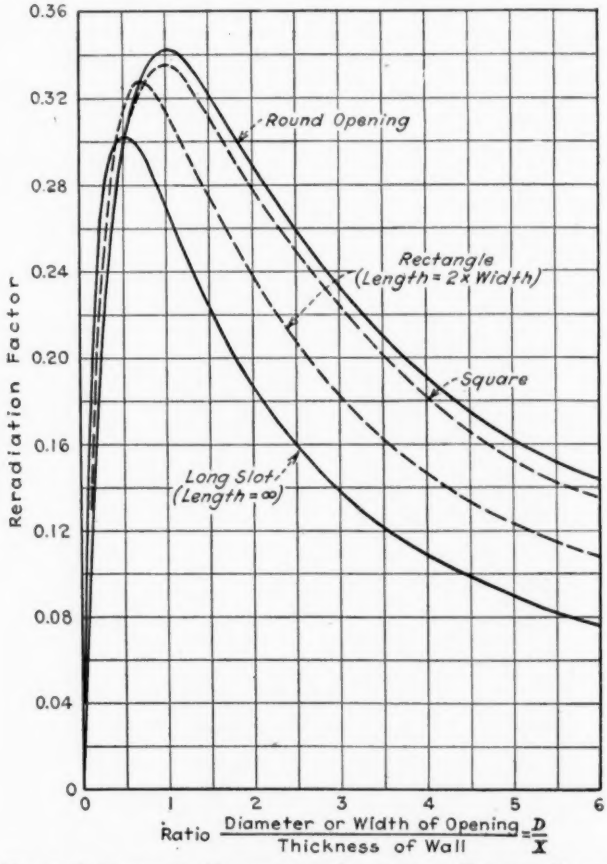


Fig. 8 Reradiation Factors for Openings of Uniform Cross-Section

to 92 per cent absorption each time. That the modification required in the reradiation factor due to reflection is negligible, is confirmed by the tests described in the following section, in which the heat absorbed by the calorimeter checked almost exactly with the sum of direct radiation plus reradiation as calculated from the measured temperature of the sides, neglecting reflection.

The factor which is actually of greatest importance in modifying the mathematically derived results is the transfer of heat to those parts of the sides near the outer end of the opening, by circulation of gases within the opening. While the effect of this action might be estimated mathematically, so many assumptions would have to be made that it seems best to depend on factors derived from tests as explained in the following section.

An assumption implicit in the basic Equation [3] is the validity of Lambert's cosine principle. Actually, the intensity of radiation from all surfaces in directions at large angles from the normal is different from that required by the cosine principle. The few data that are as yet available as to the deviation, however, indicate that the overall error introduced by assuming the correctness of the cosine principle is not great.

The effect of variation of temperature around the perimeter of a

[6] The additional condition must be imposed that the reflection be mirror-like, not diffuse.

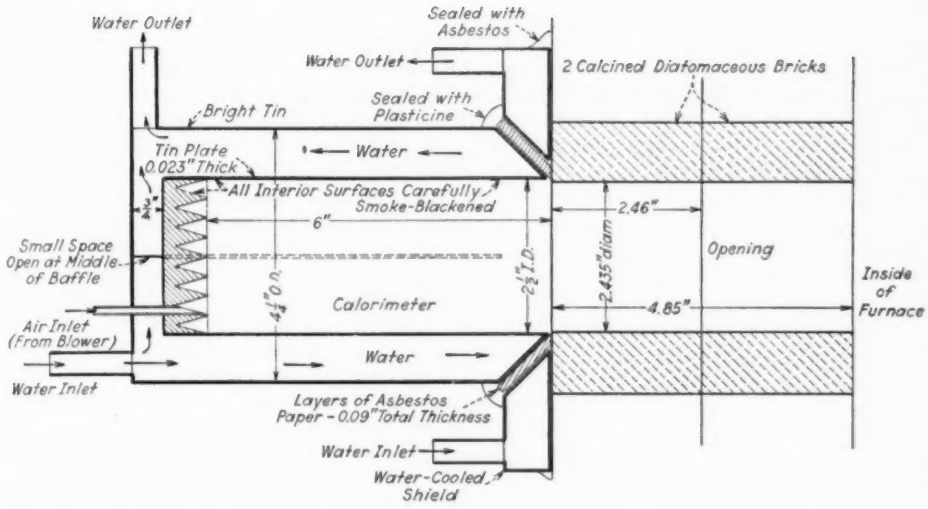


FIG. 9 CALORIMETER FOR MEASURING HEAT RADIATED THROUGH OPENING

square or rectangular opening has already been mentioned, but will be further discussed in section 8.

The factors of Fig. 2 will require modification if steady-state temperature conditions do not exist in the material of the sides of the opening. For instance, if a furnace door is opened only occasionally and remains closed during the greater part of the time, the sides of the door opening attain practically the temperature of the furnace interior at all points (except for the effect of conduction through the wall); whereas the outer door frame in which the door slides is not much above the temperature of the atmosphere. When the door is first opened, therefore, the rate of heat loss by radiation is the same as for a fully exposed diaphragm at the outside of the opening, and the total-radiation factor is unity. The sides of the opening at once begin to cool, however, and the factor decreases, becoming asymptotic to the values shown in Fig. 2 as the sides approach steady temperature conditions. How quickly the change occurs depends on the diffusivity of the material at the sides of the opening.

8 EXPERIMENTAL

In order to check the theoretical results and to determine the effect of the influences which had to be neglected in their development, tests were made of radiation through actual openings in furnace walls. The test arrangement is shown in Fig. 9. The heat radiated through the round openings was absorbed by the water calorimeter, the interior of which was carefully smoke-blackened. Temperature rise of the water was determined by calibrated thermometers. Quantity of water was determined by weighing over periods of 10 min, during which the temperatures remained practically constant. The temperature of the gas-fired furnace was determined by a platinum–platinum-rhodium thermocouple and by a calibrated optical pyrometer. A small amount of air was blown into the interior of the calorimeter through the tube shown, in order to keep it clear of burned gases that would absorb part of the radiation and would also cause condensation on the water-cooled surfaces.

The bricks, in the case of the round openings, were calcined diatomaceous bricks, held together by bolts. The hole was bored out in a lathe to a diameter of 2.435 in., care being taken to keep the edges of the opening sharp. The total thickness of the two bricks was 4.85 in., hence the ratio D/X was 0.502 (practically 1 to 2).

Temperatures at marked intervals along the side of the opening were determined by a contact thermocouple, consisting of chromel-alumel wires, butt-welded and the junction filed down to form a thin strip 0.09 in. wide (at the junction) by 0.018 in. thick (Fig. 10). During the time these measurements were being taken the calorimeter had to be removed, but the opening was closed by a metal plate having a slot for the thermocouple.

TABLE 4

	Test No. 1	Test No. 3	Test No. 4
Ratio = $\frac{\text{diameter of opening}}{\text{wall thickness}}$	0.502	0.502	0.502
Average furnace temperature, F	2180	2198	2262
Water flow, lb per hr	29.84	53.76	53.40
Temperature rise of water, F	34.15	19.44	21.52
Heat to water, Btu per hr	1019	1045	1150
Correction for heat from air, Btu per hr	—11	—11	—11
Heat radiated through opening, Btu per hr	1008	1034	1139
Heat radiated from freely exposed diaphragm, Btu per hr	2704	2773	3054
Ratio = radiation factor	0.373	0.373	0.373
Radiation factor computed from wall temperature	...	...	0.370
Theoretical radiation factor (from Fig. 2)	0.356	0.356	0.356
Average ratio = $\frac{\text{factor as measured}}{\text{theoretical factor}}$	...	...	104.2

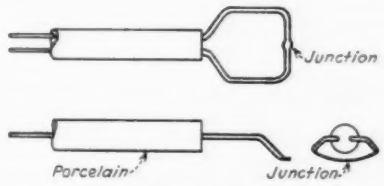


FIG. 10 CONTACT THERMOCOUPLE

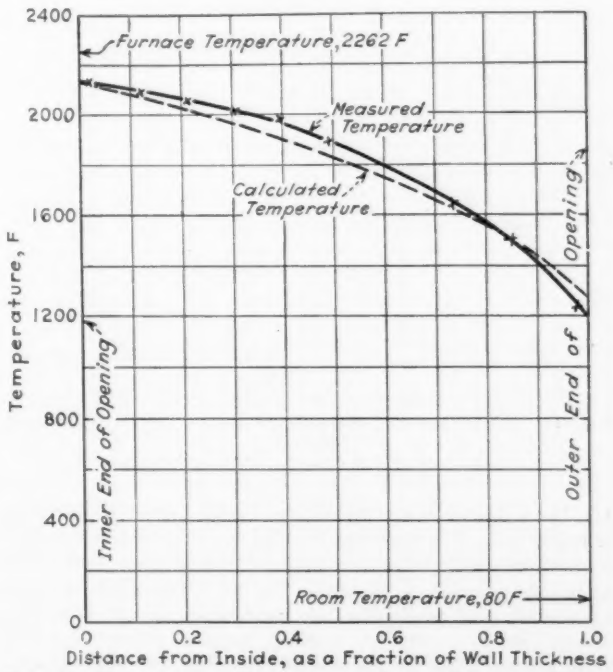


FIG. 11 TEMPERATURE DISTRIBUTION ALONG SIDES OF ROUND OPENING

(Diameter equal to 50.2 per cent of the wall thickness.)

The results of these tests on the round opening are shown in Table 4 and the temperatures in Fig. 11.

The remarkably close check between the quantity of heat as measured by the calorimeter and the amount calculated as direct radiation plus reradiation from the walls on the basis of the measured temperature distribution along the sides of the opening indicates that neither reflection nor variation from Lambert's law can be responsible to any large extent for the deviation from the theoretical figures.

The form of the temperature-distribution curves (solid line in Fig. 11) is very similar to the theoretical curve (broken line in Fig.

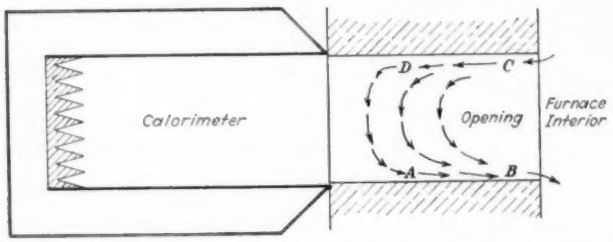


FIG. 12 CONVECTION OF HEAT BY CIRCULATION OF GASES WITHIN OPENING

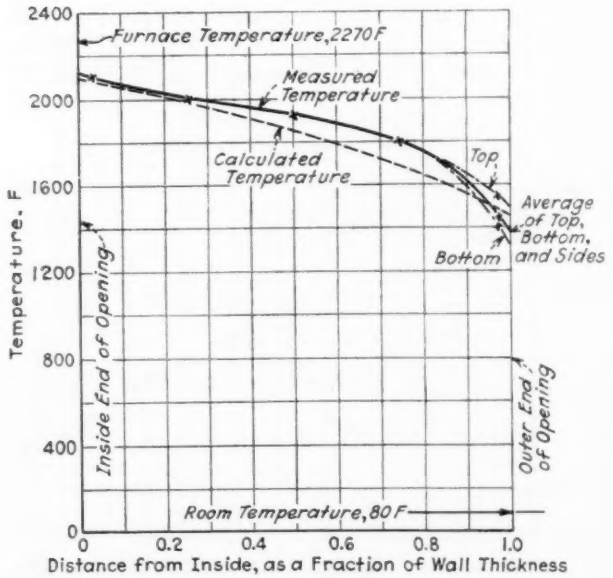


FIG. 13 TEMPERATURE DISTRIBUTION ALONG SIDES OF ROUND OPENING

(Diameter equal to 98.8 per cent of the wall thickness.)

11), but is higher in the middle and slightly lower at the outer or colder end. The explanation of this at once becomes evident from consideration of the circulation of gases which must occur within the opening, as shown in Fig. 12. The colder, heavier gases at A tend to displace the hotter gases at B; they flow toward the inside end of the opening, and are replaced by hotter gases floating along the top toward the outer end. The circulation carries heat from C to D and A. It may tend to cool points near B, but since this is the high-temperature end, radiation from the inside is so active that it prevents the temperature at C from dropping. The net result is a transfer of heat by convection toward the outer end of the opening and an increase of temperature at D and A. At the outer end, the top is higher than the bottom, as indicated by the dot-dash curves in Fig. 13.

As would be expected, the effect of convection is greater in an opening which is short in proportion to its diameter. This was shown by further tests in which the inner brick was removed, the single brick thickness of 2.46 in. then giving a ratio D/X equal to 0.988 (practically 1 to 1). The temperature distribution along the sides of the opening, as determined with the contact thermocouple, is shown in Fig. 13. The transfer of heat by convection to the outer end is even more clearly shown by the measurements in this case. In fact, with this comparatively wide opening in a thin wall, convection reached even into the calorimeter, as was evidenced not only by the difference of temperatures at the outer end of the opening, but especially by rapid condensation on the inner surfaces of the calorimeter, in spite of the blast of air blown through the tube. With the thought that the air inlet should have been at the top instead of at the bottom, the calorimeter was inverted; condensation still occurred as before, however, and the heat absorption was practically the same.

On account of this direct heat transfer from the furnace interior to the calorimeter by convection of gases, the heat absorption by the calorimeter was excessive, and the results were useless for determining radiation in this particular case. Since the tests with the first opening had shown, however, that the radiation calculated as the sum of direct radiation plus reradiation from the sides having the measured temperature distribution agrees almost exactly with the heat absorption of the calorimeter so long as there is no convective heat transfer to the latter, a similar calculation was carried out for the second opening. It resulted in a total-radiation factor of 0.551, as compared with the value 0.515 read from Fig. 2. This indicated that convection or circulation of gases within the opening increased the total radiation of heat by 7 per cent in this second opening (D/X = approximately 1 to 1).

A rectangular opening was next tested, to determine whether the assumption made in sections 5 and 6 that the variation of temperature around the perimeter of the opening in planes parallel to the end planes could be neglected led to appreciable errors. This opening was 2.64 in. high by 4.50 in. wide in a firebrick wall 4.50 in. thick. Hence the ratio of sides was 1.71 and the ratio D/X was 0.587. Calorimeter measurements were not made, but temperatures were measured with the contact thermocouple in many different places on the top, bottom, and sides of the opening, the latter being closed at the outside by a metal plate which was kept cool. The results are shown in Fig. 14, many of the points of which represent several coinciding readings. At the inner edge, the temperatures never varied more than 10 F, but toward the outer end, where temperatures were lower and interradiation less active, the differences were quite large. These differences, again, were doubtless accentuated by convection or circulation of gases, which is well known to be more active near the middle of a rectangular passage than in the corners.

The measurements having been plotted in Fig. 14, the tedious task was next undertaken of calculating $\phi(a)$, the "angle factor" of radiation from the walls to the outer end of the opening, for each of the points at which measurements were taken. The products $(T_a^4 - T_2^4)\cdot\phi(a)$ were then plotted for top, bottom, and sides, and the average over the whole area of the walls of the opening was determined; from it, the reradiation to the outside was found. Adding to this the direct radiation (see Fig. 5) the total-radiation factor was found to be 0.493 for this opening. The corresponding value from Fig. 2 is 0.45; hence the actual radiation is $9^1/_2$ per cent more than the "theoretical" value.

As a further check, values of $(T_a^4 - T_2^4)$ corresponding to the temperatures of Fig. 14 were averaged (with regard to linear distance around perimeter) for each plane at a/X = 0.1, 0.2, etc. to 1.0, and each average was multiplied by the average value of $\phi(a)$ for the whole perimeter, intermediate between those values given by the expressions in Table 1 for n = 1 and 2. The total-radiation factor found by this method, which more nearly

corresponds to the assumptions for the rectangular opening in sections 5 and 6, is 0.476, or 5½ per cent higher than the "theoretical" figure. Roughly, then, it may be said that about half of the increase is due to the neglected variation of temperature

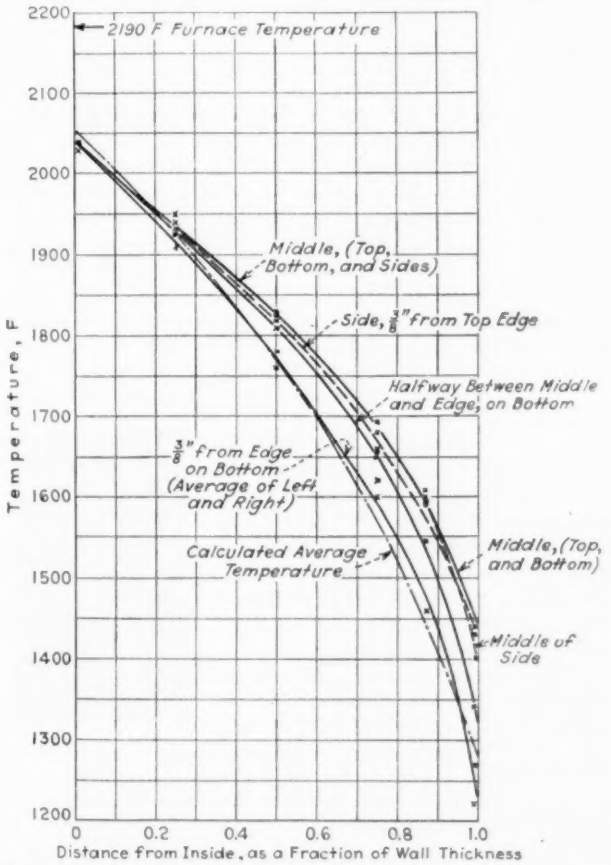


FIG. 14 TEMPERATURE DISTRIBUTION ALONG TOP, BOTTOM, AND SIDES OF RECTANGULAR OPENING

(Opening of 2.64 in. height, 4.50 in. thick, in wall 4.50 in. thick.)

around the perimeter, while the remaining half is caused by convection (and possibly, to a slight extent, by conduction through the fireclay). The latter figure checks well with the 4.2 per cent increase by convection in the first of the round openings, which had a ratio of width to wall thickness not very far different from that of the rectangular opening.

Where the axis of the opening is vertical instead of horizontal, as in the case of a furnace chamber with radiating surface under the roof and absorbing surface on the floor, or vice versa, it is probable that these convection factors would no longer apply; they would doubtless be decreased when the hotter surface is at the top and increased when it is at the bottom.

In the tests, any actual outward flow of gases was prevented by the calorimeter or by the cooled plate. In furnace openings where such flow does occur, the temperature of the sides will be raised still more. The additional heat loss cannot, however, be charged to radiation; in fact, radiation from the interior is actually reduced thereby, on account of the higher wall temperatures.

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Mr. Hottel wishes to acknowledge the assistance of Mr. D. M. Gordon in the numerical calculations of section 5.

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Discussion

ROBERT B. SOSMAN.[7] One accidental "modifying factor" that needs to be looked out for in studying the radiation through an opening in a furnace where a water-cooled jacket is being used is the conversion of an absorbent surface into a reflecting surface by the condensation of water on the wall. We have observed this effect in experimenting with the radiation from an open-hearth furnace. In a tube containing no diaphragms, the condensation of a large number of small water drops on the inside of the water-cooled tube causes a surprisingly large increase in the radiation received, because the water drops act almost like specular reflectors, although the dry wall is of dull black metal and is a good absorber of radiation.

It might be worth while to call attention to the influence of wave length on reflection, especially at large angles of incidence. (This is discussed, for example, in R. W. Wood's "Physical Optics," New York, 1905, pp. 36–39.) Radiation that is incident nearly parallel to a surface is almost completely reflected if the wave length is comparable in size with the granules of the surface. This effect will be the more important the lower the temperature that is being measured, because the wave length of maximum intensity from a full radiator increases with falling temperature. Even the rough surface produced by cutting a hole in a fireclay brick will be a perfect reflecting surface for rays of the far infrared that strike the sides at angles of only a few degrees.

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The Rolling and Extrusion of Aluminum and Its Alloys

By ROBERT L. STREETER,[1] PITTSBURGH, PA.

This paper outlines the history of the development of the manufacture of aluminum sheet, wire, and rod. It then discusses the growth of the demand for aluminum alloys and the problems that research laboratories and engineers had to solve before the manufacture of these alloys became practicable. A description is given of mills for rolling standard structural shapes, with reports on tests made to ascertain the power required to roll strong alloys. The extrusion of aluminum alloys is discussed, touching on the reasons for using extrusion in place of rolling, and explaining the limitations of the extrusion method of manufacturing shapes.



UNTIL 1914 the outstanding properties of aluminum were lightness and resistance to corrosion. It did not boast of strength and was not used where structural strength was required. Cooking utensils and automobile bodies were the principal outlets for sheet and electrical conductors for wire and rod. However, these conditions were changed, first, by the demands of the war in Europe and, second, by the possibilities opened up by the discovery by Wilm in 1912 that changed the whole future of the manufacture and application of aluminum. This discovery showed that certain aluminum alloys could be greatly improved by heat treating. The demands of the World War gave great impetus to research work based on Wilm's discovery, and the result was that group of high-strength aluminum alloys of which duralumin is typical. During the war and for ten years after, the output of duralumin was comparatively small and was absorbed almost entirely by the aircraft industry. But even though the quantities manufactured during that decade were small and the output took the form of small thin sheets demanded by the manufacturers of aircraft, considerable trouble was encountered in making the sheet. Research, however, came to the assistance of the mill man, and many of the difficulties were solved—not, however, without the expenditure of considerable sums for new equipment. The double-duo and continuous mills in use at that time were found to be quite inadequate for rolling duralumin sheet. For this purpose they were found to be deficient in power even though equipped with 800-hp or 1200-hp motors. Gears were too light, spindles and couplings too small, and housings too "springy" for rolling the new alloy.

About 1925 engineers in other lines than aircraft began to realize that if duralumin could be used to build a strong, light airship, it might be used to advantage to construct other strong and light machines. Great interest was shown by designers of trolley cars, railway cars, buses, trucks, traveling cranes, steamships, and many other structures moved by power. All of this interest resulted in a definite demand for large sheets and heavy plates, large rods for making rivets, large tubing of various sections, and structural shapes of a size never dreamed of when the first duralumin went into aircraft. As an example of the trend of these demands, designers of railway equipment insisted on channels and beams up to 14 in. and 85 ft long, heavy angles 6 by 6 in., 4-in. Z-bars, and rounds up to 8 in. in diameter, to say nothing of plates as thick as 1 in., with widths up to 132 in. and lengths to 90 ft.

Plans to meet these demands were under consideration by the Aluminum Company of America for several years before mills to take care of them were finally erected at Massena, N. Y., and Alcoa, Tenn., in 1928 and 1930. (See Figs. 1 and 2 for the blooming mill and the structural mill, respectively, at Massena.) It was no simple matter to build such mills and put them into operation. Experience had shown that what was good for steel was not necessarily good for aluminum. Much experimental and research work was demanded before design could be started. Rolling ingots weighing 3000 lb were needed. The heaviest used up to that time was about 250 lb. A heat-treating furnace 90 ft long was required, five times as long as the largest then in use. Soaking-pit temperatures must be held within extremely close limits. In order to secure best physical properties, quenching after heat treating must be done in a very few seconds. These are just a few of the problems that confronted the designers. How they were solved is shown in the description of the blooming and structural mills which follows.

The demands for the size of shapes to be rolled could be met by an ingot 20 by 20 by 72 in., and these were the dimensions used in the design. The melting and the casting of the ingots are accomplished along much the same lines used in steel open-hearth practice. Open-hearth furnaces with a capacity of 10 tons each are used, but without air preheating. Various fuels are used throughout the world for melting aluminum—coal, coke, oil, gas, wood, or electricity. In the plant under discussion the decision was to use coke.

The metal arrives at the furnaces in pigs that are similar to those used for iron in form and size, but they weigh only 50 lb. These pigs are melted down with the necessary alloying materials, which are usually rich alloys with aluminum. Magnesium, however, is added in the form of pure metal. Casting of the rolling ingots is done in heavy iron molds similar to those used for

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

steel, and stripping is accomplished by means of a press instead of the standard stripping crane.

Electricity was available in large quantities and at reasonable cost; hence the decision to use electric soaking pits. As far as general arrangement is concerned, these pits follow steel-mill practice quite closely. Each pit accommodates five 20 by 20-in. ingots on end. The first installation consisted of 14 pits in one row, and the battery requires 2660 kw for heating. (See Fig. 3.) Temperature is maintained by automatic control, and recording pyrometers give continuous records of temperatures. All these instruments and the control desk are installed in a compact group in a central control room. One operator has complete control of the pits and issues the necessary instructions to the operating crew.

The control desk is worthy of special mention. It is made up of 14 sections, one for each pit, each section consisting of the following: (1) At the bottom of the front edge is a nameplate showing the number of the pit. (2) Just above the nameplate is the handle of the special Westinghouse type-W selector switch. (3) Immediately above the switch handle are three red lamps, one for each phase of the heating element in the pit. Each lamp is lighted only when current is actually flowing through the heating element in its respective phase, as determined by the automatic temperature control and the fact that none of the circuits has been accidentally opened. (4) The next five lamps are of different colors: clear, opal, blue, red, and green. These are dark or lighted according to the position of the selector switch. (5) At the top of the board is a graduated dial divided into 24 parts numbered for the 24 hours of the day. These dials are used in conjunction with pointers set at different time intervals to be determined after the proper cycle of operation has been learned by experience. The dial and pointers are to enable the operator to determine at a glance the proper operating program for each pit.

In addition to the signal lamps on the control desk, there are four lamps for each pit: red, green, amber, and opal. These are mounted on the roof of the control room where they can be readily seen by the mill-crane operator and the pit-cover-crane operator. These lamps constitute the mill signal board and burn in conjunction with the lamps of the same color on the control desk and serve to issue orders to the crane operators. It is apparent that there are more signal lamps on the control desk than on the mill board; the extra lamps on the desk are for condition signals only, while those common to both the control desk and the mill board are action signals and signify that some particular operation is required of the crane operator.

In order to increase the possible number of signals with the minimum number of lamps and to make some signals more noticeable than others, provision is made to energize the lamps from a steady or from a flashing source of current. All action signals are indicated by a flashing light. Steadily burning lamps indicate only an existing condition in the cycle of operations, requiring no action on the part of the crane operators. For instance, if the switch is on the first position, the green lamps burn steadily, indicating that the pit is empty but hot. When the switch is turned to position 2, the power is automatically cut off from the pit and the green light flashes, signifying that the pit is

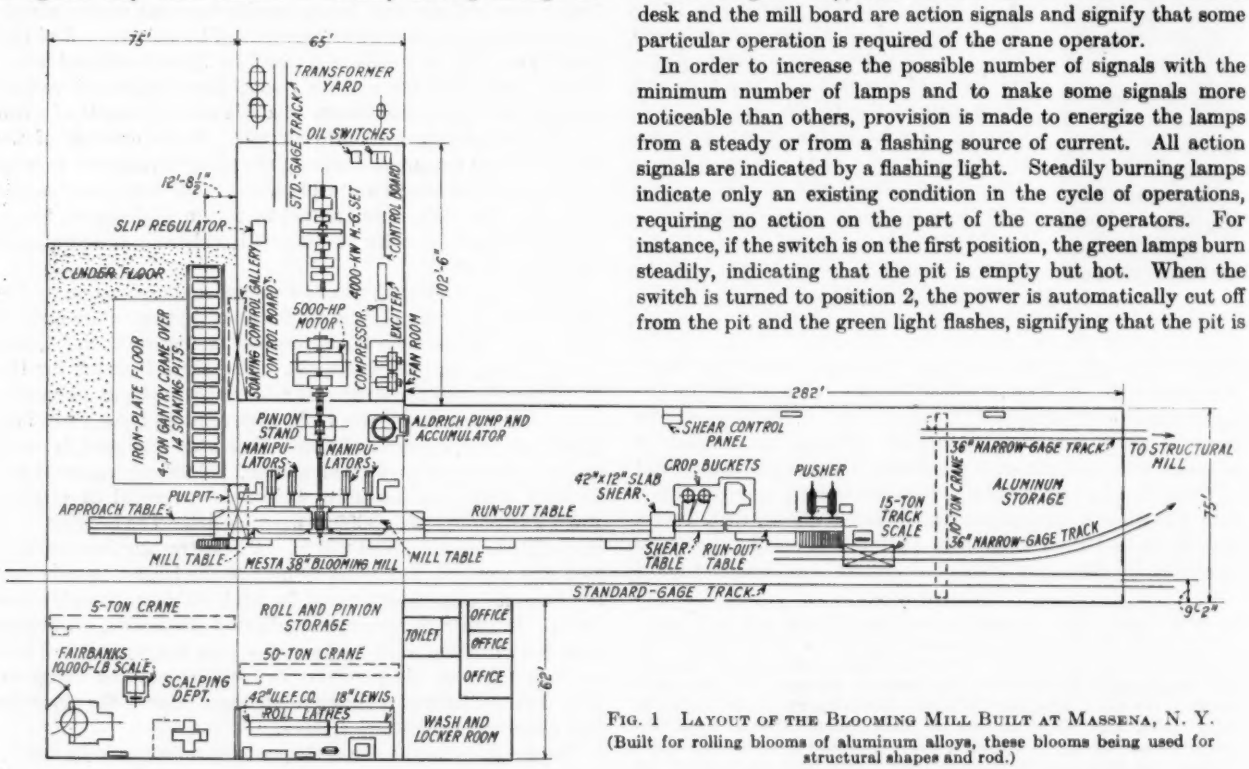


FIG. 1 LAYOUT OF THE BLOOMING MILL BUILT AT MASSENA, N. Y.
(Built for rolling blooms of aluminum alloys, these blooms being used for structural shapes and rod.)

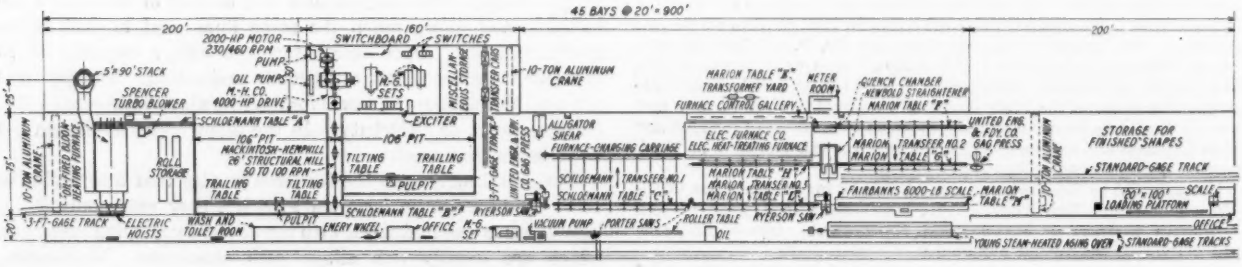


FIG. 2 LAYOUT OF THE STRUCTURAL MILL AT MASSENA, N. Y.
(Built for rolling aluminum alloy shapes up to 14 in. and in lengths up to 90 ft.)

to be charged. When the switch is on position 7, the power is on and the red lights are burning steadily, indicating that the pit is ready for discharge; but when the switch is put on position 8, the power is cut off and the red light flashes and the operators know that the pit should be unloaded.

The pit covers are handled by a special gantry-type crane which lifts them and carries them either forward or backward, giving a full pit opening for the operation of the pit crane. This latter crane is of the ordinary traveling type and utilizes automatic Gillert ingot tongs suspended from the crane hook.

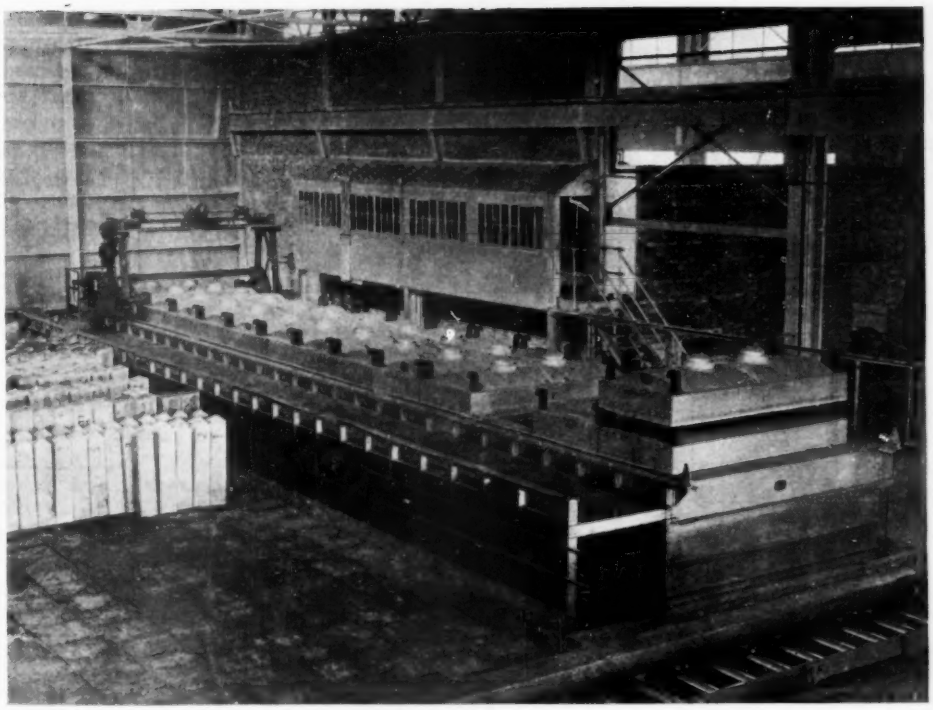


FIG. 3 FOURTEEN ELECTRIC SOAKING PITS
(Each with a capacity of five 3000-lb ingots)

BLOOMING MILL

The selection of the size and type of blooming mill depended on many factors that are not usually met with in rolling steel. Among the many things taken into consideration were: (1) Size of ingot necessary to produce largest section, (2) shaped blooms for large sections, (3) rectangular blooms for small sections, (4) slabs for sheet and plate mills, (5) blooms for flat stock, and (6) numerous alloys to be rolled, these varying greatly in chemical composition and physical properties.

Different rolling speeds for the various alloys had to be taken into consideration. Owing to the nature of the aluminum business, in which buying is done in comparatively small quantities and warehousing on a country-wide plan had not yet been developed, roll changes would be frequent, necessitating a type of mill well adapted to frequent and rapid roll changes.

On account of the 20 by 20-in. ingot that had been decided on, it was thought best to use a 38-in. mill. One of the two-high reversing type was fixed upon. Rolling of flat slabs which necessitated edging passes fixed the roll face at 84 in. and the maximum lift of the top roll at 36 in.

Speed and power were not easily decided on, as there were no data available on the speed at which aluminum could be rolled on a blooming mill and on how much power would be required at the various speeds and for different alloys. In order to obtain data on which preliminary calculations could be based, permission was obtained from the Bourne-Fuller Company, Cleveland, Ohio, to roll a few ingots on their blooming mill. These ingots were cast at the Cleveland Works of the United States Aluminum Company, preheated there in an electric furnace, packed in silocel powder, and trucked to the Bourne-Fuller Works.

The rolling of these ingots indicated that power requirements would vary greatly with the alloys rolled. It was apparent from these tests that 17S alloy with physical properties comparable to duralumin required more power than open-hearth steel and that pure aluminum required about 37 per cent as much power as 17S. Taking the meager data obtained into consideration, the speed of the mill was fixed at a maximum of 120 rpm, and a motor with a nominal rating of 5000 hp was decided on.

The mill installed was built by the Mesta Machine Company and was in all general respects such a mill as would be used for rolling steel. On account of the softness of hot aluminum, it was necessary to hand-finish and polish all surfaces coming into contact with the metal rolled. This is found necessary generally to prevent all scratching and gouging of aluminum as it is being rolled, as such marks do not heal or roll out, but appear as defects in the finished piece.

This inherent characteristic of the metal prohibits carelessness of any kind in the rolling, and it also tends to discourage obtaining rolling costs comparable with those obtained in rolling steel.

The motors selected for the blooming-mill auxiliaries are rugged in construction and of ample power. Two screwdown motors are used, each of 150 hp. The manipulator and table motors are all of 85 hp each, and all these motors are controlled from the operating pulpit and are equipped with full magnetic control. The switchboard for all the motors is in the main motor room.

The bloom shear is worthy of special mention as it illustrates very clearly the proportions to which the aluminum-alloy business has grown and shows the trend of future development. This shear, which is placed 138 ft back of the mill, has a knife 42 in. wide and a stroke of 12 in. The eccentric shaft is 28 in. in diameter. The capacity of the shear is 3,500,000 lb, and it will cut a hot duralumin bloom 10 by 36 in. The depressing table is balanced by a hydraulic cylinder and is on the same system as the top roll balance of the mill.

The main drive of the mill is by a motor built by Allis-Chalmers Manufacturing Company that operates at constant torque, 1,200,000 ft-lb, from 0 to 50 rpm, and at constant power from 50 to 120 rpm. It operates on a nominal voltage of 600. The current for the motor is furnished by two direct-current generators with a combined capacity of 6600 amp at 600 v. They are driven at 507 rpm by a 4000-hp, three-phase induction motor operating at 6600 v. This motor is started by means of a liquid slip regulator. The flywheel is of the built-up plate type with no hole in the center, as it is supported on two flanges. It weighs

80,000 lb, is 16 ft in diameter, and has a rim velocity of 25,500 fpm.

The lubrication of all the bearings of the motor and motor-generator set is by circulating oil, filtered and cooled. The bearings are equipped with thermostats which shut down the motor-generator set if the temperature rises to a dangerous degree. A drop in oil pressure will do the same thing.

Structural Mill

The structural mill consists of three 3-high stands 26 by 60 in. and one 2-high stand 26 by 48 in. On each side of the mill are one traveling tilting table and one trailer table. Provisions have been made to roll lengths up to 135 ft. Duplicate tables can be added when production warrants.

The mill is driven by a 2000-hp, direct-current, variable-speed motor operating at 230 to 460 rpm through a herringbone reduction gear with a ratio of 4.60 to 1. The rolling speeds are from 360 to 720 fpm. The gear housing is arranged so that an additional motor and pinion may be installed opposite the present one, thus doubling the driving power without dismantling the drive.

Current for the mill motor is furnished by a motor-generator set of 1600 kw capacity, the motor of which is of the synchronous type, operating at 6600 v. In this motor room there are two 750-kw motor-generator sets which furnish direct current for all the auxiliary motors in this and the blooming mill.

Blooms are handled from the blooming mill on narrow-gage cars and gasoline tractors with rubber tires. The blooms are heated in an oil-fired, chain-conveyor type of furnace operating at a temperature range of 700 to 950 F. At the discharge end of the furnace the unloading device lifts the bloom clear of the chains and deposits it on the mill-approach roller table. The rollers in this table are of the internal-motor type, operating at 25 cycles, and are not of variable speed. The run-out table from the finishing stand of the mill is equipped with rollers of the same type, but since it is desirable to have them operate at variable speeds to coincide with those of the mill, current for them is furnished by a motor-generator set, the motor of which is a variable-speed, direct-current motor, the speed of which controls the cycles of the current furnished by the alternating-current generator and hence the speed of the motor rollers. This table carries the bar to a saw and a gag press, where ends are cropped and straightening done, where necessary to have the bar in proper shape to enter the heat-treating furnace. After the cropping and straightening operation, the bar is carried by a transfer table to the heat-treating-furnace car.

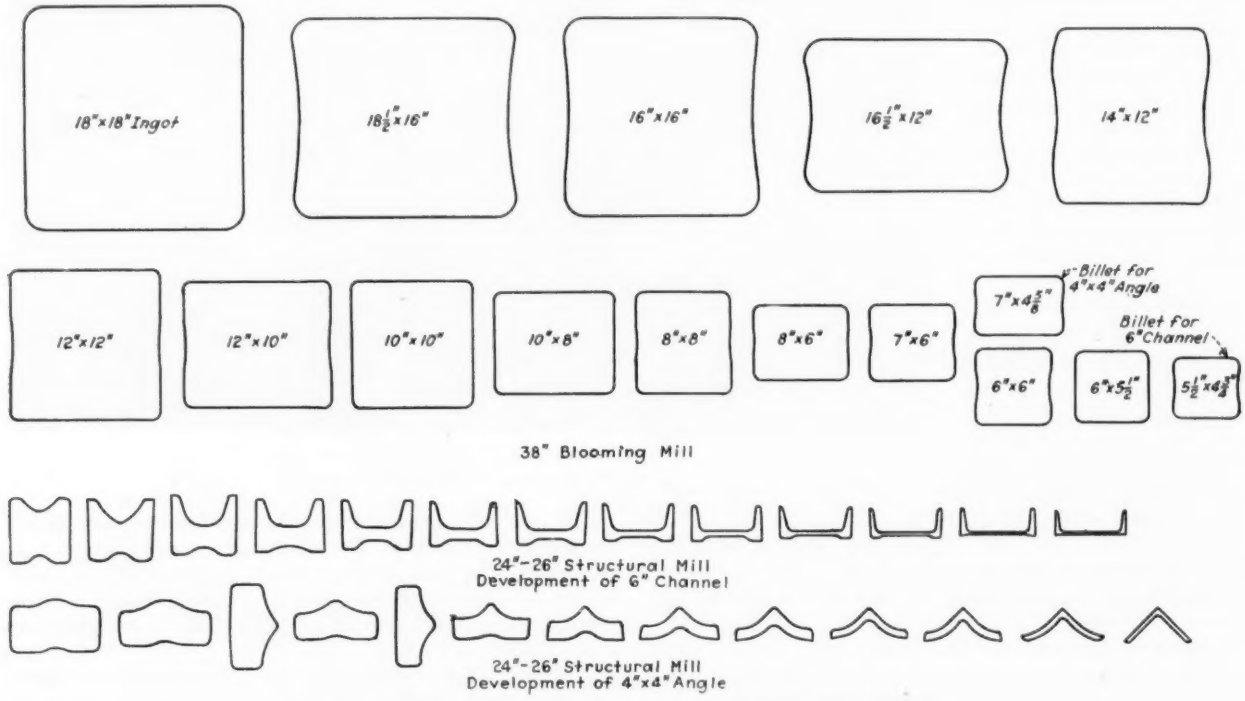


Fig. 4 Diagrams of Bloomer and Shape Mill Roll Passes
(These range from an 18 by 18-in. ingot, a 4 by 4-in. angle, and a 6-in. channel.)

Fig. 4 shows the bloomer and shape mill roll passes, from an 18 by 18-in. ingot, a 4 by 4-in. angle, and a 6-in. channel.

Heat Treating

In the heat treating of aluminum alloys there are two very important requirements: first, close control of furnace temperature, about 950 F, and second, quenching must take place in as short a time as possible when the alloy is withdrawn from the furnace. On account of the first requirement, it was decided to use an electric furnace, and the one constructed is the largest of its kind ever installed. It is 93 ft long and 20 ft wide. The heating elements are of the cast-grid, metallic-resistor type. The furnace is divided into 27 zones, and the temperature of each zone is separately controlled by an automatic temperature regulator. Pyrometric equipment is installed to indicate and record the temperature in each zone. A temperature differential of $\pm 7\frac{1}{2}$ deg F is maintained in the working area of the furnace, insuring uniformity of heating throughout the length of the piece being heated.

The heat-treating car is of a special type, having 16 pedestals on which the bar rests. This car enters the furnace through a door at the right of the furnace end. The shuffleboard arms in the furnace lift the bar from the pedestals, and the car is immediately withdrawn from the furnace. During the time in which the bar is carried across the furnace by the shuffleboard

arrangement, it is brought up to heat-treating temperature, and is finally discharged on a roller table, which carries it through the quenching chamber, shown in the lower right-hand corner of Fig. 5. Here quenching is accomplished by a series of high-pressure water sprays, as the bar comes out of the furnace. This system of quenching was adopted after a careful study of all available methods. While it is not as rapid as tank quenching, it is entirely satisfactory and much cheaper to install and is less complicated to operate than the tank method. As the bar leaves the quenching chamber, it is carried on a transfer table to the straightening press or roll straightener. After the bar is straightened, it is transferred once more to the run-out table, which carries it to the stock pile or the loading platform.

Certain aluminum alloys must be "aged" after heat treating to bring out their maximum physical properties. This aging is accomplished by soaking at temperatures ranging from 250 F to 340 F, and the time required is from 24 to 36 hr. In order to secure adequate production, this aging is done in large batches, as aging furnaces are simple in design and construction and comparatively inexpensive to build. In this mill the aging furnace is large enough to hold two standard-gage railway cars loaded with bars up to 90 ft in length. The furnace is heated by steam, and the air in the furnace is circulated by overhead fans, insuring rapid and uniform heating throughout.

Characteristics of Aluminum-Alloy Manufacture

There are many details of practice in the rolling of aluminum and its alloys that are not generally present in the practices with other metals. It might be of interest to mention a few of the more important points in which aluminum differs from other metals in regard to handling during the process of rolling shapes.

The most widely used strong aluminum alloys contain about 4 per cent copper. When the ingot freezes, a high-copper alloy, still molten, is forced out to the surface of the ingot by the pressure of the freezing crystals within. This process of liquation results in a network, which sometimes contains as high as 35 per cent copper, being distributed more or less generally over the surface of the ingot. This alloy is extremely hard, and while it is present in very small quantities compared to the weight of the ingot, it will cause considerable grief if not removed. Scalping all four sides of the ingot is resorted to, and this may be accomplished by any one of several machines, planers, boring mills, or milling machines, whichever machine seems to be best suited for the particular size and shape of the ingot that is being used.

The operation in the soaking pit is a little more involved than just a mere heating up to the rolling temperature. A metallurgical change must take place, and this change requires much longer time than bringing the ingot to temperature. In fact, sometimes it is desirable to soak the ingot for a considerable period above the best rolling temperature. In this case the ingot must be cooled carefully after soaking in order to bring it down to the rolling temperature and still have its temperature uniform throughout its mass. This complicated procedure cuts down pit capacity, necessitates expensive heat-control equipment, and increases production cost.

The hot ingot on the mill-approach table, with its four sides carefully machined, and found to be at the correct rolling temperature by the last pyrometer test, appears to be all ready for the job of rolling, which should be carried on quickly and without trouble. But the interior of the ingot is not visible. It is made up of large crystals, which must be broken down carefully by a few light passes on the mill. The metal is brittle in the cast state, and ingot cracking is likely to occur. There is also a marked tendency for the ingot to twist and curl after a few passes on the mill. Sticking to the rolls, which is very likely to occur with aluminum, is prevented by the use of a flooding system of oil solution. This solution must be filtered and cooled continuously, which necessitates a costly installation of tanks, filters, coolers, and pumps. Rolling must be done without marring the surface of the metal, which means that main rolls, manipulator bars, table rollers, guides, and aprons must be kept in perfect condition all the time.

Shearing the blooms and billets does not present any particular difficulties, but here again every precaution must be taken against marring the surface of the metal. It is much safer to pick up the bloom back of the shear than to push it off the table on slides or let it fall on the car that carries it out of the mill. And of course it cannot be picked up with a magnet. Very little is possible in the way of reconditioning bloom surfaces by chipping or filing. In some cases blooms that might otherwise have to be scrapped are scalped in order to reclaim them.

Heating blooms for the structural mill presents no particular difficulties if proper temperature control of the heating furnace is secured. The actual rolling of the shape is not attended with some of the difficulties met with on the blooming mill. The metal is not in the cast condition, and hence there is less danger of cracking, and high rolling speeds may be used if sufficient power is available. However, there is still a tendency for the metal to twist in the passes, and stickiness of the hot aluminum still gives trouble. Guides must be in perfect condition and well lubricated. Roll grooves must be smooth and also well greased. It is difficult to force aluminum up into roll grooves, and once there, it is equally difficult to get it out again.

Power Required to Roll Aluminum

On account of the comparatively short time that suitable equipment has been available, tests made to find the power required to roll aluminum alloys are still very incomplete. On account of the numerous alloys being used and the large variety

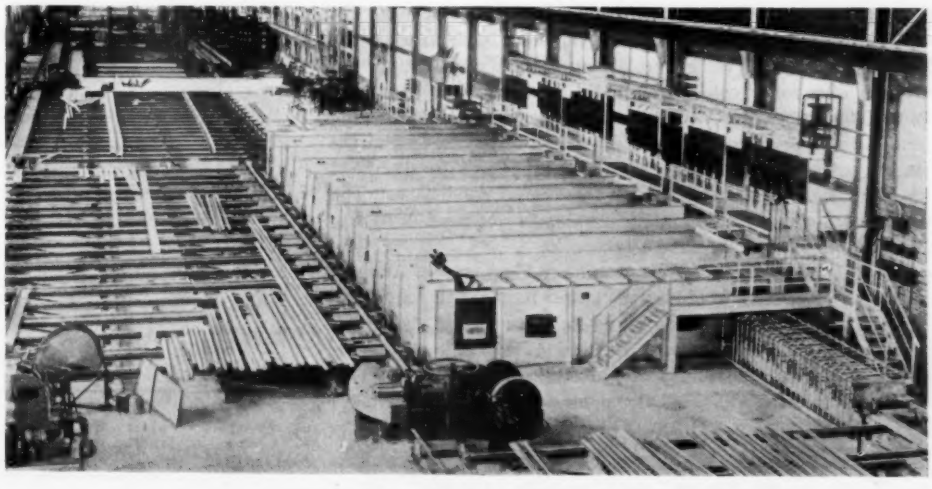


Fig. 5 Discharge End of Electric Heat-Treating Furnace
(This will handle shapes up to 90 ft in length.)

of sizes being rolled in small quantities, it will take several years to accumulate sufficient data to form the basis of any very definite conclusions regarding the power required for specific reductions in area and how the power is influenced by the many variables met with.

The reason that no tests were made before 1929 is that there were practically no mills used for rolling aluminum that were individually driven except some double-duo hot mills and small strip mills rolling foil. The former mills were not suitable for obtaining accurate power readings, and the foil mills were small, rolling only cold thin strip, so that the range of data possible to secure would be very limited.

During the rolling tests made at the Cleveland Works of the Bourne-Fuller Company, data were taken in order to ascertain, if possible, the actual power required to roll large alloy ingots and to find out the relative power required to roll aluminum and steel under the same conditions, each being rolled of course at its proper temperature. Unfortunately, only four aluminum-alloy ingots were available. Owing to the fact that the mill was set up for rolling steel and that the mill crew had never rolled aluminum, the results obtained were not conclusive. They did indicate, however, that high-strength aluminum alloys rolled at the correct

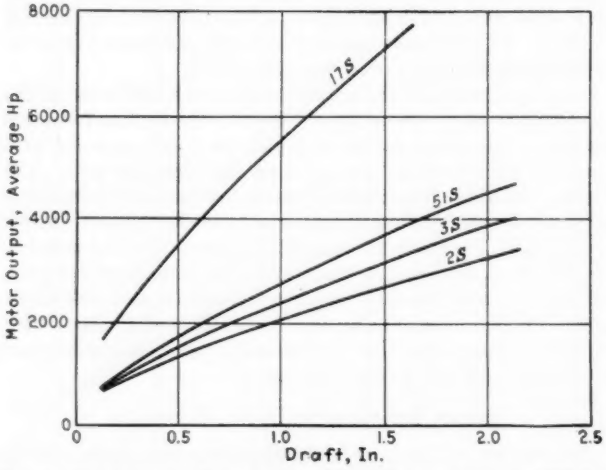


FIG. 6 HORSEPOWER REQUIREMENTS FOR REDUCTIONS ON 1500-LB PLATE INGOTS

temperature required about 50 per cent more power than open-hearth steel rolled at its proper temperature.

In 1929 the author made some rolling tests in Switzerland on three-high strip mills using pure aluminum and duralumin, both of which were rolled cold. The metal used was 12 in. wide and about 0.06 in. gage. Various reductions were tried, ranging from 25 to 70 per cent. The object of the tests was to find the relative power required to roll pure aluminum and duralumin at the same gage and under exactly the same conditions. The results obtained indicated that duralumin required 2.70 hp, where the pure metal required 1 hp.

During 1931 engineers of the Aluminum Company of America made a series of tests to obtain the power required to roll aluminum alloy slabs and plates on an 84-in. plate mill at the Alcoa, Tenn., works of that company. In the data that follow reference is made to four different alloys, designated as 2S, 3S, 51S, and 17S. Typical compositions of these alloys, by percentages, are:

	Cu	Mn	Mg	Si	Fe	Al
2S.....	0.08	....	....	0.15	0.40	99.37
3S.....	0.07	1.25	....	0.13	0.38	98.17
51S.....	0.08	....	0.60	1.00	0.35	97.97
17S.....	4.00	0.70	0.50	0.10	0.30	94.40

The tensile strengths of pure aluminum (2S) and various alloys are as follows (the point of 700 F having been selected as this represents the lower rolling-temperature range):

	At 75 F, lb per sq in.	At 700 F, lb per sq in.
2S.....	13,500	1850
3S.....	15,500	2900
51S.....	14,000	2000
17S.....	25,500	4300

These figures apply to rod instead of sheet, as no figures are available on sheet at the higher temperature. The 2S and 51S were rolled rod, and the other two were extruded. These figures are from "The Aluminum Industry," Edwards, Frary, and Jeffries.

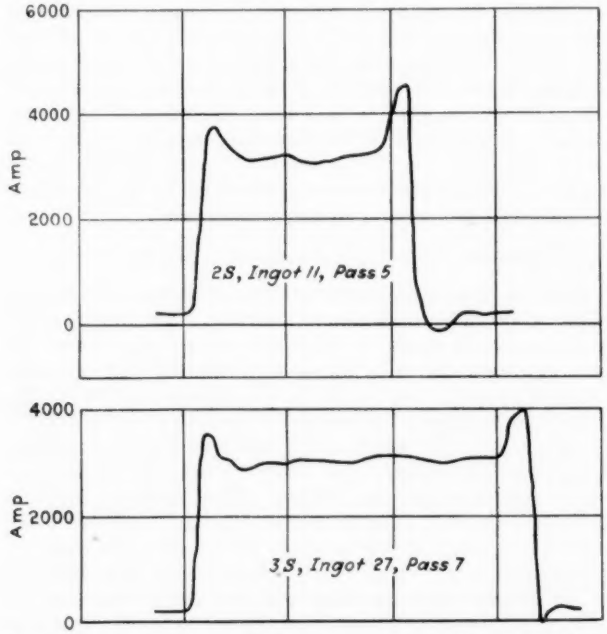


FIG. 7 TYPICAL AMPERE CURVES FOR AN 84-IN. THREE-HIGH PLATE MILL
(Rolling pure aluminum and a 1.25 per cent manganese alloy.)

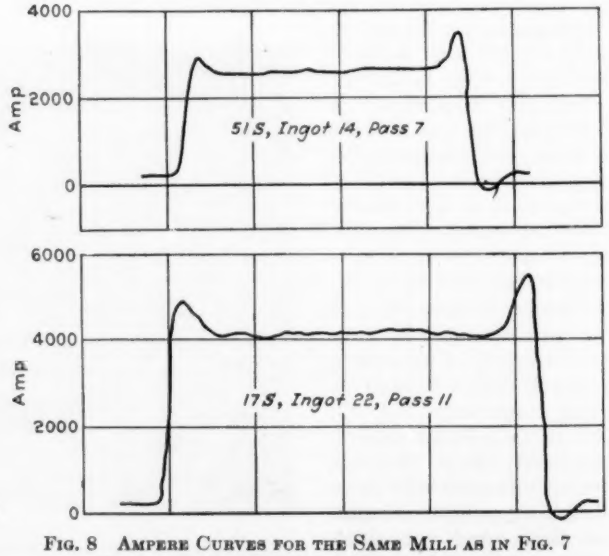


FIG. 8 AMPERE CURVES FOR THE SAME MILL AS IN FIG. 7
(Rolling a magnesium-silicon alloy and duralumin.)

The mill on which these power tests were made is driven by a 2000-hp, variable-speed, direct-current motor having a rated speed range of 165 to 330 rpm. This motor is geared to the mill through a reduction gear with a ratio of 6 to 1.

A two-element oscillograph was used to record amperes to the motor, and time was recorded by a jog in the ampere curve at 0.209-sec intervals. A McCollum-Peters telemeter pressure plug was used under one mill screw, and this was connected to the voltage element of the oscillograph and produced a curve representing pressure in exact time relation to the ampere curve.

Horsepower requirements for various reductions for the four alloys are shown in Fig. 6. These curves are based on reductions of about 25 per cent, the higher powers being obtained on the first passes. No widening passes were taken, and the width of the slab remained at 42 in. The ingots as started were 7 by 42 by 50 in. and weighed 1500 lb. The rolling speed was held at 230 fpm, and the ingot temperature were 850 F, which dropped down to about 700 F before the slab was finished. These curves show the average horsepower over the rolling pass, but the ampere curves from the oscillograph indicate peaks at the beginning and end of each pass. Typical curves for the various alloys are shown in Figs. 7 and 8. The peaks are from 0.25 to 0.50 sec duration, and their values vary from 125 to 140 per cent of the current during the rest of the rolling pass.

The investigations made at Alcoa verify the results of the tests made at the Bourne-Fuller Works, indicating that 17S alloy requires about 50 per cent more power than steel. However, 2S requires much less power than steel, possibly in the ratio of 6 to 10. The curves in Fig. 9 compare these two alloys with steel. The data for 2S and 17S were taken on 1500-lb plate ingots having the same dimensions as those mentioned. These ingots were rolled to $^3/_8$-in. plate on the 84-in. plate mill at a rolling speed of 230 fpm. The curve for steel is taken from a paper by H. A. Winne in *Iron and Steel Engineer*, June, 1932, and refers to an ingot $12^1/_2$ by 33 by 50 in., weighing 5700 lb, rolled to plate $^3/_8$ by $35^1/_2$ in. on a three-high mill having plain bearings.

A comparison of the power required to roll steel and 17S channels is made in Fig. 10. The information for the 17S curve was secured during a test made at the Massena Works of the Aluminum Company of America when an 8-in. channel was rolled from a $5^1/_2$ by $8^3/_4$-in. bloom on a three-high mill with plain bearings at a speed of 350 fpm. The steel curve is from the H. A. Winne article and refers to a 10-in. channel rolled from a $8^1/_2$ by 11-in. bloom on a three-high mill with plain bearings and a mill speed of 400 fpm.

When comparing power required to roll aluminum and steel, results are misleading unless the rolling temperatures are taken into consideration. The increase in power required, due to falling off in temperature during rolling, is quite different for the two metals. This relation is shown in Fig. 11. For 17S the increase in power required due to a decrease in temperature from 850 F to 700 F is in the neighborhood of 38 per cent, while for steel the power required at 1800 F is 100 per cent greater than that required at 2100 F. This indicates that while 17S at its normal rolling temperature requires more power than steel at its normal rolling temperature, the former may be rolled through a relatively wider range of temperature without serious increase in power and consequent stresses on rolling equipment. Results obtained from the tests on 17S previously mentioned as compared with results from various tests on steel indicate that while 17S at 700 F requires 50 per cent more power than steel at 2100 F, 17S at the same temperature requires 30 per cent less power than steel at 1900 F.

Extrusion

The first question that arises in regard to the extrusion of aluminum is why this process is used in preference to rolling. There are several answers to this question, each being in itself a logical reason why extrusion is resorted to. Suppose we classify these answers or reasons by products extruded in practice showing why extrusion is to be preferred over rolling or, in the case of tube blooms, over piercing.

It is hardly conceivable that even a mill devoted entirely to rolling structural shapes would be equipped with rolls for rolling all of the many shapes in their various sizes that are listed in the handbooks as standard products. But it is conceivable that customers will buy what they need or, in some cases, what they think they need, without giving much thought to the question of available rolls on which to roll their requirements. No matter what the reasons may be, producers will be offered business for

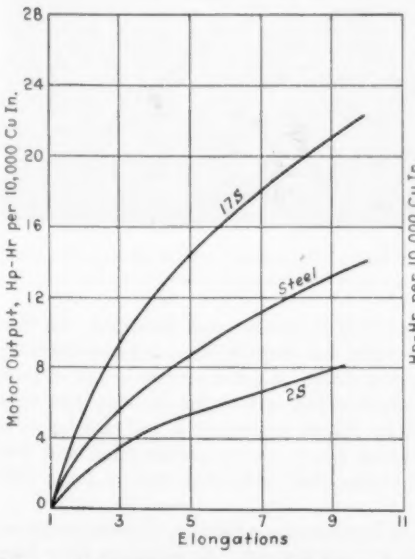


Fig. 9 Comparison of Power Required to Roll Pure Aluminum, Duralumin, and Steel Plate on a Three-High Plate Mill With Plain Bearings

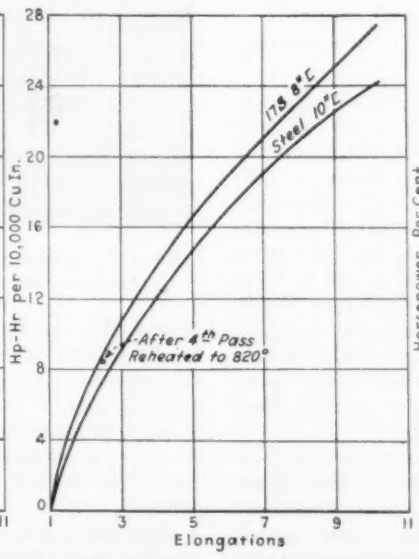


Fig. 10 Comparison of Power Required to Roll Duralumin and Steel Channels on Three-High Mills With Plain Bearings

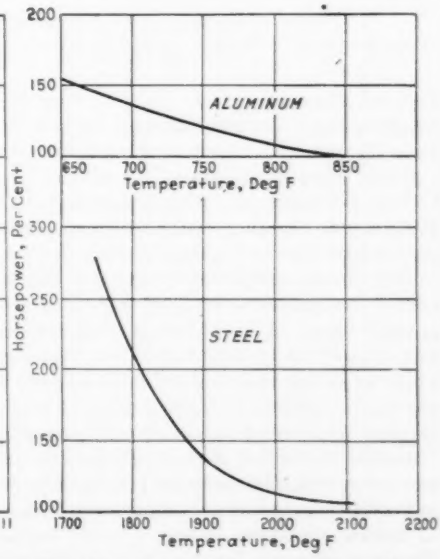


Fig. 11 Curves Showing How Power Varies With Temperature of the Metal Being Rolled, in This Case Steel and Duralumin

standard shapes for which they have no rolls available, and in the case of aluminum alloys, the way out is to turn to extrusion. It is a much simpler problem to provide extrusion tools for a new section than to provide rolls for a shape mill, as the cost will be less than a tenth of that required for rolls and the time required will be days instead of weeks.

The question of special shapes is of extreme importance, as in this classification falls a large percentage of the extruded aluminum-alloy business. These special shapes may be odd sizes of

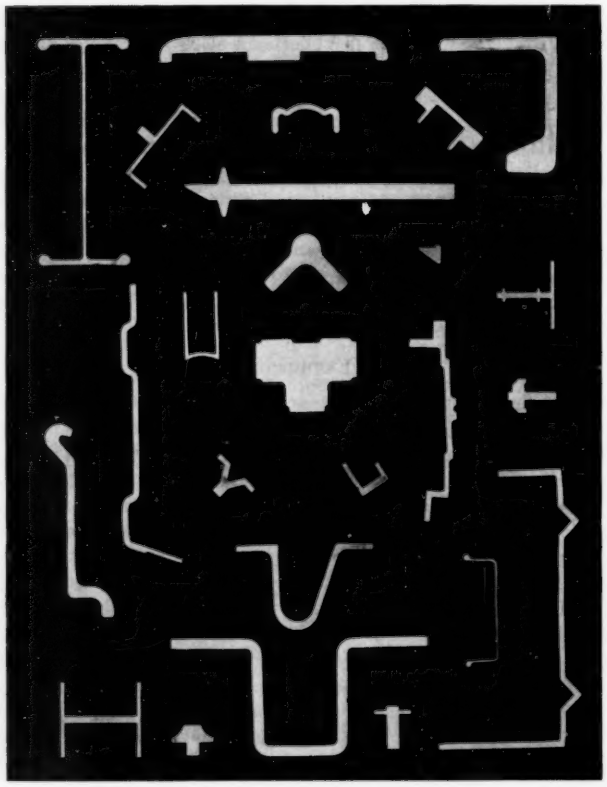


FIG. 12 MISCELLANEOUS EXTRUDED SHAPES
(These are typical of the sections used for aircraft, bus, and truck bodies and the like.)

so-called standard sections or they may be sections that are nearly standard but are changed just a trifle to suit the designer's necessity or whim. Some of the sections shown in Fig. 12 fall into this latter classification; for instance, the I-beam with the bulbs at the extremities of the flanges and the angle strengthened at the point. Another class of special shapes is that which covers the sections used in building construction for trim. These are usually of such complicated shape that they could not be rolled even if the quantities involved warranted the purchase of rolls to make them. A few of such sections are illustrated in Fig. 13, and an assembly showing how these sections go together to form a pilaster joining two window frames is shown in Fig. 14. Generally these sections are thin and carry no loads whatever. They are used for covering and ornamental purposes.

Another important class of extruded products includes those sections on which further work is to be done, such as tube blooms and sheet slabs as well as rectangular billets. Today practically all drawn aluminum tubing is made from extruded blooms. The principal reason for this is that the bloom that goes on to the draw or push bench must be as nearly perfect in finish and dimensions as it is possible to make it. Extrusion seems to be the only practical method for obtaining these results. It is possible to pierce aluminum-alloy billets, and this practice was followed for several years, but was abandoned for the extrusion process principally for the reason that the extruded blooms were much more nearly perfect, resulting in lower scrap losses and better finished product. Extruded blooms may also be made in a larger variety of sizes than is possible on a piercing mill, thus relieving the draw bench of considerable work in reducing the area of the bloom. It was also common practice until within a very few years to make tubing from sheet circles, using a cupping press and hydraulic push bench to make the blooms, but this practice has also been discarded for the extrusion press. It is regular procedure today to extrude tube blooms as large as 12 in. in outside diameter with $^1/_2$-in. walls and as small as 1 in. in diameter, with a wall thickness of $^1/_{16}$ in.

The extrusion of sheet slabs may be accomplished without serious difficulty, but it is done only under exceptional circumstances. If a hot mill is convenient and in operation, rolled slabs are much less expensive, and of course can be rolled in sizes not possible by extrusion. Reports have been received from Europe of the extrusion of sheet slabs in tubular form. These tubes are slit and flattened for rolling. There seems to be nothing to recommend this process if a rolling mill is available for rolling hot ingots.

There is still another class of extruded products—those made from alloys that cannot be rolled. Certain aluminum-mag-

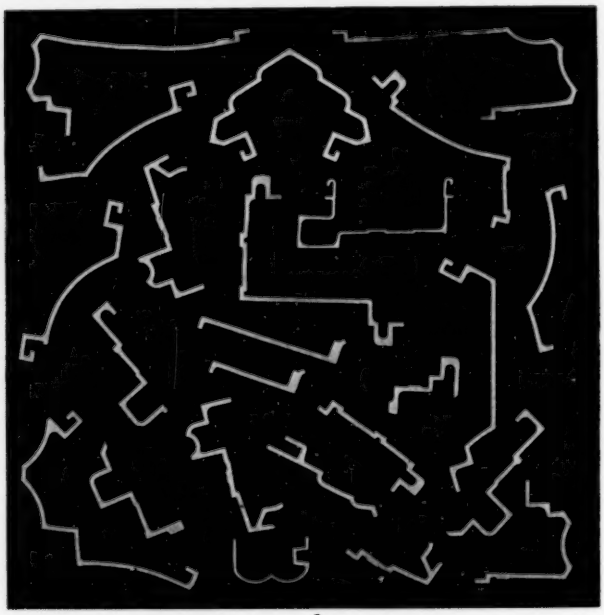


FIG. 13 EXTRUDED SHAPES USED FOR ARCHITECTURAL BEAUTY
(These sections are made thin and usually carry no stresses.)

nesium alloys are so brittle in the hot cast condition that they break up badly on rolling, but extrude without great difficulty. On account of its very desirable physical properties one of these alloys is used to a considerable extent for drawing fine wire. It has been found to be almost impossible to roll cast ingots of this alloy, so billets about $1^1/_2$ in. square are extruded, and then put on the rod mill, where they are rolled without great difficulty.

All extrusion is done from ingots extruded in the cast condition; in other words, they are not scalped. Experiments have been made to see if rolled blooms have any advantage over cast ingots for extrusion, but there seems to be nothing to be gained by their use. Extrusion is done hot, the temperatures used being the

same as for rolling the corresponding alloy, and there is evidence to show that power required for extrusion varies with the temperature of the metal in about the same way that the power required for rolling varies.

While it is true, as has been stated here, that certain alloys extrude with less difficulty than they roll, this is not true of all alloys. Pure aluminum rolls and extrudes more easily than any

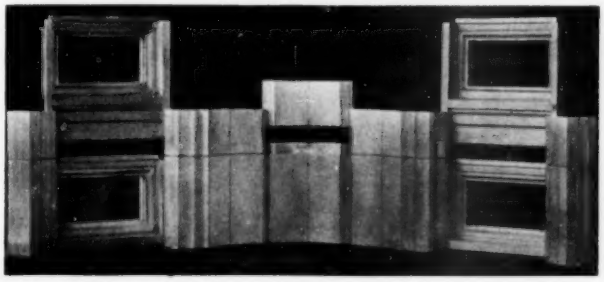


FIG. 14 AN ASSEMBLY OF EXTRUDED SHAPES INTO A PILASTER AND TWO WINDOW FRAMES

of its alloys. Those alloys of the duralumin class, containing about 4 per cent copper, require about three times as much power to roll as pure aluminum. While there are no definite data on the comparison of power required, if possible speed of extrusion be used as the measuring stick, then the duralumin-type alloys require ten times more power for extrusion than does pure aluminum. On the other hand, the alloys containing higher percentages of magnesium cannot be rolled at all, but extrude without serious difficulty.

In color and surface finish extruded products are quite superior to rolled shapes. This is very important where aluminum is used for decorative purposes, particularly for architectural work. There an alloy containing 5 per cent silicon is used for its desirable properties of color, strength, and resistance to corrosion.

Just what the immediate future will bring forward in the art of rolling and extrusion of aluminum alloys is hard to forecast. It does not seem probable that there will be much change as far as size of equipment and dimensions of product are concerned, except in the case of rolled shapes. Here there seems to be a tendency to go to sections as large as 15 in., where 10 in. is the largest used up to the present. As far as plate is concerned it would appear that the present equipment, which includes a 144-in. mill, will be large enough for some time to come. The same is true of extrusion, where the present equipment, which extrudes a 12-in. tube bloom, should be sufficient for all purposes for the present and the near future, at least. Progress during the next few years will more than likely be in new alloys with better physical properties and lower costs of production.

MAN-55-1

Progress in Management

INDUSTRY during the last year has had to face not only the problems that arise from the conditions brought about by business depression, but also those that come with any financial crisis. The year 1931 left a heritage of world economic disasters that commenced with the failure of the Austrian Credit Anstalt and spread throughout all Europe, affecting Germany and England, and that finally shook our own economic structure to its very foundations. The victory of the American dollar in June of this year marked the turning point in economic affairs, and reestablished, for the time being at least, business confidence to the extent that the administrators of business are now able to face the future with some degree of assurance that the worst of the financial storm has passed. Management has yet many crucial problems to solve before industry can again fall into step in the steady march of progress.

Management learned many valuable lessons in the depression of 1921, and it has the opportunity to learn many more out of the business depression and financial crisis of 1931 and 1932. The administration of business in a time of rising prices is not as difficult nor as fraught with danger as it is in a time of falling prices. Business executives have followed in the past the fetish of volume and mass production, without regard to the ability of the consuming public to absorb the ever-increasing flow of goods that the highly specialized wheels of industry are prepared to turn forth. It is now evident that to artificially stimulate business to bring in volume, for the purpose of reducing costs and supposedly for increasing profits, merely puts off the evil day of reckoning. Industry in the future must organize itself and be prepared to supply that quantity of products, first, which can be economically distributed and, second, which can be economically produced. In fact, all phases of business activity must be balanced and equally well managed if a state of economic equilibrium is ever to be attained.

The day of competition is past. Industry must learn how to cooperate. There is just so much volume of business for every one when the flow of goods and money or credit is normal. Plants must be built or reorganized with this in view. Large expenditures of capital in productive equipment of a specialized nature are to be avoided. Volume can no longer be sought as the road to profits. Profits will come through the recognition that there is a certain normal volume of goods which at any time can be consumed, and that productive facilities for single-shift operation will be based on this quantity. Temporary increases and decreases in business activity can then be adjusted for, without serious dislocation, and productive capital can be much more efficiently employed.

Management, heretofore, has been too greatly concerned with the evolution of methods, systems, and mechanisms. The techniques of management must now be brought into a greater state of simplicity, flexibility, and sensitiveness if the hoped-for stability of business is ever to be gained. More than ever the management of industry must be entrusted to men of creative ability and leadership, with judgment and foresight, who will be able to sense the natural change in economic conditions and adjust the course of business to maintain a more stable situation, provide adequate opportunities for employment, and return a reasonable profit from all operations.

At least it is encouraging to note that throughout this country there is a growing appreciation of the importance to industry of an enlightened public opinion on manufacturing, social, and economic matters. In particular there is a broadening interest among engineers in the fields of economics and distribution. Furthermore, the attitude toward labor-saving machinery is changing in that it should be employed to save labor rather to increase capacity and production; that is, to reap the benefits of greater leisure through shorter hours. There is also the recognition of the advantages of versatile employees and supervisors which arises as a corollary to slower mechanization in industry to maintain greater flexibility. Likewise, there is evidence of an increase in the caliber of those in supervisory positions in industry by the employing of high-school- and college-trained men and women. Even at this time when the greatest economy must be practiced there is an ever-increasing recognition of the necessity for organized research into marketing, new products, production methods, and business methods. Studies are also being made of the costs of distribution and of production with particular regard to overhead costs in order to detect the characteristics of the fundamental cost structure peculiar to individual plants.

Developments in the Marketing Field

The past year, having been one of extreme business depression, has demanded more attention than ever before to the study of marketing. Furthermore, existing conditions have enabled those who gave proper attention to this subject to prove with satisfaction to themselves and to others, not only the value, but also the absolute necessity of thorough and continued study of this important factor of business.

In addition to materially decreased volume, prices generally were lower; as a result, the manufacturer was confronted temporarily with a prohibitive marketing cost, a situation which required prompt attention and careful analysis. Among many manufacturers this situation was realized and dealt with successfully.

Such manufacturers, including many nationally known companies, have succeeded, not because their problem was simpler, but because they took such steps as were necessary at the proper time.

Sales personnel, both direct and indirect, was reduced, particular attention was concentrated on competitive situations, and territories were carefully studied to eliminate those sections expensive to cover. Information available from advertising media was utilized to take full advantage of the best markets. Through careful study, better coverage per salesman was effected and greater thought given to allocating individual sales effort, eliminating certain difficult customers whose business was previously sought.

One large food company falling under this category has succeeded in keeping its percentage of marketing expense within 1 per cent of that of the previous year. Some motor-car manufacturers and certain utilities show results almost as interesting.

On the other hand, many companies have had a most difficult situation. Business at times had to be refused due to unsatisfactory credits. It was found in some quarters that expense reductions were immediately followed by decreased income. Reconstructed budgets were continually found short on the income side, resulting in continued unbalanced situations, so that in many cases all profits and assets were sacrificed, with receiver-

Presented at the third session of the Management Division, Thursday morning, December 8, 1932, during the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Personnel of the Executive Committee of the Management Division for 1932: Fairfield E. Raymond, Chairman; George W. Kelsey, Secretary; William B. Ferguson, Vice-Chairman; Charles W. Lytle, John R. Shea, Joseph A. Piacitelli.

ships resulting and many companies going into the hands of the banks or of other creditors.

Some of these results were inevitable due to lack of sufficient reserves, insufficient purchasing power, or lack of demand for the product; others were due to delay in action; and some were due to failure to properly follow out the basic principles of marketing in what has become a buyer's market.

The machinery industry has been severely hurt; some sales organizations have been almost sacrificed; orders have in many cases been taken at less than cost; and terrific losses have been sustained.

Other industries, sugar for example, awaiting the outcome of a government suit, have just marked time.

Certain farm-machinery manufacturers have shown ingenuity in stimulating sales by agreeing to accept payments for some items on the basis of corn and wheat at certain fixed prices above those prevailing at the time.

Several outstanding developments in marketing have occurred during the year, such as:

(1) Development of a general sales consciousness among factory and office personnel of companies as well as stockholders, with definite material assistance to their respective sales departments
(2) Growing recognition of necessity for research into business methods and marketing
(3) Better study of consumer requirements—that is, determination of products most suitable for customers' use and providing the least sales resistance
(4) Greater utilization of data furnished by agencies, associations of manufacturers, marketing associations, chambers of commerce, etc. to determine the most lucrative territories for various products
(5) Continued active interest of bankers and financial institutions in assisting industrials to develop improved marketing methods
(6) Government activities for collection of marketing data, such as the Louisville grocery survey, national drug survey at St. Louis, retail credit survey, study of causes of business failures and bankruptcies of individuals in New Jersey.

Many industrials and even industries have used the services of trained economists, engineers, and marketing counselors effectively in revising their sales policies and organizations.

An interesting situation existed in one representative concern where over 90 per cent of the business was obtained from less than 15 per cent of its customers. A careful study developed a change in policy of handling the other 85 per cent approaching that of the mail-order house, which considerably reduced total expense and improved the business from the smaller customers.

Margins previously existing for marketing have so diminished that little remains available for errors or guesswork; hence the necessity for even more exacting marketing efforts in the future.

Stabilization of Employment

In March, 1932, there were between eight and nine million unemployed in this country, and it is doubtful that any material reduction has taken place since then. The seriousness of this situation has brought home the need for a better planning process such as will provide guidance and control over our economic life and lessen the likelihood of devastating and socially harmful unemployment. As a result, five official and semi-official as well as many other private proposals have been presented, all aiming at a more stabilized economy, which in turn may be translated into stabilized employment. These "plans" do not reflect the desire to imitate Russia, but rather the eagerness to develop our own American planning based on voluntary cooperation, assisted by the official establishment of a National Planning Organization and the amendment or even repeal of the American anti-trust laws. The proposals differ largely in the matter of control. Some would intrust it to the state, others to public bodies, others[1] to industry itself; some advocate an advisory control, while others a real supervisory control.

Aside from the foregoing proposals, the United States Chamber of Commerce offers a stabilization program for the "continuity of business and employment," with long-time measures similar to those of other plans. Immediate measures are also proposed for meeting the problem of unemployment. These include centralized local organization following the lines of the so-called Rochester plan; central registration of workers seeking employment; and planning and execution of public works. The directorate of this body recently approved the "40-hour week," which is in line with its stabilization program.

A study of the causes of seasonal fluctuations in business was made by a group of New England business men, and attempts toward stabilization of employment for the individual worker are summarized by Edwin S. Smith as follows:

1 Training of versatile workers, permitting transfer from less active to more active operations
2 Reluctance of worker to transfer
3 Maintenance of usual earnings when on temporary job
4 Transferring between industries to meet seasonal peaks that do not occur at the same time
5 Part-time employment
6 Shortening working hours.

With regard to this last item a recent survey in 77 different industries involving nearly 38,000 establishments and affecting approximately 4,000,000 showed that 5.6 per cent of all the employees covered in the study were on a permanent five-day-week basis. As far as the percentage of workers on the five-day-week basis is concerned, the automobile industry leads, with 44.3 per cent of the employees in the establishment reporting working on that basis. It is followed by radio, dyeing and finishing textiles, and aircraft industries. Specific information has been gathered from 50 companies regarding their operation of the five-day week, particularly relating to the use of skeleton crews on the sixth day. Of the 29 companies using skeleton crews three run a staggered system whereby every one in the office is out one-half day some time during the week, but at least half of the personnel is on duty at all times.

Thoughtful consideration is being given to the adaptation of the six-hour day to industrial operation. It has already been put to the test by a few companies that have considered the present an opportune time for experimentation. As a fine example of this, the Owens Illinois Glass Company restored its entire working force to employment by operating four six-hour shifts instead of three eight-hour shifts. Of the 318 companies reporting in a survey of what is happening to office working hours, 188 have made no change, 116 have decreased the hours, 4 have lengthened them, and 10 have special plans in some departments. Of those reducing working hours, 17 consider the change a permanent one, 32 are undecided whether the change is permanent or not, 37 consider it a temporary emergency, and 30 have made the change for the summer only. About 80 per cent of those reducing working hours have done so by discontinuing Saturday work. The pay in these cases has been reduced proportionately.

Legal steps have been taken in some states to provide for unemployment insurance that will serve for the purposes of both

[1] See "American Planning in the Words of Its Promoters," Hugo Haan.

relief and of an incentive for regularization. Wisconsin's Unemployment Reserves and Compensation Act compels employers to establish "systems of unemployment compensation," if they have not already voluntarily done so, by June 1, 1933. Similar measures are being considered by Indiana, Ohio, and Michigan.

In view of the legislation already enacted in Wisconsin and under contemplation in other states providing for the establishment of funds for unemployment relief, a committee appointed by the National Electrical Manufacturers Association to consider the plan presented by Gerard Swope has given its first attention to this feature of his proposal. The committee proposes a Nema mutual unemployment benefit plan which includes all employees that have served 12 consecutive months.

Accounting and Finance

During the boom period of 1929, working capital was frequently diverted to fixed assets for new equipment, plant extensions, and the like. This was subsequently replaced, as volume transactions made the need of it apparent, by bank loans. Then the business tide turned, and those bank loans either were a decided embarrassment or an actual menace in too many cases. Similarly, high bonded indebtedness, too high a proportion of borrowed to owned capital, particularly in industrial enterprises, has proved to be the *bête noire* of many boards of directors.

There was a feeling at one time that if the market for preferred or common stocks was unfavorable and the discount for financing too high and that if the bond market was good, it was just as satisfactory to finance with bonds, not recognizing what the ultimate effect might be or even contemplating the possibility of earnings shrunk below the fixed charges on capital.

Serious consideration is being given to the appropriate value to be assigned to assets. They have been revalued in the light of present conditions and possible future use, surplus and obsolete assets have been written off in many cases, and the surplus account at least partially reestablished by scaling down the stock and restating the capital in a number of instances. The published statements of many of the well-known companies in the country reveal this process as going on.

It is evident that industrial managements from now on will give more thought to the character of their financial structure, its set-up, and what the relationships of the balance-sheet items really mean.

A change has taken place in the use of budgets, and the technique has improved. With sales forecasts becoming obsolete by the time they were made, during the most violent period of the business recession, a means had to be sought that would regulate and control operating expenses more closely than heretofore. Therefore the accounting statistics were utilized as a basis to predetermine the fundamental relationships of non-variable, variable, and semi-variable expenses to production of varying rates. Similar studies applied to the distribution and general expenses determined their fundamental relationships to varying sales volumes or other suitable bases. This permitted the establishment of budgets monthly in advance, on a ratio basis and adjusted to the immediate future program.

Standard costs for predetermined steps or ranges in production were similarly established, and more attention has been paid to marketing and distribution expense accounting. The accounts were thus used to predetermine goals, as set forth by the budgets to furnish the basis for establishing standard costs and standard expense relationships rather than a source of recorded history when it is usually too late to do anything about it.

More and more efforts have been made during this period of subnormal business activity to mechanize accounting operations to the fullest economical extent, to simplify the procedure, to eliminate useless forms and reports and to consolidate the methods wherever practicable, and so utilize the accounting methods as to permit the establishment of and attainment of a predetermined goal, even if these goals were set up for comparatively short periods.

Plant Layout

The prolonged business depression has in some cases accelerated plant rearrangement. Many of the larger industries are changing from the functional type of manufacture to the unit type where a product is largely made complete in one department or by one group of machines. Micro-motion study and time studies have pointed toward a more intensive correlation of machines, which in turn has affected plant and building layout arrangement.

Further applications have been made in the use of artificial lighting and air conditioning, largely in connection with special processes where direct economies could be obtained.

Equipment

One of the outstanding things pointed out by this depression is the fact that most industries are overequipped from the standpoint of taking care of a normal growth in business. In the previous depression the outstanding fault was one of large inventories of raw materials and work in process; in this depression it is plant investment, including grounds, buildings, and machinery. More sound, long-termed predictions and planning for future growth must be devised in order to hold such investment to a normal figure.

In the development of new and improved machines a greater application of ways and means for reducing noise has been effected and improved devices have been made for the measurement of such noise.

A wider application has been made of hydraulic devices in the movement of machine elements, such as on heavy punch presses and drill presses.

Machines used in processes resulting in dust, etc. have until recently been exhausted by separate duct systems; self-contained units are now being made a part of such machine units, which together with individual motor drives make such equipment complete.

Manufacturing Processes

Most manufacturers have directed their engineering effort almost exclusively to the design of new products and their production on a low cost basis, using existing machines and processes, and as a result the last year has shown a minimum of development in the manufacturing processes field, outside of further extension of the use of tungsten and tantalum carbide and other improved cutting materials. In this connection equipment has been developed which accurately measures the force exerted on a cutting tool, and data obtained in this manner should aid in the further application of these materials and also in the redesign of machine tools in order to use them at their maximum efficiency.

There have been further refinements in the application of optical and X-ray methods of inspection of material for defects, and greater accuracies have been obtained in the gaging of parts by improved electrical and mechanical devices.

The use of lighter metal alloys is finding an ever-increasing application in industry and is bringing with it a decided change in processes.

Future product design and process development are being influenced by new plastic molding compounds whereby a wide variety of permanent colors may be obtained.

Drying processes have been aided mechanically, and as a result both the direct cost and the investment in material in process have been reduced.

Maintenance

Increased recognition has come to the function of maintenance, particularly in its relation to unit cost of product. Such recognition takes account of its—

(*a*) bearing upon indirect costs
(*b*) responsibility for prevention of breakdown and shutdown losses
(*c*) operation of plant and facilities at lowest cost for power, transmission, heating, lighting, and similar services
(*d*) strategic position in making observations as to equipment performance.

Progress has been made in the application of wage incentives and in task standardization. For example, the plant engineer of a company of note and considerable size has broken down and charted the inspection, testing, and repairing details in connection with all important equipment and has his operatives perform their work constantly by these charts. The maintenance department is conducted on the wage incentive basis.

Cost accounting, as it applies to maintenance, has been scrutinized, looking toward isolation and simplification. In the plant referred to, the cost-accounting method is being revised to be operated for maintenance as directly as is the function itself.

Acceptance of the practice of budgeting expenditures has increased in spite of inherent difficulties. The plant engineer of a manufacturer expending annually more than $450,000 for maintenance reorganized the maintenance department to insure centralized control and positive operation on a budget and schedule basis.

Much thought and discussion have been given to organization by personnel and duties for efficient and economical conduct of maintenance. Intense interest has been shown.

Production Control

There has been little significant progress during the last year in production control due largely to the lack of orders. For the most part the trend has been decidedly toward simplification and flexibility. What progress has been made has come chiefly through the results of cost analysis and the discarding of those methods that become uneconomical and unwieldy when applied to the markedly reduced operating schedules with which industry has had to contend. The trend toward "product production" is characteristic of the times, as the organization of the processes of manufacture according to products eliminates much of the old-time routines of production control, with its planning, scheduling, routing, and dispatching. A production budget for each manufacturing period takes the place in such instances of the continuous attention which had to be paid to the maintenance of production schedules.

Material Control

The conservation of material in manufacturing processes has been greatly aided by further extension of the use of raw-material specifications, improvements in tooling methods, and (what is most important) a more exact accounting control of actual wastes as they occur and the reductions of such waste by carefully planned intensive campaigns. In such cases the operator who is directly responsible for each step in the process has contributed greatly to the elimination of waste once he has realized the full significance from a dollars and cents standpoint of the loss and how such losses may be controlled.

Safety

Industrial accidents have generally decreased. This is due largely to reduced personnel. The careless workers have been weeded out, while the most efficient remain. In the paper and pulp industry lost-time accidents in 125 mills were reduced 19 per cent from 1930, while the number of days lost were 25 per cent lower in 1931 than in 1930.

Job Standardization

The depression has not lessened the amount of "motion and time study" as much as might have been expected. The necessity for reducing all costs and for getting out improved products has, in fact, advanced this work in many instances. In other instances where this work has been less essential than economy in overhead, curtailments have occurred, but in many of these cases the men, because of their training in analysis, have been transferred to effect other economies, and in a few instances such analysts have been lent to customer companies for the purpose of studying equipment from the user's point of view. This latter innovation has not only helped customers, but has resulted in equipment improvement and in some new business for the equipment makers.

The most outstanding development is, however, the continued extension of interest in micro-motion as a means of training any job analysts. Allen Mogensen, through "Factory and Industrial Management," and now through his new book, "Common Sense in Motion and Time Study," has succeeded in arousing company after company, while David B. Porter, through New York University, has succeeded in enrolling numerous company men for this training by means of night classes. Men so trained are able to make use of the principles of micro-motion without using the measuring technique originally thought necessary for micro-motion. What time measuring is necessary is done with the stop watch; hence the term "compromise method."[3] The results of this compromise procedure are infinitely superior to the best results of ordinary stop-watch procedure, and the opportunities are greatly extended over those of the full micro-motion procedure. Finally, there is a movement on foot to improve the definitions of the therbligs which has long been needed.

Wage Rates

The almost universal reduction in wage rates which has come during the last year and a half has led some to think that the American high-wage theory has been discountenanced. Nothing is further from the truth. The general movement for reduction did not get under way until commodity prices had come down, so that it is only by reduced working time that there is any reduction in real wage rates. An early-summer survey of the N.I.C. Board covering 1718 American business concerns does report a 26.6 per cent decline in employment, to say nothing of part time and short time, but the average reduction in executive salaries, weighted to the number of persons affected, was only 14.9 per cent, that of other salaries 13.1 per cent, and wages 11.1 per cent. The worker's dollar was found to be worth 22.7 per cent more in May, 1932, than in September, 1929. Thus it is not a matter of paying less in real wage rate, but rather a matter of too little working time. When the latter is taken into account, there is an average reduction of 23.1 per cent for the real pay envelope. This is certainly bad, but it does not indicate any repudiation of the high-wage policy. In fact, a few employers, who could do so, have actually raised wage rates in order to help recovery. On the other hand, since those fortunate enough to have full time are now receiving more real wages than in 1929, it is possible that further reductions in rates might be justifiable, especially if recovery does not come promptly. The unprofitability of agriculture is in part due to high urban standards. Since the products of agriculture are priced by world

[3] See forthcoming revision of "Management's Handbook."

competition, they cannot be artificially raised in a single country. Thus if further reductions are made in urban wage rates, they will be made with the purpose of extending work opportunities, and if standards of living are lowered for full-time employees, they will be corrected as promptly as conditions of the whole permit.

Wage Incentives

When companies are particularly hard pressed, there is a temptation to do away with the extra financial incentive plans, and that is what has happened here and there. American employers have, however, learned that properly designed and correctly operated incentive plans pay for themselves. The more pressing the labor cost problem, the more is this true. The solution of unemployment does not lie in lax work habits. Furthermore, it takes time to develop correct work habits, to say nothing of dropping lax ones. The best-managed companies may have reduced the amount of "bonus," but they are not abandoning all "bonuses." It is a good time to revamp plans and to see that the best type of plan is applied to each kind of work. In short, employers are more particular about all such adjustments during depression than during times of easier profits. The chapter on this subject in the forthcoming revision of "Management's Handbook" should be helpful in making this readjustment.

Industrial Relations

The lack of stability in employment has made progress in some phases of personnel management rather difficult. However, the unemployment situation has brought about a wider interest in this field on the part of general executives.

Of the replies received by J. E. Walters from representative industrial and business concerns of the Middle West, nine companies have added new activities such as foreman training, relief-fund work, and unemployment pensions. Eighteen personnel departments have increased or decreased less than other departments of their companies, while only three have decreased more or discontinued. No records were found of any company that maintained relief activities for employees laid off during the depression of 1921. This in itself is indicative of a trend toward more humanism in industry. Another trend worth mentioning is that the smaller companies are becoming more personnel minded.

The Institute of Human Relations is conducting industrial investigations centering about a "total science" study of human problems of technological progress. The instance selected for the first investigation is the so-called "multiple loom or stretch out" system in the textile industry—the increase in number of looms or spinning frames tended by the individual operator. To date, studies of twelve[3] typical mills in the North and South have been completed. The work is being done by a staff headed by Prof. Elliott Dunlap Smith and supervised by a committee of scientists representing various fields.

Dr. E. K. Strong, of Stanford University, has devised a test known as the Strong vocational test, and has given it to a number of people in various occupations to discover range of interest. The student may then compare his interest pattern with those of the various occupations in which he is interested. Stevens Institute has set up a summer camp for high-school juniors and seniors in an attempt to measure quantitatively the interest of students through the methods developed by Johnson O'Connor and W. V. Bingham.

[3] Ten more than reported in previous publications.

Education

For the first time data are available as to the extent of industrial engineering education.[4] There were, in 1931, 35 engineering colleges in the United States giving full industrial engineering courses, 30 of which graduated that year 596 men. Four of these courses were too recently started to have graduates. The ratio of these graduates to business-school graduates is estimated at only 1 to 10, a proportion that is too small for normal conditions. Because of present conditions the graduates of 1931 and 1932 have not yet been absorbed, but industrial engineering graduates of earlier years have very largely retained their jobs, doing better in this than graduates from any other department of engineering. Recent meetings on both industrial engineering education and industrial training in the factory have indicated increased interest in these matters.

Research

Throughout this last year, when industry has been called upon to make most drastic retrenchments, it is of importance to note that there has been a point where the far-sighted executive has refused to curtail further the research activities of his enterprise. Many enterprises have weathered the effects of this depression chiefly through their reliance in research. Evidence of this is given in the report of the National Research Council which appeared this spring, covering data secured from some 400 companies representing 20 industries. Of these, 51.3 per cent increased their research expenditures in dollars in 1931 as compared with 1929, 18.7 per cent made no change, and 30 per cent decreased. Translated in percentage of sales, it is found that 75.3 per cent increased their research expenditures in percentage of sales in 1931 as compared with 1929, 15.7 per cent made no change, and 9 per cent found it necessary to decrease these expenditures. These figures, with slight variations, hold true of practically all of the industries. There has also come about a change in the emphasis of research. In 1928 a similar survey was made, and at that time the most important emphasis of research programs was in lowering production costs. Today the major emphasis of 37.7 per cent of the companies reporting was in the development of new products, 37 per cent in improvement of quality of existing products, and only 19.4 per cent in lowering production costs.

While the situation has undoubtedly changed materially in 1932, nevertheless where research expenditures were cut it was one of the last of the business aids to feel the pruning knife.

Aside from the imposing record of scientific research and engineering development, it is significant to note in the recently published report of the Business Research Council the extent of business and economic research being conducted by American colleges and universities. The research projects listed cover many aspects of finance, labor, marketing, production, transportation, public utilities, prices, business cycles, business management, economic history and geography, accounting, statistics, and business law, as well as many other topics of value to the business executive to a lesser degree. The report states that 108 institutions are conducting 620 research projects of this nature. Moreover, 80 series of statistical data are being continually compiled by these institutions on banking, securities, labor, wholesale and retail trade, foods, textiles, metals, lumber, real estate, buildings, power, automobiles, prices, and the like. Evidently there is a wealth of information being procured by these institutions which is of inestimable value to industry.

[4] "Collegiate Courses for Management, Engineering Education," vol. 39, S.P.E.E.

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Ten Years' Progress in Management, 1923-1932

By LEON P. ALFORD,[1] NEW YORK, N. Y.

Industrial management has undergone a severe test in the decade 1923–1932 due to violent fluctuations in industrial activity and sharp contrasts in the results of operating performance. Despite these changes it has made steady progress. It has been extended to all human activities where work is done. It has contributed to the accelerated rate of industrial development of the first six years, and its principles and practices have withstood the decisive trial of the deep depression of the last three. As an outcome of its application, well-managed concerns have passed through this period of business recession better than all others; as a result of its progress, management is now firmly established as a function in industrial operation. An examination of the changes in trends of some 25 industrial-management developments supports this major finding.



THIS PAPER is the fourth of its kind to treat of the progress of industrial management for the attention of the members of The American Society of Mechanical Engineers. The first was a committee report of 1912 under the title, "The Present State of the Art of Industrial Management." The second, a paper of 1919, was on "The Status of Industrial Relations." The third, a paper of 1922, a review of ten years' progress, was founded on the report of 1912 and was titled "Ten Years' Progress in Management." This paper seeks to review the changes and progress of another period of ten years, the decade just closing. For its foundation it reaches back to the three preceding summaries.

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On the whole the ten-year period, 1923–1932, has been marked in the United States by sharp contrasts in results of industrial operation. The mid-point recorded the greatest peak of peacetime activity that industry has ever experienced. As this paper is being written operation is estimated to be only a little more than 50 per cent of normal. There have been three movements of the business cycle in this decade. These are dated as expansion to May, 1923, contraction to July, 1924, expansion to October, 1926, contraction to December, 1927, expansion to September, 1929, contraction to date. Physical volume of industrial production surged up to its highest point in 1929 and then dropped sharply. Commodity prices and manufacturing costs have likewise receded. Statistics of production of the Federal Reserve Board are:

Year	Industrial production combined index, unadjusted
1923	101
1924	95
1925	104
1926	108
1927	106
1928	111
1929	119
1930	96
1931	81
1932 (January through June)	65.6

A comparison of the relation of profits to capital investment for 432 manufacturing concerns gives averages for 1920–1921 and 1929–1931. The figures for the latter period show the decline in profits during the present business recession.

Year	Percentage of profits to capital investment
1920	9.89
1921	1.17
1929	12.82
1930	6.55
1931	2.47

A remarkable increase is shown in the productivity of American industrial wage earners. The percentage rise for all industry from 1899–1919 was 4.7 per cent; that is, productivity had changed but little during the first 20 years of the twentieth century. The change from 1919–1927 is a 53.5 per cent increase. For many industries this change became manifest in the year 1921, thus placing most of the advance in the decade under consideration.

Upon the surge of prosperity that these increases indicate was founded the belief that economic poverty in the United States was almost banished. Some of the more optimistic prophesied that by 1931 or 1932 every one would have an abundance of the necessities of life. In gloomy contrast with this prophecy is the hundreds of millions of dollars already paid out for relief during the past two years and the organization of national relief agencies on a scale never before attempted to mitigate the human misery impending during the winter of 1932–1933.

Unemployment in the United States has fluctuated widely

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

during the past ten years. The estimated minimum number of unemployed among non-agricultural wage and salary earners was at its lowest point in 1923, with an average of 1,532,000. The corresponding estimate as this paper is being written is 11,000,000, with well-founded fears that this number will be increased by 1933. During the first seven years of this decade, employment for industrial executives was quite stable. Shifts in business connection were comparatively few. Beginning with 1930 this situation changed, and the turnover of management personnel has since been rapid.

These circumstances have brought a reversal of belief in regard to American industrial achievement. At about the mid-point of the decade statements like the following were commonly made concerning us: America has made an "immense advance" industrially, "America has experienced a period of unusual industrial prosperity. Millions of people have found their earnings increasing at a more rapid rate than their standard of living." America has become "the first power in the world." As the decade closes this quite different matured judgment of the situation has been publicly declared by engineers: "The problem of the relation of consumption, production, and distribution is one of supreme importance. More and more it will command the attention and devotion of the most intelligent, unselfish, and far-seeing men and women of the body politic. There is no human problem which compares with it in difficulty, in magnitude, in hopefulness. . . . what is being contemplated is nothing less than a purposeful changing of the course of history."

In spite of these violent fluctuations, management has made steady progress during the decade ending in 1932. It has extended to all human activities where work is done. It has contributed to the accelerated rate of industrial development of the first six years, and its principles and practice have withstood the crucial test of the deep depression of the last three. Increasing recognition is being given to the fact that well-managed concerns have come through the period of business recession better than all others. As a function of industrial operation the art of management is now firmly established because of the general acceptance of its fundamentals. The promise of the future is to lead on to a higher professional standing through the development of intellectual, societal, and moral attitudes and values. Should a report like this one be written at the end of the next decade, much of its narration may well be a stirring epic of human achievement.

The nature and extent of the change make the task of reviewing the decadal progress of management difficult. But this difficulty is somewhat relieved by the availability of certain management literature, no prototype of which existed in 1912 nor in 1922. The items which have been most helpful in the study are: "Bibliography of Management Literature," published by The American Society of Mechanical Engineers first in 1927, and revised in 1931; "Progress Reports of the Management Division of The American Society of Mechanical Engineers" from 1925 to date; and parts of the "Report on Recent Economic Changes."

In these contributions two trends in management are definitely pointed out—the one toward increased specialization, the other "more management per man." The "Bibliography" lists in round numbers 800 references (827 by count) covering the 35 years prior to 1923. The comparable figure for 1923–1932 is 5000 items (3915 by count for eight years, 1923–1930). This record indicates an amazing expansion in the volume of management literature. Its character has also changed. Quoting from the foreword of the 1931 edition: " . . . the trend of the development of management [is] indicated by its literature during this period. No longer are there many works dealing with the broad subjects of shop management and scientific management, but rather do they treat of specific phases or branches of management, such as economics, material handling, research, and safety. This, of course, is a logical result of the widespread tendency toward specialization."

Henry S. Dennison, in the chapter on "Management" in "Recent Economic Changes," commenting on the increasing release and utilization of creative and managerial abilities, has this to say: "There is today not only more production per man, more wages per man, and more horsepower per man, but more management per man as well." The reference date is 1928, the mid-point of the decade under consideration.

The approach to this review, as already indicated, is through the emphasized points in the three preceding studies of like nature. From this background the exploration has been broadened to include the activities and developments of management which were found worthy of comments in the annual progress reports. An advantageous difference in treatment between this paper and its predecessor lies in the rather generous introduction of quantitative evaluations. Facts and data of this kind were unavailable in the preceding report years.

Important Findings in Report of 1912:

1 Advance in doing work. "The unquestionable proof of the advance that can be made in unskilled work . . . and in ancient trades . . . . These are the most striking phenomena of all. (Par. 1.)
2 Change in mental attitude. The shift in viewpoint toward production was declared to be "the most important change and one that comprehends the others . . . seeking for exact knowledge and then shaping action on discovered facts." (Par. 22.)
3 Transfer of skill. The application of this principle was emphasized as "the prominent element in present-day industrial management." (Par. 29.) And "we have pointed out that the underlying principle, that is, cause in the widest sense, the application of which has built up modern industry is the transference of skill." (Par. 41.)

Important Findings in Paper of 1919:

The major lines of development in the effort to work out better industrial relations were indicated as:

(a) Profit-sharing plans
(b) Methods of wage payment
(c) Methods and laws to reduce the hazards in industry and mitigate the effects of injurious and occupational diseases
(d) Employment management
(e) Systems of mutual or joint control by employers and employees. (Par. 18.)

Important Findings in Paper of 1922:

1 Management principles. "Acceptance of the principles of management has broadened among engineers, executives in industry, and educators." (Par. 10.)
2 Management methods. " . . . demand for . . . facts as a basis of judgment has grown insistent in all good management. Appreciation of the possibilities and advantages of standardization, simplification, and elimination of waste has spread rapidly during the past two years. . . . The engineering or scientific method has extended in industrial cost accounting. (Par. 11.)
3 Significant developments. Especially significant developments include broadening the applications of management far beyond original Taylor mechanisms, establishment of management courses in colleges, "appreciation of the importance of the human factor in

industry," and declaration that the "service motive must prevail in industry." (Par. 12.)

4 Economics of industry. "Without doubt if a study of this kind be made ten years hence, the developments in economic knowledge will be found to have been one of the great steps taken in the development of management in industry." (Author's closure.)

No similar group of significant developments and trends is offered from the annual progress reports, for the reason that they deal principally with current evaluations rather than trends, but numerous quotations are included to show points of view and steps of progress.

It is unlikely that any two investigators would pick identical items to present as significant in a decade of progress, nor give, even approximately, the same emphasis to each one, nor the same interpretation to the surrounding facts. In this paper the first situation is made partially impersonal by the choice of the items. Of the 25 that are discussed separately, only one, "National Planning," lacks a reference in the preceding reports of The American Society of Mechanical Engineers. As to the second situation, a free use of quotations and citations tends to broaden the area of opinion and to impersonalize the comments.

From this place on, this paper gives a series of summary word pictures of the circumstances of change and progress of the selected developments of management in action. Each is considered separately, although none is independent. "The art of management today [1928] is in large part the progressive adjustment and integration of conflicting needs, conflicting influences, and conflicting purposes."[2]

National Planning. Management is faced with the greatest challenge of its career in the need and demand for national planning and world planning. This summons to still greater achievement comes from the profound influence of the depression that broke like a storm in 1929. More questioning and searching has ensued, directed at the organization and operation of industry, than at any other similar national institution. H. S. Person[3] presents the issue, clearly outlined like the ascending steps of a vast pyramid: "Stabilization of material factors is not sufficient; human relations must be stabilized. Stabilization of production is not sufficient; merchandising must be stabilized. Stabilization of production and merchandising is not sufficient; general administration must be stabilized. Stabilization of an individual enterprise is not sufficient; all enterprises in the industry must be stabilized. Stabilization of one industry is not sufficient; all industries of a nation must be stabilized. . . . stabilization of national industry alone is not sufficient; international economics must be stabilized. Achievement of any of these ends is a step toward a more balanced and harmonious industrial and social world life; each end is but a means to another and greater end."

International planning as a matter of fact must be dismissed at this moment as a stimulating speculation. National planning for the United States is in quite another situation. The idea has grown directly from the experiences with shop planning and the application of Taylor's theories.

The objective of national planning in its economic aspect is to attain and maintain "in the United States a standard of living that is high, broadly distributed, and free from severe fluctuations;"[4] in its industrial aspect it is to stabilize the operating performance of industrial enterprises; in its national aspect it is to minimize the fluctuations of business and avert disaster.

The principles to apply are the enduring fundamentals of management. Methods and technique must be evolved to satisfy insistent needs. Their prototypes are here in the mechanisms and procedures now applied successfully in hundreds of individual plants and in many coordinated industries.

Numerous plans have already been proposed. In general they vary between these extremes:

1 Those that seek to make industrial and business planning a function of the National Government. A governmental bureau, for example, would tell manufacturers what to make, how much to produce, and at what price to sell

2 Those that seek for voluntary regulation of the operation of industry by industry itself, under some governmental supervision to safeguard the rights and interests of the public.

We may confidently look forward to attempts at national planning during the coming decade. Management will give a good account of itself in this new test.

Service Motive in Industry. Scrutiny of the literature and programs of conventions dealing with management reveals that "profit management" has been a favorite topic of discussion. Industrial executives have, in the main, felt and expressed their responsibility toward their own organizations and their investors, and to a lesser degree toward their customers. Toward their employees there has been little evidence of a sense of managerial accountability. This situation might seem to indicate that the fine idealism of Taylor and Gantt had been lost in the general emphasis on profit making. Taylor had the welfare of his fellowman at heart, and spent much of his life's energy in trying to establish mutually satisfactory relations between employees and employers. Gantt challenged industrial leaders with the statement that the life of the community depends upon the service it gets, and that the community needs service first, regardless of who gets the profits.

The statements of two industrial leaders that now follow show that the motive of service is still recognized, although applications may seem to be few.

Gerard Swope (1925):[5] "The reason for the existence of industrial organizations is not first and primarily for profit, but to furnish the community with something the community desires to have. If an organization furnishes something that the community wants, of good quality and at a fair price, it will always be rewarded for that service by an adequate profit."

Benjamin A. Franklin (1926):[6] "Every industrial executive and every owner in an industrial enterprise, whether in whole or in part, strives for and desires a profit. This is the essential reason for ownership. But ownership is no longer, if it ever was, a warrant for profit, for profit must come from the public and from nowhere else. The public is only willing to pay a profit for service, and, with the wider spread of industry, for the best service; therefore, essentially and fundamentally, industry is a service."

No facts can be presented to show how far the ideal of essential service is shaping industrial policy. But the feeling of responsibility toward employees is manifesting itself in operating procedures. The unemployment payment plan of the General Electric Company (1930) and the Unemployment Allowance Act of the State of Wisconsin (1932) recognize such responsibility. Numerous bills to set up unemployment insurance may be expected to be introduced at the 1933 sessions of state legislatures. It will be surprising if some do not become law. The sense of

[2] Recent Economic Changes, vol. 2, p. 545.

[3] "Principles and Practice of Scientific Management," World Social Economic Congress, 1931.

[4] Progress Report [1932] of Committee on the Relation of Consumption, Production, and Distribution of the American Engineering Council.

[5] From an address before the Illinois Manufacturers' Association, December 9, 1925.

[6] "The Industrial Executive," p. 100.

responsibility of industrial executives toward their workers seems to be increasing.

Change in Mental Attitude: Adoption of Scientific Method. The change in mental attitude which was noted 20 years ago is still going on. It is a shift to the scientific method. The adopted viewpoint is one that seeks for exact knowledge, and shapes action upon discovered facts. As a doctrine of management it has had great prominence during the past decade. But no one has presented it more forcefully than Gantt: "We have no right morally to decide as a matter of opinion that which can be determined as a matter of fact."[7]

This doctrine has had an influence in developing management procedures. Examples are forecasting, budget planning, and standard cost systems. Management reports, the result of studies and investigations, have been numerously produced, both by organizations and individuals. A few indicate their nature. A complete list of such studies is beyond the scope of this paper.

American Engineering Council:

Safety and Production (1927)
Industrial Sections of "Recent Economic Changes" (1928)

Management Division, American Society of Mechanical Engineers:

Economic Life of Equipment (1931)
Measurement of Management (in process)

National Association of Cost Accountants:

Cost Reports for Executives (1928)
How to Set Standards (1931)

National Industrial Conference Board:

Supplemental Bonuses (1927)
Industrial Standardization (1929)
Budgetary Control in Manufacturing Industry (1931)

Industrial Relations Counselors, Inc.:

Vacations for Industrial Workers (1927)
Pensions for Industrial Workers (1928–1929)
Unemployment Benefits in the United States (1930)

A. G. Anderson:

Industrial Fatigue (1931)

Charles W. Lytle:

Collegiate Courses for Management (1932)

Economics in Industry. Studies of economic problems in industry with the purpose of resolving into mathematical formulas relations formerly expressed qualitatively in words have made substantial progress. The efforts have been directed principally toward the establishment of rates and optimum levels. The mathematical expressions derived are simple in themselves, but because of the large number of variables involved, and unstandardized economic and accounting practices, which provide many of the numerical quantities, the symbolization adopted has been complex. Simpler expressions must follow before general application of these economic formulas will be possible in industrial practice. Generous appreciation is due to those who have pioneered in this field, and who are thereby assisting in laying the foundation for a new economics. In this group those listed in the adjoining column deserve mention, most of them members of this Society.

Investigators	Economic problems studied
Kimball	Minimum cost point
Davis, Lehoczky, Pennington, Raymond	Economic manufacturing lot sizes
Coes, Hagemann, Kurtz, Norton, Raymond, Shepard, Vorlander	Economic life of equipment
Roe	Economics of small tools
Davis	Economic purchase quantities
Shewhart	Economic control of quality of manufactured product
Knoeppel, Rautenstrauch	Break-even point

Management Principles. Continuing attention has been given to the fundamentals of management. Two codifications of principles and laws have appeared during the past decade. These contributions have important differences when compared with their predecessors. They recognize a comparatively large body of fundamentals rather than a few; they have been stated to connect cause and effect. Three principles have had their first formulation during this period: the Law of Simultation, by Blanchard; the Law of Motion-Time, by Segur; the Law of Operating Rates, by Alford and Hannum. None of these principles as now formulated nor the codes into which they have been combined can be considered as final. Statements of fundamentals need to be brought under review and revision at recurring intervals to keep them abreast of the advances in theory and practice.

Advance in Doing Work: Transfer of Skill. The most striking single industrial advancement in this decade is the increase in the productivity of the individual industrial worker. From 1919 to 1927 this increase, as represented in the physical volume of product, was about 50 per cent. The corresponding increase in utilization of primary horsepower per worker was some 30 per cent. A recent study made by Hannum and the author indicates that in well-managed concerns the increase in the productivity of the worker is continuous, is not affected by business depressions, and is of the order of 4 per cent per annum. These increases are evidence of the expanding mechanization of industry, the widening use of machines and tools, and the extension of the application of the principle of transfer of skill.

The number of references in the Bibliography to Production and Production Control is 16 times more for the decade 1923–1932 than for the 35 years preceding 1923. The comparative figures are 52 to 875; in the latter figure the totals for 1931 and 1932 are extrapolated. This bursting increase in the literature on how to do work suggests the advancement in productivity—namely, the further application of science to the work of manufacturing. Wesley C. Mitchell[8] in answering the question, "What is the prime factor in prosperity?" says: "Since 1921 Americans have applied intelligence to the day's work more effectively than ever before. . . . The old process of putting science into industry has been followed more intensively than before; . . . ."

The Human Factor in Industry: Employment Management. Taking the recurrence of the items in the Bibliography as an indication of interest and importance, labor and personnel matters, the human factor in industry, rank with production and production control. The number of items for the decade 1923–1932 is 871. No one has more succinctly presented the development of man management in industry than Henry S. Dennison.[9] At first " . . . men were called to personnel management who were new to the problem; . . . . Their experience was academic but their outlook was fresh, their sympathies and earnestness were high, many of them had keen and well-trained minds, and they knew enough to begin at once to learn from each other by

[7] "Industrial Leadership," pp. 88–89.

[8] "Recent Economic Changes," vol. 2, p. 862.

[9] Ibid., p. 518.

pooling their experiences. They overcentralized and overelaborated the job, as is always likely at the installation of a new functional department; and in this case all the more so because of the intense pressures.

"With the business stagnation and glutted labor market of 1921, they were virtually swept away, but they left behind them new practices, and old ones modified, psychological and trade tests, job analyses and classifications, rating scales, systematic training, and shop committees. What is of more importance is that they left a habit in the business mind of considering personnel management as a difficult, distinct, and major function of business management. Previously such little selection, training, health and safety work, insurance and social contacts, and joint relationships as had been attempted, had often been administered with no coordination, . . . seldom as serious projects of human engineering."

The process of sweeping away personnel activities, which was so widespread during the depression of 1920–1921, has recurred during the present business recession but in a mitigated form. The situation indicated by the Progress Report of 1931 has not materially changed down to the present time.

"Modern labor management as a whole has by no means been abandoned. Most companies have maintained their industrial-relations programs, and some have even extended and liberalized their labor policies. Unemployment of personnel directors and their associates has probably not been greater than that of other supervisory and staff employees.

"This situation, which is strikingly different from that which characterized the depression of 1921, is to be accounted for largely by the growing usefulness of modern labor management, and by the increased recognition of its usefulness by the leaders of industrial enterprises.

"This gain in usefulness has been due in part to the gradual acceptance of the theory that personnel management is properly a staff or advisory function in the industrial organization, and in part to the rising professional status of industrial-relations directors."

Management Methods: Organization. Three developments in organization have commanded the attention of industrial executives and management engineers: the operating structure itself, the functionalization of duties and specialization of the personnel, the measurement of operating performance.

The depression of 1921 with its disturbance of business relationships gave a new importance to the internal structure of industry and business. Examinations of the principles of organization and the mechanisms whereby they are made effective and are coordinated have been made afresh. One result is the placing of more reliance upon the personnel of an organization, and less upon the static design of that organization as indicated by the regulation organization chart.

Functionalization of types of activity has spread and with it the specialization of individuals. Managers have recognized that to take full advantage of the national aptitudes and exceptional abilities of executives, problems and duties must be narrowed in scope and submitted to special consideration and action. Apart from the Taylor installations, the most significant applications of functionalization have been made during the decade just ending.

Once the structure of an organization has been perfected there arises the need for measuring its operating performance. Numerous studies, but little real progress, have been made in this field. An extensive, though uncoordinated, group of ratios, turnovers, rates, efficiencies, engineering and experiential factors has grown up, the work of many investigators. It is reasonable to expect that a satisfactory mechanism of measurement must be a part of a system of control.

Shewhart,[10] in establishing a mathematical basis for quality control by random sampling, has shown the application of scientific theories to the solution of every-day management problems. But he has done more. He implies that for every function of management, standards of performance can be established in accordance with the normal distribution of each characteristic of the function when influenced solely by a constant system of chance causes.

Three widely used mechanisms of control employ standards as an integral element in their procedure. The Gantt chart plots requirement against actual performance for the same interval of time, a day, week, or month; budgets estimate expenditures for a year, for example, and divide the amounts into weekly or monthly subdivisions; standard costs compare the actual costs with a predetermined standard. In each of these mechanisms means are provided to set a standard performance, to compare the actual performance against it at specified intervals, and to analyze the causes of failures and variations.

The most recently devised mechanism of this nature is based on a somewhat different theory. It is the system of kmh-rates of Hannum and the author, and is based on the principle of operating rates which is stated as: Operating performance is controlled most directly through control of the rates of expenditure of labor, materials, and expense.

In the comparison and harmonizing of actual performance with present standards, or in the control of the rates of input of production factors in industrial operation, may lie a means for the direct measurement of management.

Management Methods: Cost Accounting. The most significant development in management procedure that has taken place during the past decade is in cost accounting. The major credit for this progress is due the National Association of Cost Accountants and that organization's able leaders. This society was founded at the end of 1919, thus the major part of its work is included in the decade under review. Three of its contributions are particularly noteworthy:

1 Raising the status of cost accounting
2 Promoting the use of budgets and developing budgetary procedure
3 Extending the use of standard costs.

Budgets made extraordinary progress following the publication in 1922 of "Budgetary Control" by J. O. McKinsey. No other mechanism of management of similar scope and complexity has ever been introduced so rapidly. A 1931 publication of the National Industrial Conference Board analyzes 162 budget systems. It is estimated that 80 per cent of the budgets installed in industry have been put in since 1922. The depression years 1930–1932 have created situations which have made budgeting difficult in many instances, and have shown the necessity of introducing adjustable means to make the estimates correspond with the possibilities of changing conditions. It is likely that the budget as a master plan will emerge from these troubled years a more useful and effective management mechanism.

Standard cost procedure is second only to budgeting in its growth in cost-accounting practice. Although the first presentation of its methods by G. Charter Harrison was made in 1918, the literature of the art and extension of the methods have come principally since 1922. Standard costs have now been applied in nearly every kind of industry, both intermittent and continuous. The concept behind the development is scientific, the prediction in advance of what the cost of product is to be.

Management Methods: Waste Elimination. Three developments in the economy of industrial operation which have had a rise to a position of prime importance since 1923 are: the elimina-

[10] "Economic Control of Quality of Manufactured Product."

tion of waste in industry, simplification of manufactured products, and engineering standardization. The "Waste in Industry" report of the American Engineering Council was issued in 1921. From the impetus given by its presentation and recommendations have come the movements for waste elimination and simplification. In engineering-society circles no other management topic has received more attention, been the subject of more meetings, and the object of more special effort than that of waste elimination. The concept of preventing or avoiding wastage, a distinct contribution to national economy, has been accepted far beyond engineering and management circles, and has become a part of the directive thinking of many non-technical individuals and groups. President Hoover, when Secretary of Commerce, said of the Waste in Industry report, that if the American Engineering Council produced only one such engineering economic document in fifty years, the existence of the organization would be justified.

Simplified-practice recommendations have eliminated unnecessary variety in sizes, dimensions, models, and types of over 130 commodities. About 20 new projects are constantly before the Bureau of Simplified Practice, which is the coordinating agency for the developments and promulgation of the recommendations.

Some elimination-of-waste surveys in manufacturing plants have indicated a possible saving of $50 per employee per annum.[11] " . . . the savings of American manufacturers from simplified practice and the application of the recommendations of the report on Waste in Industry [is estimated] as upward of $600,000,000 annually." On the basis of this estimate the total saving for the decade 1923–1932 is of the order of five billion dollars.

Standardization while well established before 1923 has made substantial progress since that date. The work has been participated in by individual plants, trade associations, technical societies, and governmental agencies. The movement to create "American Standards" was initiated by a group of technical societies which founded the American Engineering Standards Committee. In 1928 this committee was reorganized to include a broader industrial scope, and the name changed to the American Standards Association. It is now a federation of some 40 national technical societies, trade associations, and Federal Government departments, with numerous affiliations abroad. Up to January 1, 1931, it had approved 181 American Standards, of which only 10 antedate 1923. The economic advantage of standardization cannot be evaluated. The automobile industry claims a saving of $750,000,000 per year. A survey made in 1926 showed that 91 trade associations and technical societies were expending $2,600,000 annually on standardization activities. At the same time the yearly expenditure of the Federal Government was $4,250,500.

Management Methods: Wage-Payment and Profit-Sharing Plans. Financial-incentive wage-payment plans have made steady progress in the extent of application. C. W. Lytle gives a comparison of the distribution of such incentives and day work for industrial workers: "A rough idea of progress can be gained by comparing data from the Gemmill[12] survey of 1922 with those from the Jucius[13] survey of 1932. The former was from representative plants in and east of Chicago, while the latter was from representative plants in Chicago. Both surveys include all sizes of plants and all types of product. Despite differences in scale and location, the figures are probably representative of the progress in the better-managed plants during the last decade." The comparisons are:

	Number of employees		Per cent of whole	
	1922	1932	1922	1932
On time wages	15,630	44,674	53.5	29.4
On financial incentives	13,600	107,489	46.5	70.6

The invention of wage-payment plans has continued. A listing of 1924 recorded 17; a similar compilation of 1932 showed 32. It is becoming more generally recognized that many of these are but variations of more basic plans that are well known.

Five of the annual Progress Reports of the Management Division comment on incentives. From them excerpts have been selected to show progressive changes.

1926: "There has been a revival of interest in this subject. . . . Group bonus has become common, especially in the automobile industry. . . . The prejudice which formerly existed against piece work has lessened as such rates have become guaranteed and employees are learning that piece rate gives them *all* of the wages saved from decreased time."

1927: "Interest in incentives seems to be gaining . . . The cutting of piece rates or commission rates has become uncommon, and the response to such incentives is therefore more wholehearted than ever before."

1929: "For some time a tendency has been developing to make more and more group applications of incentive plans. This is a natural accompaniment of mass production in that more and more work is interdependent. As in all new things, there is, however, a danger of this change swinging too far. Since task standards need only be set for the whole assembly and the subordinate parts merely balanced, it is an easy way to avoid some of the preliminary job standardization and much of the paper work of operating. If equal results were possible, there could be no objection, but it is well established that group effort is neither that of the least efficient nor that of the most efficient, but a mean between."

1930: "There has been a revival of interest in financial incentives. This is undoubtedly due to the fact that the high-wage principle has come to the front, and also somewhat to the mature stage which job standardization has now attained."

1931: "Incentives have continued to spread into lines of work formerly considered too difficult of job standardization."

Incentives for executives, fostered by prosperity, were freely introduced up to 1929, and as freely dropped after the beginning of business depression. The Progress Report of 1929 makes this comment: "Here the incentive is coming from two directions; first, from profits in the form of a fund and distributed according to the estimated contribution of each executive; second, from measurable achievements such as departmental standards of quantity, quality, delivery, and the all-inclusive costs. . . . the type of incentive based on measured results is infinitely superior for all minor executives, and usually so up to the general manager."

Profit-sharing plans have had a negligible record during the decade under review. There is no reference to them either in the Report on Recent Economic Changes or in the annual Progress Reports.

Management Methods: Job Standardization. The high wages that prevailed during the first seven years of the decade just closing stimulated the study of operations and jobs with the purpose of reducing labor costs. Much consulting work has been done in this field. The controversy over the relative merits and possibilities of time study and motion study, which was alive in 1923, has subsided. The harmonizing of these methods of job standardization by Roe and Lytle in "Management's Handbook" has been an important influence in bringing about the change of attitude. In addition to the extension of the utilization of time and motion study, important developments are improvements in the technique and devices for time and motion study, and in motion-time analysis. Segur has based the technique of the latter on the principle of motion time.

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[11] "Recent Economic Changes," vol. 1, p. 119.

[12] P. F. Gemmill, "A Survey of Wage Systems," *Industrial Management*, October, 1922.

[13] M. J. Jucius, "The Use of Wage Incentives," *Journal of Business*, University of Chicago, January, 1932.

Three of the annual Progress Reports contain these comments:

1928: "There has been a pronounced tendency toward a better understanding of time study."

1930: "The application of time and motion study is now spreading in the textile industry, especially in the South. . . . Some of the most progressive labor organizations have begun to insist upon the establishment of standards in their operations."

1931: "There is definitely a revival of interest in the micromotion method of job standardization. This has arisen from the General Electric Company's practice of using the full micromotion method to train its analysts, and of allowing the work of the analysts to be done without the full use of the method. . . . when analysts are trained by the full micromotion method they are so familiar with the principles of correct motions that they are able to make drastic improvements without taking elaborate measurements of existing work methods."

The Society of Industrial Engineers has completed a survey of time-study devices and methods, but the report has not been published.

Management Methods: Inventory Control. Inventory control has been refined and made more effective during the decade of this paper. Concentration upon methods for its accomplishment was stimulated by the shortage of materials during the World War and the excess of stocks during the depression of 1921. Control is sought to hold quantities within prescribed limits, to reduce the risks of damage and obsolescence, and to save the wastes of handling. "One of the dramatic developments . . . is to be found in some automobile plants, where the average inventory has actually decreased from several months' to a three or four days' supply. . . . materials . . . are planned to be taken directly from the freight cars and placed in the manufacturing departments where they are to be used, thus eliminating handling and the expense of stores control."[14]

Management Methods: Materials Handling. An indication of the creative energy directed toward better handling of materials is given by the number of references in the Bibliography before and after 1923. The record is: before, 8; after (1923–1932 adjusted) 251. Improvement has been in the economics and control of handling, in the equipment available, and in organization of material-handling departments. Economies in many instances have been substantial. Thirty-seven installations reported in 1928 had aggregate annual savings of over $850,000. Under this general head is the extension of assembly equipment to allow "straight line" production. Numerous installations in many different kinds of industry show the wide possibilities of applying this principle.

Management Methods: Accident Prevention. The seriousness of the industrial-accident problem began to be appreciated in the United States some 12 years prior to 1923. Workmen's compensation laws were passed, the safety movement initiated, and thousands of persons began to give their attention to ways and means to reduce accidents. The effect of this work was significant. The prevailing number of accidents was cut in half, or to a third, in many industrial establishments. But following the revival of business in 1923 an alarming situation became evident, brought about by the increasing frequency and severity of compensatable accidents and mounting accident costs. The real situation was at first masked by the great increases in production and productivity. While there had been an increase in the hazard of industry per man-hour, production had increased so much more that the hazard in terms of production had decreased.

Following 1923, therefore, the accident problem became this: Can accidents be controlled under modern industrial conditions of high productivity? The National Bureau of Casualty and Surety Underwriters asked the American Engineering Council to investigate this matter. The report of 1928 on Safety and Production presented the results of the study. In brief the findings are:

1 Industrial accidents can be controlled under modern conditions of highly efficient production
2 Safety and production are shown to have a coefficient of correlation of 0.835; "a 'safe' factory is eleven times more likely to be 'productive' than an 'unsafe' factory"
3 Major industrial executives have as much responsibility to initiate accident prevention as to initiate improvement in productivity.

The situation has changed but little since the issuance of this report. The industrial-accident problem is still of major importance. But the attitude toward the question has improved, and the means for its progressive solution are more clearly defined.

Management Methods: Works Councils. Employee-representation plans, or works councils, spread rapidly after the recovery of business following 1922. By 1923 some 700,000 workers were included in these plans. By 1928 this number had doubled. In the management survey of 1928 for the Report on Recent Economic Changes, 20 per cent of the plants studied had active, formal employee-representation plans, 10 per cent dealt occasionally with especially elected employee committees, and 6 per cent were working under union agreements. One of the consequences of the rise of these plans has been the retarding of the growth of organized labor. What the effect of the present depression will be ultimately on works councils cannot be foreseen. It may not be so far reaching as that experienced in 1921–1922. "The year 1921 saw the discontinuance of many arrangements for joint relations, both union agreements and independent employee-representation plans. The serious unemployment at that time and the weakened position of the unions resulted in the abrogation of many arrangements made during the war and boom years. Many plants closed, and when they reopened the employee committees which had previously existed were not revived."[15]

Wage Levels. No other theory of management has ever had such a meteoric rise of acceptance and seemingly an equally meteoric drop in favor as the doctrine of high wages. The theory that high wages accompany high production was practically unknown in 1921, although it had been suggested by a few management engineers. In that year "more than 300 articles appeared telling of methods used in cutting wages and speculating on how far they would fall. By 1922 articles of this sort had disappeared and those about wage incentives had taken their place."[16] By 1926 the doctrine was well established and had been accepted by many able business executives and trade-union leaders. The high wages of these years down to 1930 were an impelling force in the management progress that this paper records. As the depression of 1929 began to be felt, the high-wage doctrine was invoked to maintain wage rates, with considerable success through 1930. The Progress Report of that year states: "The principle of high wages is standing the test of the depression better than many expected. . . . Relatively few of the leading large companies have cut wages per piece or per hour. . . . The very fact that so many leaders are deliberately trying not to cut wages is enough to vindicate the statement that the high-wage theory has come to stay in this country." However, during 1931 and 1932 wage cuts became common.

A study made by the National Industrial Conference Board of the record of 1718 industrial concerns employing 3,358,666 workers in 1929, and 2,391,009 in 1932, yielded these facts:

[14] "Recent Economic Changes," vol. 2, p. 509.

[15] "Recent Economic Changes," vol. 2, p. 529.

[16] Ibid., p. 529.

Nature of reductions	Per cent of companies
Executive salaries	80.5
Other salaries	81.0
Wage rates	75.4

Extent of reductions	Average reduction in per cent
Executive salaries	20.3
Other salaries	15.9
Wage rates	13.9

In general, salary reductions have preceded wage-rate reductions. In 1930, 212 of the companies reporting reduced salaries; 148 reduced wage rates; about 12 per cent of the reported reductions in salaries and 11 per cent of the reported reductions in wage rates were made during this year. More reductions occurred in 1931. The peak seems to have been reached early in 1932.

The ultimate fate of this doctrine cannot be predicted. It has not been abandoned. It may return to general favor and acceptance with the modified meaning that high wages can be paid only for high productivity. The Progress Report of 1931 gives an excellent summary of opposed points of view: "A veritable storm has raged between the orthodox economists, including many of our bankers, and the progressive manufacturers, including most of the engineers. The former group believes that the sooner wage rates are reduced to the same extent that commodity prices have been reduced, the sooner will be the recovery of industry. The latter group, while conceding that a proportionate reduction in money wages need not affect adversely the economic status of the worker, maintains that the real strength of American business lies not in the maintenance of an even balance between wages and cost of living, a continuing equilibrium from year to year, but in a steady advance toward the higher standards."

Working Hours. In 1926 the Ford Motor Company startled industrial executives by announcing the adoption of a five-day week. The move was a natural result of the higher productivity per machine and per man-hour which had been attained. A report of the National Conference Board published in 1931 showed that 270 establishments employing in the aggregate 216,921 wage earners were operating on a year-round five-day-week schedule at the close of 1928. The annual Progress Report of 1930 carries this item: "While a few companies have lengthened the working day in order to pay the same daily wage, in general hours have been shortened and the five-day week has been given greater trial than ever before. In 48 national and international unions there are 532,894 employees on this basis." Continued agitation for shorter working hours and a flexible working week is active, and the practices have many advocates. "Shorter normal working hours would constitute a permanent spread of unemployment. . . . For instance, it is reasonable to believe that if in an industry normally operating 50 hours per week, the working hours were shortened to 40, with 50 per cent overtime pay up to 45 hours, and 100 per cent overtime pay beyond that, the influence of the change would be in the direction of maintaining good business . . . ."[17] The trend toward shorter working hours is one of the most pronounced in industrial management today.

Working Conditions. Increasing attention has been given to working surroundings and conditions during the past decade. Evidence of this fact is offered by the following tabulation of items in the Bibliography:

Item	References before 1923	References 1923–1930
Air conditioning	0	60
Artificial lighting	2	80
Fatigue	7	30
Fire prevention	0	26
Noise elimination	0	6

Ventilation and air conditioning have been utilized to remove dust and fumes and to control temperature and humidity that working spaces may be more comfortable and better adapted to manufacturing processes. Artificial lighting has tended to use brighter and higher unit intensities and is credited with substantial increases in productivity. The situation in regard to the elimination of fatigue was characterized in the Progress Report of 1926 in these words: "This matter has had more propaganda than research. . . . there has been practically no advance in our knowledge of the nature of fatigue and the part it plays in efficient production." Since that statement was written fatigue has had intensive study with a reversal, or substantial modification, of former common beliefs. Anderson's investigation (1930–1931) was under actual working surroundings and conditions. He shows that with good management fatigue is not a major consideration: "Progressive, modern manufacturing methods and management are rapidly eliminating real fatigue from industry. At the present time satisfactory outputs at low cost may be had without unduly timing workers, and in industrial work can be, and probably will be, of little importance." Fire prevention and protection in industrial plants has kept pace with the improvement in maintenance. Noise elimination is just beginning to receive attention, with as yet no tangible results to record.

Industrial Plants. Two possible tendencies in the locating and planning of industrial plants are worthy of attention, although neither is strongly evident at this time. The first is toward the locating of industries in small centers; the second is toward selecting the small unit as most economical in operating performance. In regard to the one, the American Engineering Council says:[18] "The decentralization of manufacture which is engaging the attention of industrial leaders offers a useful method of striking a balance between agriculture and manufacturing. While a few industries, like the manufacture of steel, must be organized on a scale so large as to surround themselves inevitably with urban conditions, this is not true of the greater number. They can properly be divided into smaller units. The present tendency is to locate units in smaller communities in closer contact with rural conditions." In regard to the other, recent studies strongly indicate that the optimum size of plant for effective operating performance is much smaller than is commonly believed. On the operating side, plant maintenance has assumed new importance because of its influence to increase production when bad-order machines are avoided and its effect in arresting depreciation.

Obsolescence of Equipment. Management has begun to deal with the obsolescence of equipment from two points of view. The one is that far too much machinery and other physical property is in use, or held for use, which actually is obsolete and should be disposed of. The other is that, due to the depression, actual values of previously acquired plants are lowered and assets should be written down accordingly. Two comments appear in the annual Progress Reports:

1928: "The replacement of obsolete equipment is considered to be one of the principal factors in good management."

1929: "Probably the most important questions that confront the manufacturer today are the relation between the condition of equipment and profits and the replacement of machinery which in point of service may have many years of usefulness but which is incapable of satisfying the latest fancies of the consumer market or competing with more recent and improved machinery. . . . Experience is showing in these competitive days the mathematical relation between obsolete machinery and profits is an inverse ratio. . . . The important phases of the industrial-equipment studies are: (1) to bring out facts showing the relation of equipment obsolescence to production costs, and (2) to dis-

[17] Progress Report of Committee on the Relation of Consumption, Production, and Distribution of the American Engineering Council.

[18] Progress Report of Committee on the Relation of Consumption, Production, and Distribution.

cover, if possible, a means for correctly evaluating and providing for obsolescence in cost accounting."

Industrial Research. Industrial research has as fully demonstrated its value during the months of business depression as any other management function. In spite of bad conditions, expenditures for research in well-managed plants increased in 1931 over 1929, laboratory efficiency has improved, and research has aided in these definite ways: by reducing costs, improving products, developing new products, increasing customers' goodwill, and in some instances is credited largely with the earning of profits.

The demands of the war gave a great impetus to research through the physical sciences. A report of the National Research Council of 1920 lists 300 organizations for research. Included are company, joint, consulting, commercial, trade association, and university laboratories. A further report in 1927 lists 999. Rough estimates of expenditures for basic production research as of 1928 ranged upward to $200,000,000 annually. Since 1921 business procedures and economic forces have been added to chemical and physical research. Numerous references in this paper are evidence of the work already done in these newer fields. Methods for statistical research and market analysis are examples.

An assay made by the National Research Council in 1928 gave the major emphasis on items of industrial-research programs in this order of frequency:

1 Improved product or service
2 Reduction of production costs
3 Development of new fields of application
4 By-products and new materials
5 New products.

A similar, though less extensive, assay in 1932 gives indication of the effect of the depression.

Expenditures for research in percentage of establishments, comparing 1931 with 1929:

Increased	75
No change	15
Decreased	10

Laboratory efficiency, comparing 1931 with 1929, expressed in percentage:

Increased	87
No change	10
Decreased	3

Emphasis in research programs in 1931 expressed as percentage of frequency of appearance:

New products	38
Improved quality	37
Reduction in production costs	19
Development of new fields of application	5
By-products	1

Industrial-Engineering Courses. The major change in industrial-engineering courses of engineering grade has been the increase in number and in enrolments. The number of institutions giving this work has increased from 10 in 1922 to 35 in 1932. The number of graduates in 1922 was 311, and the number of graduates in 1931 from 30 institutions was 596.

In 1923 the Society for the Promotion of Engineering Education undertook a comprehensive study directed to the improvement of engineering education. The investigation was completed in 1929. In the fact-finding stage of this study engineering-college graduates were asked what subjects they regretted missing in college. Seven of these and the number of men mentioning them follow:

Business administration	456
Economics	226
Accounting	179
Psychology	122
Salesmanship and advertising	118
Industrial management	104
Production and production methods	33

This record is an indication that graduate engineers feel the need of courses such as are included in industrial-engineering curricula. An indication that engineering educators are alive to the situation is found in the fact that industrial-management subjects occupied a prominent place on the program of the summer schools for engineering professors, held under the auspices of the Society for the Promotion of Engineering Education at Purdue University in 1930 and at Stevens Institute of Technology in 1932.

Lytle[19] has summed up the major purposes of industrial-engineering courses as:

1 To formulate the principles of an emerging profession
2 To prepare more adequately many men who are to enter this profession.

The situation seems favorable for major advances in the teaching of industrial engineering in the years immediately to come.

Marketing. Among the new problems that assumed importance during the decade 1923–1932, none has led on to more discussion than management of marketing. One-fourth of the section on management in the Report on Recent Economic Changes is devoted to this topic. In the Bibliography there are seven items on sales management up to 1923. From 1923 through 1930 the total is 181. Marketing is commented on in each of the annual Progress Reports beginning with 1926. The Taylor Society has been the leader in developing the science of marketing and in applying scientific-management principles to selling. The American Management Association has a division devoted to distribution problems and has made marketing practice a major topic in its programs. However, the progress made must be judged as slight. Marketing technique is crude both in its planning and execution. A few tendencies have asserted themselves: functionalization of selling activities similar to that which has taken place in manufacturing; adoption of the sales budget; accounting for sales expenditures; scientific pricing of product; analysis of markets; setting sales quotas.

The year-to-year comments in the Progress Reports show something of the developments of the attack on the problems involved:

1926: ". . . these principles [of management] are extending beyond the old engineering type of industries and are becoming popular in industries less accustomed to engineering methods. Offices and retail stores are resorting to them. . . ."

1927: "The trend is toward intensive rather than extensive marketing. . . . Part of the advertising budget is being taken for research to ascertain what the consumer really wants."

1928: "There is a growing interest in the cost of marketing, and more specifically in the cost of marketing activities."

1929: "Relatively little progress has yet been made in analyzing distribution costs from the standpoint of the cost of handling individual items or performing individual services. . . . Investigations in wholesale and retail distribution for several lines of trade, with particular reference to the small as well as the

[19] "Collegiate Courses for Management," International Management Congress, Amsterdam, 1932, and *Engineering Education*, (S.P.E.E.), vol. 39.

larger producer and also to the several methods of distribution, show the possibilities for reducing costs through more uniform methods."

1930: "Experiments in distribution have continued and some progress has perhaps been made, but it has not been sufficient to allow much specific discussion. . . . There has been increased appreciation of the fact that as a rule it is necessary to market those commodities which the consumer wants or can readily be brought to want . . . industry is coming to realize that its raw materials are the needs and wants of persons."

1931: "The field of marketing is now in the greatest need of sound common sense and engineering skill. . . . The demand for useful products cannot be forced into being. . . . It has been recently recognized, both in private investigations and by governmental analysis, that it is very nearly universal in industry today to find that approximately 80 per cent of the volume of production exists on approximately 20 per cent of the number of products in the line. . . ."

Association and Society Activities. The open-minded interchange of trade and technical information has been a factor of importance in the progress of the immediate past, and promises to occupy an even more important position in the business practice of the future. "What is raising the whole standard of management in the United States is the habit among employers of discussing their problems openly among themselves, of comparing the methods of one industry with those of another, and of founding associations for research and conference. . . . They are not afraid to teach each other, nor too proud to learn from each other."[20]

In 1926 the Department of Commerce listed nearly 9000 business associations; in 1928 the number was 13,000; by 1930 there were 19,000. Some are local in scope, others state-wide or interstate, still others national or international. Their activities range "over almost the whole field of management, including especially statistical service, cost accounting, industrial and commercial research, simplification, standardization, credit information, traffic and transportation studies, and trade ethics. It is significant that several hundred business executives are members of some national association of the social sciences."[21]

The membership in management societies has likewise increased during the past decade, the influence of the organizations has widened and strengthened, and the reports and papers sponsored by them have greatly enriched management literature. Among themselves the societies have entered into close cooperation and have participated in numerous joint activities. Management Week meetings in the United States and Management Congresses abroad are examples. Management Week was instituted in 1922 to promote interest in the function of management and to further disseminate information on its principles and practice. During the week designated in 1926 meetings were held in over 200 cities. From that high point interest waned until the plan for a week of organized activities was abandoned in 1929. Five International Management Congresses held in Europe have been participated in through the American committee representing jointly The American Society of Mechanical Engineers, the Society of Industrial Engineers, the American Management Association, the Taylor Society, and the National Association of Cost Accountants. The meetings were held in Prague (1924), Brussels, Rome, Paris, and Amsterdam (1932). Fourteen papers were contributed to the Amsterdam meeting. In addition there were several American papers presented at the Management Session of the World Engineering Congress held in Tokyo in 1930. Still further, American support has been active in the establishment and maintenance of the International Management Institute, with headquarters in Geneva. These contacts in foreign countries have served to show, with new emphasis, that management is universal.

[20] "American Industry and Its Significance," *Round Table* (London), 1926, vol. 17, p. 264.

[21] "Recent Economic Changes," vol. 2, p. 497.

Discussion

WALLACE CLARK.[22] The Alford progress reports are of great value, not only as authoritative records of the history and practice of management, but as a source of inspiration to engineers and all others working in that field. The writer is impressed by the breadth of this report and by the certainty it gives that the influence of management on human life is greater than ever before and more widely understood.

It is good to hear that the service idea in industry has survived the grave difficulties of the last few years. All honor should be given to those companies that have made efforts to keep their workers and to spread their work. The most encouraging statement is that some real progress is being made toward national planning, for this seems to be the best protection against the recurrence of another serious depression. Even then, our national planning must be coordinated with that of other countries or it cannot be wholly effective.

DEXTER S. KIMBALL.[23] Mr. Alford again has presented in concise and clear form the progress of industrial management, and this paper is a most welcome addition to those that he has already written marking the progress of industry. The paper is well documented, indicating that as usual what has been written bears the marks of scholarly research and is not merely a passing opinion. Several of his findings are of more than usual interest.

First, he calls attention to the changed viewpoint of all manner of people at the close of this decade as concerns the ultimate good of the vast industrial machine we engineers have built up. Dismay and doubt have succeeded admiration and applause. The problem of distribution has transcended that of production, and as the author points out, this in itself challenges the engineer and the industrialist to so direct management that it will "lead on to a higher professional standing through the development of intellectual, social, and moral attitudes and values." The point is well taken and should command the attention of every engineer and industrial leader.

The next important impression gained from the paper is his findings to the effect that in spite of this industrial debacle the principles and practices of scientific management, so called, appear to be firmly established and have apparently suffered no loss of prestige. He notes rather that progress along these lines has been constant even during the last few years, and he concludes that "the change in mental attitude which was noted 20 years ago is still going on. It is a shift to the scientific method."

This is highly important when one considers the new ideas of industrial planning which he discusses and which have appeared for the first time in our literature during the past year. Mr. Alford notes quite correctly that industrial planning in an effort to make production and distribution balance is a natural outgrowth of Taylor's ideas of shop planning which are now so familiar to all industrialists. He quite properly dismisses international stabilization through planning as out of the question at present, but the writer agrees with him that we shall, in the next decade, hear much about industrial balance or stabilization through national or private planning or both. This is perhaps

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[23] Dean of Engineering, Cornell University, Ithaca, N. Y. Mem. A.S.M.E.

the greatest and the most difficult problem that confronts us.

Mr. Alford also brings to attention as a problem of management the theories of wages which are now so much discussed. Here we touch upon difficult economic theory which we as engineers are not as yet in a position to debate with much success. Yet this and the correlated question of the length of the working day are matters with which the engineer in general and the industrial engineer in particular must concern themselves, for no doubt they are intimately connected with industrial recovery and stabilization.

ROBERT T. KENT.[24] The author has undertaken a task of no mean order when he attempts to evaluate the progress of management during a period that presents such contrasts in management problems as does the decade under review. That he has been able to trace real progress, notwithstanding the chaotic conditions that have tried the souls of management during the past three years, is a tribute not only to the author's power of analysis, but also to the soundness of the basic principles of management itself. The method that the author has adopted is unique. Nevertheless, it is probably sound, for the literature of a great movement undeniably reflects the thoughts of those who are participating in that movement. The facts and data are exhibited in the paper. The interpretation of these facts and the conclusions drawn therefrom are the author's own. It is with these interpretations and conclusions that others may disagree.

The writer does not wish to take the position of disagreeing. On the contrary, he agrees with most of the statements made by the author. He believes, however, that more stress could well have been laid upon certain points, and it is these points that the writer desires to discuss.

Under the head, "Advance in Doing Work, Transfer of Skill," the author refers to the increase in productivity of the individual industrial worker, and states that the increases in productivity are evidences of the expanding mechanization of industry, the widening use of machines and tools, etc. It would seem that an opportunity has been missed here to point out the necessity of the education of the people, as a whole, in the fundamental principles of political economy. The author has not only laid open himself, but the entire engineering profession, to the charge that he is giving evidence in support of the economic fallacy of technological unemployment. Every engineer who has studied industry, its history and its future, knows that the mechanization of industry does not permanently decrease employment, but in the long run increases it. At this point, where the author shows that the productivity of the individual worker has been increased by mechanization of industry, he also should show how this mechanization of industry has created more employment for more people in the long run, and that therefore the mechanization of industry still further is a desirable thing. This is true, notwithstanding the outcries of those who oppose the machine age, sometimes from political motives and at other times from pure ignorance.

In his section on "Machine Methods and Organization," the author speaks of the measure of operating performance. A careful reading of this section indicates that he refers more to the measurement of the operating performance of the personnel in the organization than to measurement of the performance of the management as a whole. The writer believes that the late and the present unpleasantness through which we are now passing has revealed as nothing else could do the woeful need that we have for some standard of measuring the effectiveness of management itself. This has been pointed out several times, but apparently no systematic study of this subject has ever been made. Several years ago Prof. J. W. Roe presented a paper before this Society outlining a tentative method for the measurement of management. It was offered as a starting point from which we could begin the development of a system or method or plan for the real measurement of management itself. So far as the writer has been able to discover, and he has made many earnest efforts so to do, no one has ever even attempted to apply the principles enunciated by Professor Roe, and one of the most valuable papers ever presented before this Society is an unused tool of exceptional value in a crisis such as we are now experiencing. The most important thing before management today is to learn how to evaluate itself and its own effectiveness.

Budget control is the subject of a single paragraph in the section on "Management Methods, Cost Accounting." Here again the experience through which we are passing has demonstrated that budgets are far from being the instruments of perfection that they were considered to be during the golden years that ended with 1929. This is not said in derogation of budget control. On the contrary, if it had not been for budget control, our experiences during the last three years would have seemed mild compared to what they would have been had not budget control been as well established as it was. The point is that budgeting has failed us only in the respect that we have not yet learned how to accumulate data which will enable us to make better forecasts—in other words, to make more accurate predictions of income. In a period of falling commodity prices, falling volumes of business, and all the other concomitants of a first-class business depression, those who have been in the management end of industry have seen the most carefully made budgets become worthless within a month of their completion, simply because of inability to forecast the income figures correctly.

Budgeting will not become an exact science until we have learned how to accumulate data and to interpret these data in such a manner as will enable us to forecast income as accurately as we can now forecast expense. The problem is thus simply stated. The solution of it is one of the major jobs ahead of the engineering and management professions.

Wages will always be a subject of controversy. Whether to keep wages high and maintain individual purchasing power or to cut wages and keep more people at work will always afford a theme for discussion between two different schools of economics. The author states that during 1931 and 1932 wage cuts became common. These wage cuts were made in most cases only in desperation, as the last thing that many manufacturers could do to save themselves from bankruptcy. More and more have manufacturers and leaders of industry come to the conclusion that the way to prosperity is to build up the purchasing power of the great body of industrial workers. Hundreds of manufacturers in the past three years have gone deeply into their surplus, and in some cases have wiped out their surplus, before wage cuts were put into effect. The table of percentages of reductions in wages and salaries given by the author is significant in that executive salaries were cut more deeply and in more cases than wages. This table shows very clearly that the executives in charge of industry realized not only their duty to their country, but to their employees as well, and that when punishment was necessary, they took their own punishment first.

Under the head of "Working Hours," the author calls attention to the decrease in the length of the work week or work day. While this has nothing to do with the paper except as a comment upon the last sentence—"that the trend toward shorter working hours is one of the most pronounced in industrial management today"—the writer believes that it is significant that every improvement in mechanization and in management has tended toward the shorter working day and the shorter week. It will be recollected that when the factory system first started, the 14-hour work day was common in England. The writer remembers,

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as a boy, of hearing of certain places where they still worked 12 hours a day, and in the 1890's the working week throughout industry was 60 hours. Inasmuch as consuming power can grow but slowly while productive power can be increased rapidly, it is only logical to assume that the remedy for our so-called overproduction is to produce less; that is, we should follow the trend that has continued since the beginning of the factory system and use the greater productivity to give us more leisure in which to enjoy the increased earnings that better machines and better management will give us. The writer believes that the five-day week is inevitable within the next few years. It would not surprise him to see the four-and-one-half- or even four-day week become universal within the next quarter century.

The author gives considerable attention to the improvement in working conditions. Working conditions have a direct relation to costs. Improved working conditions have not been brought about particularly by any altruistic spirit on the part of management, but by the realization that with better working conditions, increased production and consequently lower costs are possible. In this same section is mentioned the reversal of ideas concerning fatigue. Arguments pro and con in regard to fatigue are rather academic. Fatigue automatically is eliminated by better working conditions and by good management methods. The fundamentals of good management, as stated by Taylor, in which he demonstrated that fatigue is an important element in the determination of the daily task, still hold good. The fact that fatigue does not seen to play so important a part today is simply due to the fact that machines, work places, working conditions, etc., are so arranged that fatigue is not engendered in the way that it was when Taylor wrote his classic.

In discussing the location of industrial plants the author quoted from the American Engineering Council, which says that the decentralization of manufacture is engaging the attention of industrial leaders. One of the things that management will come to in the near future is extensive decentralization of industry. From his experience in plants located in large cities, and also in the largest city in the country, and in plants located in the smaller cities, the writer will unqualifiedly put himself on record that the plant in the small community is far and away better located than the one in the larger community. Not only are there better opportunities for good management, but the workers are more contented and have more of an interest in their employers' business, have more leisure in which to enjoy themselves, and therefore approach their work in a better frame of mind.

Under this same subject, the author discusses the maximum size of the plant. The writer believes that in future the trend will be toward the smaller plant, and more plants in the larger companies. During the war, the writer was closely associated with the management of a plant where were employed 4500 to 5000 men and women in diverse occupations, under a single directing head, the general manager. While this man was one of the ablest of managers, the multifarious duties involved in the management of such a large number of employees was beyond the capacity of any one individual. The writer then formed the theory that in every industry there was a definite upper limit to the size of plant that could be efficiently managed under a single head, and that when a company grew to a size larger than this, the proper procedure would be to subdivide it into two or more plants, each of which would be smaller than the economical management limit of size.

The author outlines in considerable detail the progress that has been made in industrial research. Extended comment on this is unnecessary, except to state that it might have been well to emphasize that research is the foundation stone of industry and that there cannot be too much of it.

The author gives some consideration to the subject of industrial engineering education. At the risk of being branded a heretic, the writer will put himself on record that, in his opinion, industrial engineering education is, *per se*, highly overrated. It is a mistake to train young men specifically in this branch of engineering. Engineering students should be grounded thoroughly in the fundamentals of science, in sociology, English, and good common sense. If they have this foundation, they will quickly absorb the principles of industrial engineering as soon as they get out into the world. The engineering educational schools seem to stress too much the mechanisms of management to make good industrial leaders.

The writer finds himself in complete agreement with the author's statements concerning marketing. We have much to learn on the application of the principles of management to marketing. While the fundamental principles of management are applicable to marketing and distribution, just as much as they are applicable to production, the application must be along different lines than those used in the shop. The standards must be different, and the technique must be built up. Nevertheless, if the principles of management as applied to marketing were as thoroughly understood as they are when applied to production, it is probable that much of the distress of the past three years would have been avoided.

In his treatment of association and society activities, the author does not make clear whether or not he includes in this category the trade associations. While some of these are mere price-fixing agencies, others, and the writer believes the majority, are powerful forces for the advancement and stabilization of management. The writer has had experience in several trade associations, and it is his observation that they are the one big factor that will tend to stabilize industry, prevent cut-throat price competition, lead to cooperation in the stabilization of markets, and that they will be the promoters of extensive industrial research that cannot be undertaken by individual companies. The trade association may well become to an industry what the planning or production-control department is to the factory.

CARL L. BAUSCH.[25] Another item should be included as a part of the ten years' progress report and as an essential for recovery. This is unemployment insurance. In February, 1931, the Rochester Chamber of Commerce reported that fourteen companies had adopted the unemployment benefit plan. The plan is as follows:

All employees receiving less than $50 a week are included. The benefit will be 60 per cent of the weekly earnings, with a maximum of $22.50 a week, payable after two weeks of unemployment. The benefit will be paid only to employees laid off on account of slack work.

The fund will be accumulated by the concerns laying aside up to 2 per cent of their pay-rolls for the year 1931 to 1932. No contributions will be made by employees during normal times. If after the plan is started and it seems to the management that the fund is inadequate, an emergency may be declared and a 1 per cent assessment made on salaries of those getting over $50 a week. In such cases the company will contribute an additional equal amount.

All plans looking toward business recovery must include means to bring about a feeling of confidence in the minds of labor as well as of capital.

WALTER N. POLAKOV.[26] There is no reference in the report to the influence of power production in the changing aims and

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[26] Walter N. Polakov & Co., Inc., New York, N. Y.

methods of factory management. In the machine age and prior to the appearance of machines the object of management was to make the worker personally more efficient. Training of workmen in the habits of industry, time and motion studies, premium systems, and task work with bonus were designed to that end. With the advent of the power age, machines became emancipated from the shackles of belts and gears. Driven by individual electric motors, machines are now easily relocated to suit the sequence of operations, thus upsetting all traditions concerning material handling, partly finished stores, shop dispatching, etc. Electrified machines are now frequently automatically loaded, adjusted, driven, controlled as to dimensions, unloaded, and output recorded, wholly automatically. There is a definite trend toward the "automatic factory" where no human hand touches the product throughout the operations. There are numerous skelp mills, rolling mills, aluminum plants, paper and pulp mills, cigarette factories, etc., where the transfer of skill and transfer of power from worker to machine are so complete that the functions of workers are chiefly supervisory in their nature.

This changes the labor specifications from demanding trade skill and physical force to new requirements of a psychological nature: (1) Sustained attention, (2) correct perception, (3) prompt reaction. These qualifications are becoming essential labor characteristics, since observations of instrument indications in continuous processes demand sustained attention to all such factors as pressure, temperature, humidity, concentration, speed, etc. Then, these instrument indications must be correctly perceived and interpreted, so that prompt action may be taken before any damage is done by the changing conditions. This new role which labor is now increasingly called to play eradicates the old distinction between manual and mental labor, fusing them as the machine itself is fused with a motor and electrical controlling devices. Consequently, the main function of management is the securing of an uninterrupted flow of production. This function is becoming the more important as the first cost of the automatic equipment, and hence the fixed portion of capital charges, is getting not only far greater than it was before, but it generally exceeds many times the pay-roll.

This definite aim of securing an uninterrupted flow of production largely depends upon the managerial function of planning. Within the last decade this problem of planning of production, originally evolved out of the principles laid out by Taylor and perfected by Gantt, is spreading beyond the walls of a shop or a plant and has become a grave problem of planning within an industry, within a nation, and even among the nations. Hence we observe the movement to modify the anti-trust law, to plan and control the production of oil and cotton and other commodities, to plan import-export trade and readjust the tariffs, and even to plan concerning international war debts. The management engineer who a few decades ago was confused with an "efficiency expert" is looked upon today as one peculiarly fitted by his experience in planning to give advice on major economic problems.

L. W. Wallace.[27] The paper has omitted any reference to the conference of the leading industrial and commercial men of the nation held in Washington in December, 1929; that in this conference they committed themselves not to reduce wages if it could possibly be avoided; that this action signified that the fundamental principles of good management have become embedded in the minds of our leaders; and that they recognized the importance of maintaining good personnel relations. As a result of this commitment and the sincerity with which it was followed, American labor realized that management was doing everything it could to safeguard its interests during this depression.

As a result of this, during these strenuous times there have been less labor disturbances than in normal years. This fact is a great credit to both labor and management, and will become historical. It is one of the most significant events in the realm of management during the last ten years.

In a broad sense and over a long period of years, there is no such thing as technological unemployment. Due to technological advance there has been a shift in occupations, all tending to elevate the position of industrial workers. More skill, intelligence, and judgment are required to construct and operate modern equipment than under old methods; hence industrial workers have been greatly relieved of drudgery and are receiving higher compensation.

Lillian M. Gilbreth.[28] In his 1912 report on the progress of management, the author referred to the transference of skill as of great importance, and his writings ever since have shown that he continues to feel this. It is not easy to define skill. The best that we have been able to do is to define it as "knowledge, plus dexterity, plus adaptability." Psychologists and engineers would seem to agree that knowledge and dexterity are necessary for skill; there is as yet little realization of the great importance of adaptability. Yet this is fundamental, for intensive studies of skill show that the man who possesses it can adapt to changing situations, and it is this adaptability that distinguishes the skilled man from any machine that may be invented to take over the work that he does.

If adaptability is accepted as part of the definition of skill, then skill can never be transferred from a man to a machine, though it can be transferred from a man to a man. And we need have no fear that machines will ever make skill unnecessary. Machines may embody knowledge and furnish repetitive dexterity, but cannot adapt themselves to a changing situation, as can a man.

The implications of this are many. Materials, if they are to facilitate machine operation, must be carefully standardized. If they are to conserve and develop skill, it is thinkable that not such close inspection is indicated. Machines should take over drudgery and such repetitive work as requires no adaptability. Men should be assigned to work that will require as high a degree of adaptability as they can supply, if their skill is to be conserved and developed.

In motion study, we are careful always to keep the consideration of maintaining, developing, and using skill a part of the program. The chief emphasis is on developing men, rather than on making things or inventing machines.

As a Society, we have made great progress by adding economics to the subjects discussed at meetings. We could make similar if not greater progress by adding psychology. A study of our programs during the last 20 years shows that we are including the human element more and more in our discussions of management.

But as yet we do not include the principles and practices of psychology, which concerns itself with the developing science of human behavior.

Our great need today is to realize that we can develop the human element along with our machines and our materials. This can be most appropriately done by discussing skill, its definition, its transfer, and its development, and it is to be hoped that every program of the Management Division will contain a place for such discussion.

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[28] Gilbreth, Inc., management engineers, Montclair, N. J. Mem. A.S.M.E.

G. M. EATON.[29] The writer challenges Mr. Wallace's statement that there is really no such thing as technological unemployment. As a statement of long-time average considerations, the writer agrees with Mr. Wallace on the job-creating achievements of technological advance. But we are evading the issue when we deny the more or less transient unemployment attendant on every broad technical advance.

To give one specific illustration, the interests with which the writer is connected laid off nearly 300 skilled men when, along with the rest of the steel industry, they replaced hand labor with machines for the removal of surface defects from steel blooms. Do you think you could convince those 300 men that technological unemployment is imaginary? As they tightened their belts, they would resent defining their hunger period as transient.

We have been told that the engineer must get off from the beaten trail in his economic thought and be guilty of appearing radical. Very well, let us be radical for a moment. We have been told of organized effort to train men of skill, men of knowledge, dexterity, and adaptability, so that they can apply their skill to carving a living in a new way. The writer wishes that it had been brought out that too much of this effort is now necessarily exerted after the emergency demanding the change has arisen.

There was time enough in the interval between the sure knowledge that hand chipping of blooms was a dying art and the commercial availability of the machines to plan the transfer of the men whose art was evaporating. But it was nobody's business to do this, and the job was left to the slow process of evolution.

Our attention has been focused on the uselessness of a budget where only the expenditure side has been forecast with accuracy and little is known about income. When we budget employment in terms of man power and know little about the coming demand for men, our budget falls into the same category. Unemployment can be foreseen in just about the degree that we organize for this advance knowledge.

When the Pennsylvania Railroad foresaw the end of wooden cars, it transferred the skill resident in the workers from wood to steel, and did this ahead of the emergency, so that no men were laid off.

We dare to disclose our epoch-making ideas to the Patent Office for our own gain. Will we ever dare to disclose them in some similar manner to a planning agency for the welfare of others? This is radical; but it is worthy of hard-headed thought.

ERIK OBERG.[30] The paper presents not only a most comprehensive summary of what has taken place in the last ten years in management engineering, but a prophesy of the future as well. The parts relating to industry planning and the service motive in industry are especially important, dealing as they do with the broader aspects of management. We engineers have been so busy delving into the physical laws of engineering that we have almost forgotten the purpose of it all. We have begun to look on the means as an end, and the result has been all too obvious during the past three years.

Management has to do with the running of industry efficiently. Industry is not running efficiently when a large part of its machinery stands idle and millions of workers are idle also. To that extent, management has failed, because it has not been built upon a broad enough basis. It has not had an adequate conception of fundamental economic laws. To acquire such a conception, however, is the business of management.

Most of the investigations into our economic difficulties, it seems, have not been undertaken with the courage that the engineer displays in attacking engineering problems. In engineering, we are not tied to tradition; but when we deal with business or economics, we are too partial to the methods and practices of the past. We fear to examine too critically into established commercial and economic principles. We dare not question old institutions lest we be labeled too radical; and so, in our fear of mere words, we fail to see clearly that which is obvious.

Now management must learn to see a few of these obvious things, if we are to build a rational industrial system for the future. Suppose that for the moment we forget completely the intricacies of our traditional economic methods and examine a few of the obvious facts in the world about us, just as an engineer would examine a mechanical problem. Let us try to formulate a few economic axioms—facts on which management would build, if there were no past practices to be taken into account. It seems to me that a few of these axioms are as follows:

1 We are physically able to produce more food, more clothing, and more housing—that is, more necessities—than the nation is able to consume. In addition, we are able to produce an abundance of comforts and luxuries. Yet, millions who are able and willing to give their full share of useful labor to society suffer for lack of these very necessities, let alone the luxuries.

2 When labor applied to nature produces abundantly of wheat, cotton, or other products, we ought to be prosperous. These bounteous gifts of nature ought to be welcome. Yet, our economic practices are such that ample crops are a menace. Unemployment, hardship, and depressions follow the presence of abundance in the things we all need for health, comfort, and well-being.

3 When men work efficiently, their reward ought to be security and comfort. As it is, efficiency apparently causes overproduction, and to both the engineer and the manual worker, it appears that the more efficiently he works the more insecure is his job.

4 Mechanical means for production that save toil and labor should be a blessing to mankind; they should create more leisure, greater comforts, and greater security. Instead, they appear to create insecurity, unemployment, and suffering.

5 Money is not a commodity like wheat, cotton, or iron ore; it is a means of exchange. Pieces of paper or pieces of metal used as money do not, by themselves, satisfy our needs or add to our comforts. But our economic practices seem to have made of money something more than a means of exchange—something that by its mere "scarcity" or "abundance" may create abnormal and undesirable conditions, irrespective of the scarcity or abundance of the actual commodities that we want and that we make use of.

6 The object of engineering is to obtain the greatest amount of product with the least amount of work; the object of trade is to obtain the greatest amount of goods for the least amount of work given in exchange. Hence it should be advantageous for a nation, as well as for an individual, to obtain an abundance of goods from abroad, in exchange for the minimum amount of labor on its own part. Yet, our economic practices make it undesirable for a nation to obtain goods cheaply—that is, with the expenditure of a small amount of labor—if these goods come from abroad. Under our economic methods, nations believe that they promote their prosperity and well-being by working harder in order to produce greater quantities of goods to send out of the country (exports) than they are to receive in return (imports). Briefly, in international trade, by getting as little as possible in return for our own products, we are supposed to grow prosperous.

7 Property is clearly of two kinds: Property that has been created by the exertion of man, and property that has been created by nature without human effort. Our economic practices

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[30] Editor, *Machinery*, New York, N. Y. Mem. A.S.M.E.

deal with these two kinds of property largely as if there were no distinction between them. There is also a form of value generally termed "the unearned increment" that is not created by individual effort, but by the cooperation of society as a whole. This we label "private property," just the same as the products of man's individual exertion. Many of our economic ills may be due to this confusion of ideas in regard to property.

8 Waste of human labor, due to unemployment, is the most costly waste of our industrial system. If calculated in dollars and cents, during years such as 1931 and 1932, it mounts into figures greater than any other waste for which management is responsible.

It seems to me that management must face these economic facts squarely. Any discussion of the cure for our economic ills that avoids facing these facts courageously, particularly the first and last—the fact that we are physically able to produce in superabundance and that the waste due to unemployment is the greatest of all industrial waste—cannot lead to a rational solution. We must be as courageous in our economic thought as we are in our engineering research; otherwise, we can expect nothing but continued confusion of thought.

Because we see the problem clearly, we need not assume that the sudden abandonment of all past economic traditions would be advisable. That would probably make the cure worse than the disease. But an engineer owes it to his profession to recognize fundamental facts. What action he takes after having ascertained the facts is a matter for sound balanced judgment.

Author's Closure

The discussion of this report on a decade of progress in management takes up two aspects: (1) interpretation and emphasis on certain items differing from that of the author, and (2) indication of a few developments mention of which was omitted in the report. These will be rediscussed in the order in which they appear in the preceding discussion.

Clark, Kimball, and Polakov stress the importance and enlarge upon the difficulty of national planning. Kimball emphasizes the change in the nature of the major problem confronting management engineers from that of production to that of distribution, saying that the new need "challenges the engineer and the industrialist." With all this the author is in complete sympathy.

Kimball and Kent comment upon the controversial topic of wages. The report does little more than present the aspects of the changing situation in the consideration of industrial wages and state the nature of the problems involved. No one can differ with the point of view that the entire matter is complex, difficult of solution, and of the utmost economic importance.

Kent, Wallace, and Eaton discuss technological unemployment, The divergences in their viewpoints indicate anew the need of a thoroughgoing study of this matter and of the related development and application of productive machinery. No greater contribution to our knowledge of economic processes could be made at this time than the findings of such an investigation. The temporary hardship suffered by individuals who lose their places in industry due to technical and managerial progress is a matter of common, continuing observation. The long-time readjustment whereby the same individuals find a new place in industry needs to be evaluated, before we can judge the probable extent of unemployment in any future situation. Kent calls the author to account for not showing that "mechanization of industry has created more employment for more people in the long run." While the author believes that is a true statement of what occurs, satisfying, quantitative evidence of the process is not available.

Kent comments upon the measurement of operating performance, saying, "The most important thing before management today is to learn how to evaluate itself and its own effectiveness." The author substantially agrees with this statement. If too little emphasis was placed upon this topic in the report, the lack is perhaps adequately made up in the paper which he presented jointly with J. E. Hannum on "Applications of the Kmh Method of Analyzing Manufacturing Operations" at the session of The American Society of Mechanical Engineers which received the Progress Report.

The author finds no reason to differ with Kent over his comments on budgeting and on shorter working hours. The trend toward the latter is unmistakable. Kent questions whether trade associations are intended to be included in the treatment of association and society activities. They are comprehended in the classification "business associations," and, as Kent remarks, they "may well become to an industry what the planning or production-control department is to the factory."

Polakov comments on the absence of reference in the report "to the influence of power production in the changing aims and methods of factory management." This development is not disregarded, for under the heading "Advance in Doing Work: Transfer of Skill," in treating of the increases in the physical volume of product is this statement: "The corresponding increase in utilization of primary horsepower per worker was some 30 per cent." Further, the question is quite properly raised whether the increase in primary power is not secondary to the development of productive machinery. As a reply to this query the author agrees with the point of view expressed in the following quotation from the Second Progress Report of the Committee on the Relation of Consumption, Production, and Distribution of the American Engineering Council: "The substitution of mechanisms, the transfer of human skill to machinery and tools, rather than the increased amount of power applied, is the essential change."

Bausch suggests: "Another item should be included as a part of the ten years' progress report and as an essential for recovery. This is unemployment insurance." Strong forces are gathering behind the proposals for unemployment insurance in some form. Under the heading, "Service Motive in Industry," reference is made in the report to this matter, and the enactment of laws to this end is foreshadowed: "Numerous bills to set up unemployment insurance may be expected to be introduced at the 1933 sessions of state legislatures. It will be surprising if some do not become law."

Mrs. Gilbreth in her comments takes up this matter of transfer of skill and gives an interpretation of the process in line with the most progressive modern thought. This topic deserves intense study that we may understand accurately what does occur both to man and machine when the process of transference takes place. This question groups properly under the broad heading of psychology which Mrs. Gilbreth urges for addition to the subjects in the range of interest of management engineers. The author strongly supports her suggestion.

Oberg challenges management engineers to face "economic facts squarely" and to be as "courageous in our economic thought as we are in our engineering research." He also gives eight "economic axioms" which formulate facts upon which management must build. The author accepts these axioms, as a presentation of the major features in our economic situation, and endorses the plea for courage in our economic thought.

MAN-55-3

Dissolving of Concentrated Industries

By HAROLD V. COES,[1] NEW YORK, N. Y.

Far-reaching changes in the pattern of industrial organization are discernible from the small changes that are now taking place in the evolutionary process of American industry. These changes, just discernible in some instances, seem to indicate the trend toward a better balance between urban and rural population, the development of new raw materials for industry and new sources for existing materials from agriculture, the location of industrial plants in agricultural communities to permit of lower costs and greater economic security for the workers, the greater use of branch plants and branch assembly plants to afford access to and to better serve the domestic market, a limitation of the concentration of industrial units in the large cities, and the dissolving of some of these concentrated units by scientific relocation to more suitable places. As the process goes on it should result in a better total economic balance, greater economic security to all, and more wholesome enjoyment of life by all. In bringing about and furthering these changes, the engineer, guided by economic and scientific facts, principles, and laws, will be an important factor and will play a leading part. More and more, and more than ever before, the engineer must become economic minded if he is to play his part well and serve society as it should be served.



General Statement of Problem

FAR-REACHING CHANGES in the pattern of industrial organization are discernible from the small changes that are now taking place in the evolutionary process of American industry. These changes, just discernible in some instances, seem to indicate the trend toward a better balance between urban and rural population, the development of new raw materials for industry and new sources for existing materials from agriculture, the location of industrial plants in agricultural communities to permit of lower costs and greater economic security for the workers, the greater use of branch plants and branch assembly plants to afford access to and to better serve the domestic market, a limitation of the concentration of industrial units in the large cities and the dissolving of some of these concentrated units by scientific relocation to more suitable localities. As the process goes on it should result in a better total economic balance, greater economic security to all, and more wholesome enjoyment of life by all. In bringing about and furthering these changes, the engineer, guided by economic and scientific facts, principles, and laws, will be an important factor and will play a leading part. More and more, and more than ever before, the engineer must become economic minded if he is to play his part well and serve society as it should be served.

During the early stages of the establishment and development of American industry, plants were located in small communities. The first instances of organized industrial activity on the American continent were the building of a ship by the Popham Colony in Maine in 1607 and of glass making at Jamestown, Va., in the same year. Neither of these industries developed or continued.[2] Industry was scattered; location was governed largely by the available water power. The steam engine as a prime mover was not an important factor. While the development of the steam engine rendered partially independent of water power those industries in which power was an important element, it was not until the advent of electricity as a prime mover in the early 80's that real concentration of industrial units began to take place. This was so even though steam and water power had brought about considerable concentration. Manufacturing then began to concentrate at those points where power generation and distribution were economical. This brought about concentration of labor, then labor specialization, until finally when the auxiliary and service industries moved in, we had concentration of industry as we now know it in large cities.

In the beginning of industry in this country, the workers in the factories were mainly agriculturists. Industry was frowned upon by the mother country, discriminatingly taxed, and discouraged before the Revolution. As industry became concentrated and specialized and labor more skilled and specialized, the tendency was to drop the farms and leave that work to others (as hours were long), until we ultimately arrive at the condition now confronting us in the large cities where the workers are absolutely dependent upon a pay check for existence. It is true that in some parts of New England, the Mid-West, and some other sections the workers still maintain their own homes and gardens and are to a considerable degree self-supporting during the periods of extended layoffs. In these communities one does not see the distress that prevails in the big centers of industry during a period of business recession such as the one through which we are now passing.

We have seen meat packing concentrated in Chicago, women's clothing in New York, tires and tubes in Akron, shoes in Lynn and St. Louis, automobiles in Detroit, silk in Paterson, woolens in Lowell, cotton goods in New Bedford and Fall River, men's clothing in Rochester, Cleveland, New York, and Chicago, and steel in Pittsburgh, Chicago, and Birmingham.

The Forces at Work

The factors and economic forces that brought this about are too numerous for discussion in this paper. They were generally availability of power, of raw materials, of skilled labor and craftsmanship, of markets, of transportation facilities, and of local capital. We are concerned here with the reverse of the

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Holmes, "Plant Location," p. 151.

process—dissolution of concentration. What is bringing this about? It seems to the author that two groups of forces are at work:

(*a*) Economic
(*b*) Sociologic

and heretofore the sociologic factors have been given relatively little attention. In the first group might be listed such factors as:

National rather than localized markets
Electrical distribution
Natural-gas distribution
Hard-surface roads
The automobile
The airplane
Communication lines
Trucks
Transportation costs
Better rail service—store-door delivery, container cars, etc.
Time of delivery to markets
Servicing
Automatic machinery
Wastes in distribution
Higher rents and values of land
Higher cost of operation in metropolitan areas
Higher taxes in metropolitan areas
Higher cost of labor
Higher costs of municipal government
Hazards of fire due to large concentrations.

In the second group are such considerations as:

Uncomfortable and bad living conditions
Health of wage earners
Racketeering
Municipal restrictions
Instability of labor
Hazards of plant-operation interruption due to concentration of large aggregations of labor
Insecurity of labor.

The operation of these factors is bringing about a diffusion of manufacturing capacity. The statement, "Present Distribution of Some Typical Industries" (see Table 1, Appendix), indicates to some extent what is taking place.

The author had occasion, several years ago, to study scientifically the cost of production of insulated wire in a large metropolitan area as compared with producing it in smaller suitable urban communities, and the total differential in cost of production and of distribution was about 25 per cent in favor of the most suitable of the smaller communities. Furthermore, the living conditions for the wage earners and their economic security were greatly enhanced.

The development of the electrical-power and natural-gas industries and of our railroad service, as well as our network of public highways, has given to plant location a new mobility.

The factors enabling this mobility away from cities began to emerge about 1910, chief of which was economical power generation and distribution to points well outside the large population centers. Manufacturing plants no longer have to locate in the large centers to secure cheap and reliable power.

Authoritative Statements on the Dissolving of Concentrated Industries

The Iron Age, January 15, 1931, page 221, makes these comments on industry dissolution:

The truly amazing development of products naturally occupies the foreground of any picture of industry's evolution. But back of the products, less obvious perhaps because of them, we may catch a glimpse of slower but far-reaching changes in the industrial organization itself. New cells are being added to this organism, old cells are being discarded, growth continuing with only occasional interruption. And in the process there is a distinguishable alteration of cellular arrangement, sufficient to bring about, say in ten-year periods, decided changes in the pattern of the organization as a whole.

Today it is not necessary for a factory to be next door to a power plant. Craftsmanship, handed down from generation to generation, is no longer a command to the plant to seek the community where skill exists. The development of automatic machinery has been gradually eliminating the factor of skill. And low-priced motor cars have done the rest by increasing the radius of the skilled worker. Transportation of raw materials and finished product is becoming less and less a determining factor since the motor truck has effected liaison with the railroad.

Our experience during this depression is likely to quicken the decentralization movement. Unemployment conditions were aggravated in those highly specialized industrial centers where like industries were concentrated. The least suffering occurred in communities where diversification was evident. A regrouping of industry made possible by its new liberty of movement might be of great social as well as economic benefit. Both plants and people get the least for their money in the big cities.

Mr. A. W. Robertson, Chairman of the Board of the Westinghouse Electric & Manufacturing Co., makes this statement in *The Iron Age* of the same issue, page 223:

Industry is seeking the open spaces and leaving the congested city factory district. Today, the factory finds it possible to escape from the city with its high rents and its uncomfortable living conditions. The springing up of the state-wide power networks and the development of automotive transportation have given industry a mobility which it did not have 20 years ago.

Most of our new plants are being built in uncongested areas. The industrial map of today is quite different from that of ten years past, and the coming ten years will bring still more diffusion.

Mr. Sidney G. Koon, associate editor of *The Iron Age*, in the issue of January 5, 1928, page 27, made a statistical analysis of the movement from East and West into the Indiana-Ohio-Michigan section. His study was confined largely to a comparison of rolled iron and steel production. In tabular form this analysis is as follows:

States	Date	Per cent of U. S. total	Per cent at end of 1927
Pennsylvania	1911–1914	49½	39
Indiana	1911–1914	6	13
Ohio	1911–1914	18	22
Ohio, Indiana and Illinois	1911–1914	34	42

Taking manufacturing in general and dealing with it on the basis of the total number of wage earners employed, the following facts appear:

States	Year	Per cent
New York, Pennsylvania, and New Jersey	1904	34
	1925	30
Ohio and Michigan	1904	10
	1925	14
Indiana and Illinois	1904	10
	1925	10½

He goes on to state that more than four-ninths of all products manufactured in the United States are produced in 68 cities, which had in 1920 a population of 100,000 or more. In some respects the study of production in the principal cities has yielded results similar to those cited. Taking the census figures for 1914 to 1925, inclusive, the cities showed their highest proportion in 1921 with 46.98 per cent of the total; their lowest came in 1925 with 44.81 per cent.

Mr. Martin J. Insull, in an article for the Engineering Foundation, appearing in *Electrical Engineering*, September, 1931, page 741, makes this statement:

By 1831, power had changed the decentralized industry of 1731 into a centralized industry. In 1931 power appears to be changing industry back into a decentralized structure—industry is not slow to recognize the advantages of decentralization in terms of lower costs and improved working conditions. In this there is hope for those who view with despair our industrial evolution. Let them not con-

clude that because we were headed toward industrial concentration for a century or more we cannot reverse the direction of our development. We need not necessarily expect still more congested cities, still higher skyscrapers, nor a depopulated countryside. The industry of the future is more likely to reinhabit the countryside, to possess all the advantages of power machinery without the disadvantages of congestion which have hitherto accompanied it.

Mr. George M. Verity, President of the American Rolling Mill Company, in an interview for *Electrical World*, May 12, 1928, page 973, comments on "Industrial Dissolution" as follows:

An outworn argument for the location of industries in so-called industrial centers has been that labor is plentiful in those centers. This is true. Labor of a kind is usually plentiful. But with that labor industry accepts the evils of a competitive labor market, of labor living in slums and less effective in consequence, of selfishly factional local governments, and of labor's disinterest in everything except wages. The plants of the American Rolling Mill Company are located in small cities. In these we know that the homes of our employees are brighter, more comfortable. We know that the eight hours of the day remaining after work and sleep are taken out are healthier and happier. We know that recreation is diversified and largely out of doors. We know that social life is at its best. We know that contact of our employees with community and municipal problems is closer and their participation in the administration of local government is on a broader and less sectarian basis. We know that the community of our employees is an important entity in the small city. We know these things because they are the reasons why our plants are located in small cities.

And finally:

Both employer and employed must develop an awareness of social duties, the performance of which will result in benefit to both. Nowhere but in the small town where relative size is greater can this condition be realized. This is why "Armco" is committed to the small town.

Mr. Henry Ford, in a paid advertisement in *Business Week*, June 22, 1932, advances this conclusion:

We are still in the early stage of world making. To stabilize conditions at a given point or to reject progress because it brings problems with it, is impossible. We need to plan how to pass through necessary changes with the least human hardship. Economic changes must come, and it is possible for men to be in a position to welcome, rather than dread them. With factory and farm as partners, with one foot on the soil and one foot planted on industry, we should be in a position to hail the new epoch without fear. The land would protect us from needless anxiety.

Comparison of Earnings of Large- and Small-Scale Industries

There is quite a popular impression that concentration is brought about by consolidation and merger, and that this is an economic necessity prerequisite to greater and more stable earnings. This impression, however, is not supported by the facts.

In the early nineteen hundreds many prominent economists wrote of the "important positive advantages," as expressed by one of them,[3] due to large-scale production. Another[4] announced that "modern production tends to become concentrated." It was not until a few years later than an economist expressed the idea that the concentration of production facilities was not necessarily essential to successful consolidation. He stated that the industrial factory must be limited in size for maximum efficiency, but that these limitations "applied with far greater force to a single local factory than to a federation of scattered plants. . . . The limits to the advantages of combination have been removed by the application of the federation plan."[5]

None of the writers ventured to suggest that there might be a point at which the law of diminishing returns would operate, a fact which would have a decided bearing on the limits of centralization and be a factor in the reverse process of dissolving concentration. The writers of that period, whether intentionally or otherwise, seemed to focus on the idea that size was the ultimate goal, the means to economic salvation, and it was inferred that there would be no place ultimately for the moderate-sized plant. Nothing, in the author's judgment, based on his observations and experience, can be wider of the truth. There are some products for some markets that lend themselves more readily to large-scale production than others, the producing units of which (i.e., individual plants or factories) can be on a larger scale for more economical use of large and expensive equipment. This does not mean that the whole production must of necessity be concentrated in one yard or plant.

Is there any evidence that individual plants of suitable size for maximum efficiency of operation scientifically located—the real core of the dissolving of concentrated industries—yield greater returns on the invested capital? A recent study by Prof. H. B. Summers, of Kansas State College,[6] sheds some light on this subject. He studied the earning records of 1130 companies over a period of 20 years, 1910 to 1929, inclusive, classified as follows:

		Companies
1	Iron and steel, including forgings, railroad equipment, and shipbuilding	139
2	Automobiles, including parts and accessories	82
3	Machinery, including heavy, light, farm equipment, office equipment, etc.	107
4	Hardware, including plumbing fixtures, electrical equipment, and specialties	82
5	Petroleum, including all branches of the industry	93
6	Chemicals, including pharmaceuticals, paints, and specialties	67
7	Textiles, including silk, rayon, cotton, woolens, linens, and heavy textiles	114
8	Foodstuffs, including baking, cane products, milling, fruits, packing, sugar	152
9	Miscellaneous, including building materials, furniture, glass, leather, lumber, paper, rubber, tobacco, and mining	294
10	Total all classes	1130

He first classified all companies in a size class (see Table 2, Appendix), then submitted a study showing average earning rates for all companies by size classification for the 20-year period (Table 3, Appendix), and then average earning rates by industries, by size classification for the 20-year period (Table 4, Appendix), and finally comparative earning rates, large and small companies, for the 20-year period (Table 5, Appendix), and he concluded as follows:

1 The highest rate of earnings for the 20-year period, all companies considered, is found in the smaller size group (see Table 3, Appendix).

2 Of the nine industrial classifications made, the highest average rate of earnings was found in the smallest size group, including companies with investment below $2,000,000 in five of the nine classes (see Table 4, Appendix).

3 In five of the nine classifications there is apparently a definite relationship between the amount of investment and long-time rate of earnings, with average earning rates decreasing as investment is increased. Marked variation from this tendency in earning rates is found only in the hardware group. In machinery manufacturing, there apparently is no relation between size of investment and earning power, and the same seems to hold true in the petroleum and miscellaneous industries classifications (see Table 4, Appendix).

4 Considering Table 5, Appendix, no matter where the dividing line is set, small companies when considered by industries, with four exceptions, show higher rates of earnings than large. If the dividing line is fixed at $2,000,000, the smaller companies show higher average earnings in each of the nine industrial classifications.

[3] Seager, "Introduction to Economics," p. 150.
[4] Bullock, "Introduction to the Study of Economics," p. 178.
[5] Fetter, "Principles of Economics," p. 320.
[6] "A Comparison of the Rates of Earnings of Large- and Small-Scale Industries," *Quart. Jl. of Econ.*, May, 1932, p. 465.

5 With respect to any positive advantages accruing to large-scale production, the study offers nothing whatever in support of the idea that size in itself brings greater earning power.

The foregoing study has been given prominence in this paper, since the findings disclosed should have an important bearing on our thinking with respect to the problems of dissolving of concentration and would seem to indicate that moderate-size plants, scientifically located, have decided economic advantages in the long run. No attempt apparently was made in this study to determine the effect of proper location of individual units in a given industrial classification. If the companies compared in the paper of Professor Summers were studied from this angle, some very interesting and pertinent facts might be disclosed. We are indebted to him for the facts he has so thoughtfully brought to our attention..

In the study of economic changes in the United States by the National Bureau of Economic Research, a table is presented showing in specified states the increase and decrease in manufacturing.[7] (See Table 5, Appendix.)

This shows a marked decline in manufacturing and establishments in the larger cities for the period reviewed, 1919 to 1925, inclusive, particularly for the so-called industrial states, and also a decided trend toward industrial growth in towns of less than 25,000 population and in agricultural states.

In commenting on the trend toward dissolving concentration, the writer makes these statements:

> It seems quite conceivable that the newer agriculture will tend to maintain a skeleton force on the land and transfer labor from town to country to meet seasonal requirements. . . . The easier way seems the shift of tractor men and practically unskilled harvest hands into the country during short periods of special needs, with the same groups productively fitted into industrial callings of the town during agriculture's slack season. . . .
>
> All in all, it seems that the hard and fast line between agriculture and other industries and between rural and urban residence will tend to disappear. . . . A tendency is noticeable among some manufacturers to get their plants distributed in towns and villages adjacent to agricultural land. . . . It is impossible to show this approach of manufacturing toward the agricultural regions by statistical methods with as much clearness as could be desired. But it is apparent from the census of manufacturers over the last few years that a drift away from the highly industrialized centers in the Northeast and toward the smaller towns and villages in the Central, Southern, and Western states has been under way. . . .
>
> That the rapprochement of city and country which appears to be under way in response to new conditions of auto transportation and to the organization of both farm and factory work, holds the possibilities for improved conditions on both sides. . . .

Plant and Branch Plant Relocations

In a cooperative survey by the Civic Development Committee of the National Electric Light Association and the Policy Holders Service Bureau of the Metropolitan Life Insurance Company, entitled "Industrial Development in the United States and Canada, 1926 and 1927," 1934 communities in the United States were studied, reporting over 10,000 plants and 371,000 workers gained in that period by the reporting communities. Plant relocations accounted for 9.4 per cent of the plants and 18.7 per cent of the employees. New industries accounted for 81.8 per cent of the plants and 56.3 per cent of the employees gained, while branch plants were responsible for the remainder of 8.8 per cent of the plants and 25 per cent of the employees. The average size of plants gained for the country as a whole was 36 employees. The average relocated industry had 71 employees and the average branch plant had 103 employees. The average new local industry had 25 employees.

The reasons in the order of their importance for all gains for the United States as a whole are shown by the following list (see Table 8, Appendix):

1 Markets
2 Labor
3 Transportation
4 Materials
5 Available buildings
6 Power and fuel
7 Near related industries
8 Living conditions
9 Financial aid
10 Taxes
11 Consolidation
12 Laws
13 Disposal of waste.

Selecting arbitrarily 50,000 population as the dividing line between larger and smaller cities, the survey shows that the larger cities gained 58.4 per cent of the total employment gains and 73.1 per cent of the total plants. (This reflects the process of concentration.) They give as the reasons for this the large number of "satellite" industries which are established around existing industries located therein and the development of style industries in clothing and related groups in metropolitan areas.

They point out that the extent to which smaller cities have gained through establishment of branches and relocations within their borders is worthy of special note. Of all gains made by these cities, those under 10,000 in population received 26.2 per cent of all branches reported in the survey and 17.9 per cent of the locations. (This shows the principle of decentralization at work.)

The percentage of the total gain in cities, classified by population, is as follows:

Cities of	Per cent
150,000 or over	39
50,000 or over	21
10,000 or over	21
2,500 or over	13
Under 2,500	6

The United States Chamber of Commerce also made a study of branch plant establishment and location, the facts concerning which are summarized in Table 9, Appendix.

The use of branch plants as a means for industrial expansion and dissolving of concentration is a comparatively recent development and is primarily the direct result of the operation of economic forces, particularly in the field of marketing and service.

It will be noticed that the diffusion is largely in the Middle Atlantic, East North Central, and the Pacific States divisions. The largest number of branches with headquarters outside of the division are located, first, in the Pacific division and, second, in the South Atlantic division.

The same study provides some information as to the size of the branches established, on the basis of wage earners employed, as follows:

Wage earners	Number of branches
1 to 5	17
6 to 10	33
11 to 25	53
26 to 50	40
51 to 100	34
101 and over	41

There is evidently a trend toward more equal distribution of manufacturing in many industries throughout the country.

Some light can be shed on the large city dissolving of concentration by studying the shift of wage earners between cities and urban localities, as discussed in "Recent Economic Changes:"[8]

Between 1919 and 1925, while the 25 largest cities in the country lost 326,800 wage earners, about 12 per cent of the average number

[7] "Recent Economic Changes," vol. 2, p. 591.

[8] Ibid., vol. 1, p. 210.

employed, the rural area (all area not included in communities having a population of 10,000 inhabitants) gained 55,204 wage earners. Nor is this decrease limited to the larger cities. Cities between 100,000 and 250,000 population lost even more heavily, 14 per cent of their wage earners, communities between 25,000 and 100,000 lost 11 per cent, and those between 10,000 and 25,000 lost 8 per cent of their wage earners.

The United States Chamber of Commerce made a study of this subject as well, and the findings are summarized in Table 10, Appendix.

As evidence of the changes in concentration, the following is of interest:

AVERAGE NUMBER OF WAGE EARNERS PER ESTABLISHMENT BY SIZE OF COMMUNITY, 1919 AND 1925[9]

Population of community	1919[a]	1925
Over 250,000	35.7	35.1
100,000 to 250,000	60.6	56.7
25,000 to 100,000	63.1	64.0
10,000 to 25,000	52.9	57.9
Remainder	31.0	42.3

[a] Corrected for establishments with product under $5000, but not for "automobile repairing" or "coffee roasting," both of which were included in 1919 and not in 1925.

Additional evidence of the dissolving of local concentration of industry is shown by the following computation:

DISTRIBUTION OF 105 INSTANCES OF LOCAL CONCENTRATION[10]

(According to the census date at which the community reached its highest percentage of the national total)

1899	32
1904	21
1909	15
1914	4
1919	14
1923	9
1925	10
	105

This was obtained "from examining all cases for which continuous records were available since 1899 for communities which at some time produced at least 3 per cent of the national product in some industry. In this way it was possible to obtain records for 105 instances of local concentration. The percentage which each community was of the national total was then computed for each census date to determine whether it was gaining or losing in relative position. . . . It is quite conclusive that one-half the cases examined reached their highest point by 1904, and that but 10 of the 105 were at their highest point when the record ended."[11]

The Shift in Population

The shift in population from rural to urban in the United States can be seen from Table 11, Appendix. In 1870, 79 per cent was rural and 21 per cent urban; in 1900, 40 per cent was urban and 60 per cent rural; while in 1930, the urban population was greater than the rural, 56 per cent being urban and 44 per cent rural.

Table 12, Appendix, shows the shift by major geographic divisions from 1880 to 1930, inclusive.

Factors That Will Assist Toward the Dissolving of Concentrated Industries

That dissolving of concentration of industry is taking place is evidenced by the policies pursued by several large companies. The Ford Motor Company builds assembly plants at suitable points to serve its various markets throughout the country, a decentralization of some of its manufacturing and assembly operations. The General Electric Company has a policy of acquiring or building new plants for new products, rather than concentration of manufacture at Schenectady. Dissolving of concentration is also brought about by the economics of distribution and marketing, resulting in many companies locating branch plants at strategic points to better serve their markets.

What has probably not been realized, up to the present, is the menace to society as a whole from large masses of workers thrown out of employment and with no visible means of support. We must learn to plan better. When we realize the fundamental relationships between the support of workers from the land and from industry, or, as Mr. Ford puts it, with factory and farm as partners or factory and suburban gardens as partners, then perhaps accelerated dissolution of concentration will take place to the lasting benefit of workers, employers, and society.

Modern means of transportation and communication permit centralized control of isolated units. With teletypes, and now with television just around the corner, this control will be rendered speedier, easier, and more satisfactory.

Local advantages become increasingly less important with increased facilities in transportation and greater mobility on the part of industrial workers. This is so markedly true that the problem of locating any new industry has become one of economic consideration into which many factors must enter.[12]

Studies of wastes in distribution will probably be a real factor in dissolving industrial concentration. If it is true, as some of us believe, that our production technique is far ahead of our distribution technique and that improvement in production technique is reaching the point in many instances of diminishing returns, then manufacturers in the future must look for their profit in better distribution technique and the elimination of many of these distribution wastes. This means that scientific plant location, branch plant location, etc. will be the order of the day and that plant locations will be chosen with a scientific regard for the factor of distribution. It is a significant fact that scientific marketing analysis and marketing has measurably lagged behind scientific production methods.

It is believed that in the past too much weight has been given to population as a basis for distribution. Mr. Holmes makes this observation:

> It has often been pointed out that manufacturing goods cannot be sold to square miles of land. Neither can they be sold to thousands of people. The theory that population is an index of the market for manufactured commodities is false. No one article is consumed or one service purchased precisely according to the distribution of population. The potential market for a manufactured product depends upon many different factors.[13]

Scientific marketing then will bring about the dissolving of industrial concentration.

There is still another factor that must receive more study and attention than it has, and that is the decreasing number of man-hours required per unit of production. Prof. Walter Rautenstrauch of Columbia University cites some figures in connection with the 100-year energy survey being conducted by Columbia University at the present time, which shows the trend in some industries:

Product	Date	Man-hours
Cement	1905	1.75 per bbl
	1930	0.45 per bbl
Automobile	1904	12.91 per car
	1929	9.2 per car
Steel	1900	7.0 per ton
	1929	1.3 per ton

These studies so far indicate that employment reached its peak in 1918 and has declined ever since, while production reached its peak in 1929.

[9] "Recent Economic Changes," vol. 1, p. 213.
[10] Ibid., p. 215.
[11] Ibid., p. 215.
[12] Kimball, "Industrial Economics," p. 62.
[13] "Plant Location," p. 21.

This would indicate a declining employment rate in basic established industries for normal consumption, and in the absence of a new industry to take up this slack in employment, such as occurred when the automobile and radio industries were born, or without increased consumption of the products of these basic industries, we have in so far as spreading employment is concerned, a real problem to solve. To what extent we can solve this problem by the partnership of the land and the factory and by a greater knowledge of consumer demands and preferences is not now clear. That it must be solved, however, is quite clear.

The author believes there are illimitable possibilities in the proposed research in agriculture to produce perennially for industry many of the raw materials it needs and many new ones. There should ensue much desired diffusion of labor to extraction plants in the agricultural districts which would further bring about as a concomitant result, in all probability, dissolving of industrial concentration.

Just what part safe and inexpensive airplanes may play in the dissolving of industrial concentration and in widening the radius of activity of industrial workers is difficult now to forsee, but it is conceivable that it might extend their radius and their mobility almost as much over the automobile as the automobiles did over the horse and buggy. We could then visualize ourselves as living on a farm or in an agricultural community 100 or more miles away from our offices and require no more, or perhaps less, time to commute than we now take. The factory could readily be 100 or more miles from the office, located in a large city, and yet be readily accessible.

That there will be a drift away from the large centers, both of industry and of people, to points where both can enjoy the benefits of relatively unconfined space is the author's firm belief.

Conclusion

The foregoing authorities have been cited to show the trend of thinking—the conclusions that some of them draw.

There is then sufficient evidence for one to come to the conclusion as a result of economic and sociologic forces that our industrial pattern is undergoing a change. There is a growing consciousness that after all there are no such beings as supermen; that perhaps there is a limit to man's comprehension; that maybe we can get physical units too big and aggregations of employees too large for sound and efficient administration and management. We are realizing that cities can become too large for comfort, well being, and safe and efficient administration; that the advantage of a large city location for industrial units, except for those organizations which must of necessity be located in or immediately adjacent to a metropolitan area, can be more than offset by the disadvantages.

Again, there is a growing consciousness of the economic insecurity of masses of our population living in the large cities whose sole dependence is on a regular pay check, and that while business continues to move in cycles, whole sections of the population can and are temporarily at least dislocated, disorganized, and in distress; that this imposes huge burdens on the employed population, the employers, the city welfare organizations, the city administration, and the state—sometimes, as at present, greater than their resources can sustain—and that therefore we have thrown our rural and urban population out of balance. (See Table 11, Appendix.)

All of the foregoing considerations are leading thoughtful men and women to the conclusion that we can best bring our population into balance, strengthen our economic position, and bring more light and happiness into the lives of thousands of families if the process of the dissolving of industrial concentration is speeded up.

Appendix

The statistical matter referred to in the text upon which the author has based some of his conclusions is contained in this Appendix.

TABLE 2 CLASSIFICATION OF COMPANIES BY SIZE

Size class	Including companies with permanent investments ranging		
1	Less than		$2,000,000
2	From $2,000,000	to	$5,000,000
3	From $5,000,000	to	$10,000,000
4	From $10,000,000	to	$25,000,000
5	From $25,000,000	to	$50,000,000
6	From $50,000,000	to	$100,000,000
7	More than		$100,000,000

TABLE 3 AVERAGE EARNING RATES FOR ALL COMPANIES, 20-YEAR PERIOD, IN PERCENTAGES

	Size group						
	1	2	3	4	5	6	7
Investment size..	Under 2 million	2–5 million	5–10 million	10–25 million	25–50 million	50–100 million	Over 100 million
Earnings of all companies.....	11.6	9.2	9.3	8.8	9.4	9.8	9.5

TABLE 1 PRESENT DISTRIBUTION OF SOME TYPICAL INDUSTRIES

Industry	Number of states containing most plants	States in which most plants are located	Number of states with greatest concentration	States with greatest concentration	State with largest percentage of total capacity
Wood pulp and paper..	19	Me., N. H., Vt., Mass., Conn., N. Y., Pa., N. J., Md., Va., Ohio, Mich., Ind., Ill., Wis., La., Minn., Wash., Cal.	5	Me., Mass., N. Y., Mich., Wis.	N. Y.
Pig iron.....	8	N. Y., Pa., W. Va., Ohio, Mich., Ind., Ill., Ala.	4	Pa., Ohio, Ind., Ill.	Pa.
Glass.....	10	N. Y., Pa., N. J., Ohio, Md., W. Va., Ind., Ill., Okla., Cal.	4	Pa., W. Va., Ohio, Ind.	Pa.
Pottery.....	10	Mass., N. Y., N. J., Pa., W. Va., Ohio, Mich., Ind., Ill., Cal.	2	N. J., Ohio	Ohio
Furniture.....	16	Mass., N. Y., Pa., N. J., Md., Va., N. C., Tenn., Ky., Ohio, Mich., Ind., Ill., Mo., Wis., Cal.	3	N. Y., Mich., Ill.	N. Y.
Petroleum refining.....	12	Mass., N. Y., N. J., Pa., Ohio, Ill., Okla., La., Kan., Tex., Wyo., Cal.	3	N. J., Tex., Cal.	Tex.
Canning.....	17	Me., N. Y., N. J., Md., Pa., Ohio, Ind., Mich., Ill., Wis., Mo., Iowa, Minn., Utah, Wash., Ore., Cal.	3	N. Y., N. J., Cal.	Cal.
Dairy products.....	19	N. Y., Pa., Ohio, Mich., Ind., Wis., Ill., Minn., Iowa, Mo., Kan., Neb., S. D., N. D., Col., Ida., Wash., Ore., Cal.	2	Wis., Minn.	Wis.
Cane sugar.....	9	Mass., N. Y., N. J., Pa., Md., Ga., La., Tex., Cal.	2	N. Y., La.	N. Y.
Beet sugar.....	14	Ohio, Mich., Ind., Ill., Wis., Minn., Iowa, Neb., Kan., Col., Wyo., Ida., Utah, Cal.	3	Mich., Col., Utah	Utah
Salt.....	7	N. Y., Ohio, Mich., Wis., La., Kan., Cal.	2	N. Y., Mich.	N. Y.
Soap.....	9	Mass., Conn., N. Y., Pa., N. J., Ohio, Ind., Ill., Cal.	3	N. J., N. Y., Ohio	N. J.
Varnish and paint.....	12	Mass., N. Y., N. J., Pa., Ohio, Mich., Ind., Ill., Wis., Mo., Ky., Cal.	5	N. Y., N. J., Ill., Pa., Ohio	N. Y.
Cocoa and chocolate...	7	Mass., N. Y., Pa., N. J., Ill., Wis., Cal.	3	Pa., N. Y., Mass.	Pa.
Confectionery.....	18	Mass., N. Y., N. J., Md., Pa., Va., Ohio, Ind., Mich., Wis., Ill., Ind., Tenn., Ga., Mo., Iowa, Minn., Cal.	4	N. Y., Mass., Pa., Ill.	N. Y.
Cement.....	27	N. Y., N. J., Pa., Md., Va., W. Va., Tenn., Ga., Ala., Ohio, Mich., Ind., Wis., Ill., Mo., Iowa, Ala., Tex., Kan., Neb., S. D., Col., Mont., Utah, Wash., Ore., Cal.	5	Pa., N. Y., Mich., Ala., Cal.	Pa.

TABLE 4 AVERAGE EARNING RATES BY INDUSTRIES, 20-YEAR PERIOD, IN PERCENTAGES

Industry	Size group 1	2	3	4	5	6	7
Iron and steel	11.2	8.0	7.8	7.8	7.1	8.9	7.1
Automobiles	18.5	13.6	17.4	14.5	14.1	9.8	17.9
Chemicals	24.5	20.4	11.8	10.7	8.0	7.2	17.3
Petroleum	18.7	10.6	11.5	10.3	11.3	10.9	11.7
Miscellaneous	12.3	8.9	8.9	9.1	8.1	9.3	9.3
Textiles	7.8	8.0	8.4	7.6	6.8	4.9	1.0
Machinery	8.0	6.5	9.4	7.5	6.7	7.4	9.3
Hardware	11.2	11.0	8.7	8.8	19.8	12.7	12.7
Foodstuffs	9.8	9.2	9.1	8.3	10.7	8.0	7.6

TABLE 5 COMPARATIVE EARNING RATES, LARGE AND SMALL COMPANIES, 20-YEAR PERIOD, IN PERCENTAGES

Industry	With investment of $25,000,000 Small	$25,000,000 Large	$10,000,000 Small	$10,000,000 Large	$5,000,000 Small	$5,000,000 Large	$2,000,000 Small	$2,000,000 Large
All companies	9.5	9.4	9.8	9.3	10.1	9.2	11.6	9.2
Iron and steel	8.2	7.7	8.4	7.7	8.9	7.7	11.2	9.8
Automobiles	15.5	13.4	16.9	14.0	15.3	14.9	18.5	14.5
Petroleum	11.4	11.3	11.6	10.5	12.4	11.1	18.7	10.8
Chemicals	14.9	9.0	16.5	9.8	22.4	10.5	24.5	12.0
Textiles	8.0	5.6	8.0	7.2	7.9	7.7	7.8	7.7
Machinery	7.6	7.1	7.8	7.6	7.0	7.8	8.0	7.4
Foods	8.9	9.3	9.5	8.6	9.4	8.8	9.8	8.9
Hardware	9.9	15.8	10.4	10.9	11.1	10.2	11.2	10.5
Miscellaneous	9.5	8.6	9.6	8.8	10.1	8.9	12.3	8.9

TABLE 6 PERCENTAGE OF INCREASE AND DECREASE IN MANUFACTURING IN SPECIFIED STATES, 1919–1925

Size group	New England States Number of establishments	New England States Number of wage earners	Middle Atlantic States Number of establishments	Middle Atlantic States Number of wage earners
Total	−11.7	−16.7	−17.5	−13.0
Over $250,000	− 3.3	−12.6	−16.6	−15.5
$250,000–$100,000	−10.7	−20.0	− 5.4	−16.8
$100,000–$25,000	− 5.4	−18.2	−14.7	−13.6
$25,000–$10,000	− 7.4	−15.1	−19.3	− 7.6
Less than $10,000	−22.6	−14.8	−23.5	− 9.2

Size group	Indiana Number of establishments	Indiana Number of wage earners	Michigan Number of establishments	Michigan Number of wage earners
Total	−19.9	+ 1.7	−17.6	+ 9.6
Over $250,000	−10.7	−16.3	−16.9	+ 3.5
$250,000–$100,000	....	....	− 5.0	+ 8.5
$100,000–$25,000	−15.6	+ 5.0	− 9.4	− 0.2
$25,000–$10,000	−10.6	− 6.7	−11.1	−15.5
Less than $10,000	−27.2	+15.0	−24.5	+44.0

Size group	North Carolina Number of establishments	North Carolina Number of wage earners	South Carolina Number of establishments	South Carolina Number of wage earners
Total	−40.5	+17.1	−21.0	+27.2
Over $250,000	....	....	....	....
$250,000–$100,000	....	....	....	....
$100,000–$25,000	− 1.8	+ 1.1	−34.7	−20.6
$25,000–$10,000	−10.7	+46.0	−24.0	+ 1.2
Less than $10,000	−47.2	+ 7.5	−17.7	+37.0

TABLE 9 COMPARATIVE LOCATION OF HEADQUARTERS AND BRANCH PLANTS CONSTRUCTED IN 1927

Location of branch	Location of headquarters: New England	Middle Atlantic	East North Central	West North Centra	South Atlantic
New England	6	6	2	..	..
Middle Atlantic	2	23	3	1	3
East North Central	1	6	22	1	..
West North Central	1	3	3	14	..
South Atlantic	1	14	4	3	8
East South Central	..	1	3	..	1
West South Central	..	..	1	3	1
Mountain	..	..	1	..	..
Pacific	2	18	19	2	1
Total	13	71	58	24	14

Location of branch	East South Central	West South Central	Mountain	Pacific	Total
New England	..	..	..	..	14
Middle Atlantic	..	..	..	..	32
East North Central	1	..	1	4	36
West North Central	..	2	1	1	25
South Atlantic	2	..	..	..	32
East South Central	2	..	..	..	7
West South Central	2	6	..	..	13
Mountain	..	..	5	2	8
Pacific	..	..	..	13	55
Total	7	8	7	20	222[a]

[a] "Recent Economic Changes," vol. 1, p. 209.

TABLE 7 THE SPREAD OF INDUSTRY

Industry	Per cent	Produced by
Steel:		
1900	60	N. Y., Pa., and N. J.
1924	48	N. Y., Pa., and N. J.
1900	31	Ohio, Ind., Ill., Mich., and Wis.
1924	40	Ohio, Ind., Ill., Mich., and Wis.
Meats:		
1900	46	Chicago and vicinity
1924	33	Chicago and vicinity
1900	28	West North Central Group
1924	33	West North Central Group
Cotton goods:		
1900	60	New England States
1924	35	New England States
Electrical machinery:		
1909	50	Middle Atlantic States
1924	38	Middle Atlantic States
Soap:		
1909	6	New England States
1924	9	New England States
1909	42	Middle Atlantic States
1924	35	Middle Atlantic States
1909	36	East North Central Group
1924	28	East North Central Group
1924	5	Pacific States
Book paper:		
1909	28	New England States
1924	12	New England States
1899	14	East North Central Group
1924	25	East North Central Group
Boots and shoes:		
1899	60	New England States
1924	35	New England States
1924	60	Mid. Atl. and East No. Cen. and Mo.
Total manufactures:		
1900	15	New England States
1924	11	New England States

TABLE 10 DISTRIBUTION OF PLANTS REMOVED IN 1927
(According to the population of the community left and the community entered)

Population of community entered (in thousands)	Population of community left (in thousands): Under 10	10 to 25	25 to 50	50 to 100	100 to 200	200 to 300	300 to 500	Over 500	Total
Under 10	12	6	3	2	1	..	2	18	44
10 to 25	12	4	1	2	5	2	2	12	40
25 to 50	5	2	4	1	..	2	2	19	35
50 to 100	6	1	4	2	1	..	5	10	29
100 to 200	4	3	1	6	3	..	..	21	38
200 to 300	4	..	..	..	2	..	..	4	10
300 to 500	2	..	..	1	..	2	..	4	9
Over 500	1	4	1	4	4	1	2	6	23
Total	46	20	14	18	16	7	13	94	228[a]

[a] "Recent Economic Changes," vol. 1, p. 213

TABLE 11 SHIFT IN POPULATION FROM RURAL TO URBAN IN THE UNITED STATES

Year	Per cent urban	Per cent rural
1930	56.2	43.8
1920	51.4	48.6
1910	45.8	54.2
1900	40.5	59.5
1890	36.1	63.9
1880	29.5	70.5
1870	20.9[a]	79.1
1860	16.1	83.9
1850	12.5	87.5
1840	8.5	91.5
1830	6.7	93.3
1820	4.9	95.1
1810	4.9	95.1
1800	4.0	96.0
1790	3.3	96.7

[a] For year 1870 and prior, on basis of 8000 inhabitants or more; since 1880 on basis of 2500 or over.

TABLE 13 COMPARISON OF POPULATION AND INDUSTRIAL DISTRIBUTION, IN PERCENTAGES

Geographic division	Census 1900 Population	1900 Manufactures	1929 and 1930 Population	1929 and 1930 Manufactures
New England	7.4	14.6	6.6	9.0
Middle Atlantic	20.3	35.7	21.4	30.3
South Atlantic	13.6	6.2	12.8	7.4
East North Central	21.0	25.0	20.6	30.7
East South Central	9.9	2.8	8.0	2.8
West North Central	13.5	8.5	10.8	7.6
West South Central	8.7	2.2	9.9	3.9
Mountain	2.2	1.7	3.0	1.7
Pacific	3.1	3.2	6.6	6.1

TABLE 8 FACTORS IN PLANT LOCATION, RELOCATION, AND LOCATION OF BRANCHES, IN THEIR RANKING ORDER

(From Industrial Development Report of the Metropolitan Life Insurance Company and the National Electric Light Association)[a]

		Group												
	Industry	1 Markets	2 Transportation	3 Materials	4 Available buildings	5 Labor	6 Power	7 Living conditions	8 Near related industries	9 Financial aid	10 Taxes	11 Consolidation	12 Laws	13 Disposal of waste
	Relocation Factors													
(1)	Foods	1	2	3	4	5	6	7	8	..	..	..	..	..
(2)	Textiles	4	3	..	2	1	6	5	..	8	7	..	..	..
(3)	Lumber	2	4	3	5	1	8	7	..	6	..	..	..	..
(4)	Machinery	4	3	6	1	2	..	8	..	5	..	7	..	..
(5)	Leather	6	8	..	2	1	..	5	7	3	..	4	..	..
(6)	Chemicals	4	2	8	1	3	7	5	..	..	..	6	..	..
	Ranking order for group	1	3	6	4	2	7	5	10	8	11	9	..	..
	New Location Factors													
(1)	Foods	1	4	2	7	3	6	5	..	8	..	..	..	..
(2)	Textiles	2	3	6	4	1	5	7	..	8	..	..	..	..
(3)	Lumber	1	4	2	6	3	8	5	7	..	..	..	..	..
(4)	Machinery	1	4	6	3	2	..	5	7	8	..	..	..	..
(5)	Leather	3	6	4	2	1	8	7	5	..	..	..	..	..
(6)	Chemicals	1	2	3	6	4	7	5	8	..	..	..	..	..
	Ranking order for group	1	3	4	6	5	9	2	8	7	..	..	..	..
	Branch Location Factors													
(1)	Foods	1	2	3	5	4	7	6	..	..	8	..	..	..
(2)	Textiles	6	2	5	3	1	4	8	..	7	..	..	..	..
(3)	Lumber	2	1	3	6	4	7	5	8	..	..	..	..	..
(4)	Machinery	1	3	7	..	2	6	5	..	..	..	..	4	8
(5)	Leather	3	4	..	2	1	8	7	5	6	..	..	..	..
(6)	Chemicals	1	2	3	6	4	5	7	..	..	..	..	..	8
	Ranking order for group	1	2	3	6	4	7	5	10	9	11	..	12	8
	Entire Industrial Group (Location, Relocation, and Branches)													
(1)	Foods	1	3	2	6	4	7	5	8	..	..	..	..	..
(2)	Textiles	2	4	6	3	1	5	7	..	8	..	..	..	..
(3)	Lumber	1	3	4	5	2	7	6	8	..	..	..	..	..
(4)	Machinery	1	3	6	4	2	8	7	..	5	..	..	..	..
(5)	Leather	3	6	4	2	1	..	7	5	8	..	..	..	..
(6)	Chemicals	1	2	4	5	3	7	6	8	..	..	..	..	..
	Ranking order for group	1	3	2	8	7	5	4	6	9	..	..	..	..
	Combined order of rank, all groups	1	3	4	5	2	6	8	7	9	10	11	12	13

[a] Industrial development in the United States and Canada.

TABLE 12 SHIFT IN POPULATION

Geographical division	1930 Per cent urban	1930 Per cent rural	1920 Per cent urban	1920 Per cent rural	1910 Per cent urban	1910 Per cent rural
United States	56.2	43.8	51.4	48.6	45.8	54.2
New England	77.3	22.7	79.2	20.8	76.3	23.7
Middle Atlantic	77.7	22.3	74.9	25.1	71.0	29.0
East North Central	66.4	33.6	60.8	39.2	52.7	47.3
West North Central	41.8	58.2	37.7	62.3	33.3	66.7
South Atlantic	36.1	63.9	31.0	69.0	25.4	74.6
East South Central	28.1	71.9	22.4	77.6	18.7	81.3
West South Central	36.4	63.6	29.0	71.0	22.3	77.7
Mountain	39.4	60.6	36.4	63.6	36.0	64.0
Pacific	67.5	32.5	62.4	37.6	56.8	43.2

Geographical division	1900 Per cent urban	1900 Per cent rural	1890 Per cent urban	1890 Per cent rural	1880 Per cent urban	1880 Per cent rural
United States	40.5	59.5	36.1	63.9	29.5	70.5
New England	72.5	27.5	75.8	24.2	68.7	31.3
Middle Atlantic	65.2	34.8	57.7	42.3	49.9	50.1
East North Central	45.2	54.8	37.8	62.2	27.5	72.5
West North Central	28.5	71.5	25.8	74.2	18.1	81.9
South Atlantic	21.4	78.6	19.5	80.5	15.1	84.9
East South Central	15.0	85.0	12.7	87.3	8.4	91.6
West South Central	16.2	83.8	15.1	84.9	12.2	87.8
Mountain	32.3	67.7	29.3	70.7	23.6	76.4
Pacific	46.4	53.6	42.5	57.5	36.2	63.8

Discussion

DEXTER S. KIMBALL.[14] The paper presents in an excellent manner a belief that has been growing in the minds of many students of industry for some time. The author's presentation is convincing, not only as to what is really happening, but what should happen for the good of all concerned. Many students of industry have pointed out the manifest disadvantages of highly concentrated industry, but it has remained for this great depression to impress upon us more vividly than ever the evils of extreme industrial concentration in bad times. A visit to any great manufacturing center, such as Detroit, leaves one distressed beyond words and wondering how such calamities can be prevented. And it is little wonder that men like Mr. Ford have begun to doubt the merits of vast size and high concentration. In the writer's opinion the author is quite right in his conclusions, and it is not believed that we shall see again such vast integration and consolidation as we now have in some of our mass-production industries.

Aside, also, from these human considerations, there is good reason for believing that the economic considerations which the author quotes will receive ever-increasing consideration. While he does not refer to it specifically, he indicates that the old law of diminishing returns applies here as elsewhere. It has been generally assumed that the larger the manufacturing unit the lower may be the cost, but there is no reason for thinking that this is true indefinitely as the size of the plant increases. There must be a maximum size for any given plant beyond which costs increase rather than decrease, or else the old law of diminishing returns is false, and so far we have known no exceptions to this law.

Unfortunately, we have no quantitative studies bearing on this problem. If large industries possess data bearing upon it, they have not published them. The United States Census Reports show that the per capita value of manufactured products has risen steadily as our industrial machinery developed. But the ratio of the value of manufactured products to the capital invested has declined steadily. In 1850 this ratio was close to 2, and it has declined steadily ever since, being 1.39 in 1919. We are in need of statistical data on the relative efficiency of manufacturing plants of varying sizes, and the writer is confident that if such data could be secured they would verify fully the author's conclusions.

J. C. HECKMAN.[15] Statistics on this subject are necessarily fragmentary, and when viewed separately can scarcely be relied upon as a basis for general conclusions. But the group assembled here by the author supplement each other impressively to establish a very definite trend.

[14] Dean of Engineering, Cornell University, Ithaca, N. Y. Mem. A.S.M.E.

[15] Larchmont, N. Y. Mem. A.S.M.E.

The concentration which took place in the early days of our industrial development was effected in direct opposition to the direction of growth in the country. Until that time the development had been agricultural, and there was a continuous spreading out of the population into freshly opened lands farther and farther westward from the settlements.

Evidently, powerful forces must have come into play to arrest and in a measure reverse this flow of population. The forces which did this were the forces that caused our rapidly multiplying manufacturing plants to be established in the larger communities and in districts where their particular industries were already operating. Principal among these forces were:

1 Cheap or convenient power supply
2 Proximity to skilled labor already trained in the industry
3 Proximity to supplies of raw material
4 Proximity to local markets.

If, as the author's statistics clearly indicate, there has been a marked swing away from this movement of concentration, it must follow that some change has occurred in the strength or direction of the forces that previously favored this movement.

In considering the data presented, it must be borne in mind that they represent only new manufacturing plants or branches of old plants established in recent years. Established businesses having reasonably modern and efficient plants are held to their present locations by bonds too powerful to be readily severed. If the figures are studied in relation to the total new plants established during the period rather than in relation to our total manufacturing plants, they acquire added significance.

Consider the present characteristics of the factors already mentioned as determining the original concentration of our industries:

1 Power at a reasonable cost is no longer the possession of a few localities or confined to the larger towns and cities. The development of interconnected power networks covering such a large part of our country makes power widely obtainable at lower costs than ever before. Notwithstanding the large increase in power used per employee, only those new plants using very large blocks of power or those in which power is produced as a by-product of the manufacturing process can afford now to build and operate new isolated power plants.

2 The factor of proximity to skilled labor has also lost much of its force, partly because skilled labor is much more widely diffused than it used to be, but mainly because the development of automatic machinery, much of it peculiar to an industry or even to an individual plant, obviates the need for skilled labor, in its old sense, in production work and makes it necessary and economical for plants to train their own production employees. A distinction must be made between skilled labor and what may be called production labor. The existence of a potential supply of labor capable of being trained to efficient production, within living distance of the plant, is of course essential to all manufacturies, but this requirement may be met in a location not in a large city or town and not adjacent to other allied industries.

3 The factor of proximity to raw materials is really but one element in the general factor of transportation which includes not only the transportation of raw material to the plant and the transportation of the finished products to market, but also a frequently overlooked item involving the transportation of all the living supplies of the manufacturing community to the homes of the families of the employees. All three elements eventually enter into the delivered cost of the finished product and must be considered in the complete analysis. We are all familiar with instances of industries that originally centralized in certain districts because of proximity to raw materials and have long continued to flourish there after the raw materials have been exhausted or have been replaced by cheaper or better supplies from other districts. With the development of means of transportation, except in the case of some special industries using large tonnages of low-priced raw materials, proximity to raw materials has ceased to be a determining factor in plant location. And even in these cases it is the location on a good transportation system from a source of supply rather than proximity to the source itself that is important.

4 It is significant that Table 8 in the Appendix to the paper, giving the relative importance of the factors in plant relocation, reveals markets as the factor of leading importance in choosing such locations. With the expansion of industries into national and international distribution, local markets have receded into a less important position. Locations have to be chosen with consideration of the broader field. Prompt customer service as well as transportation costs have also become an importnat factor. As the country has become more widely settled, this factor of markets has reversed its direction and has become an important one in the dissolving of industrial concentration.

It is apparent, therefore, that the facts of industrial deconcentration that the author has brought to attention are the result of a definite realignment of economic forces. It might be added that this new alignment not only permits but aids what we hope is the natural trend of people away from the crowded dwelling places of congested cities.

The author says that the forces at work may be divided into two groups—economic and sociologic. For purposes of analysis and discussion this is a useful division. In actual application it is difficult to draw a sharp line between the two groups. As engineers we perhaps are out of our field in discussing sociology. But we have learned that sociologic factors have a profound effect on economic results. The conditions under which our employees live have a marked effect on their efficiency and ultimately on the costs of their product.

For us as engineers who may be called upon to advise on specific cases of proposed new or branch plant locations, it may be sufficient to accept the conclusions of this paper that the sociologic results of dissolving industrial concentration are beneficial to the country as a whole and be appreciative of the conclusion that they reinforce the economic forces. A sociologist might prefer to reverse that statement and put it that it is fortunate that the economic forces act in the same direction as sociologic progress.

We all know that in most manufacturing processes there exists a maximum-sized economic unit. The actual size varies greatly with the process involved. It may be small in a simple process or very large where automatic machinery of tremendous capacity is necessary for greatest economy.

In a small business where clearly only one unit is justified by the size of the business, obviously only one location can be considered. But after market studies and other considerations have settled the question of the proper district in which the location should be made, the question still remains whether the location should be in a large or small community and whether to locate near a similar industry or not. It is at this point that the factors presented in this paper call for consideration.

With larger companies the major question of branch plants at once arises. Shall or shall not branch plants be established? If so, how widely shall they be distributed? Shall they be merely assembly plants or complete factories? Shall they make a considerable portion of the company's complete line or shall they be special units making a few allied products? A host of such questions suggest themselves.

Obviously a general solution cannot be offered. Each case is a subject for special study. But in each case if separate plants are decided on, the same decisions have to be made as to the size and type of community in which the plants shall be located.

One point may be emphasized. A group of branch plants cannot be successfully operated without an efficient central modern planning, control, and administration system.

However strong the trend, it is evident that no rapid transition to a wider distribution of our industrial plants is possible. The inertia of existing conditions is too great; the capital invested in existing plants is too tremendous. But to those of us who feel the disadvantages of existing conditions, with crowded urban populations absolutely dependent, as the author puts it, on the weekly pay envelope for existence; to those of us who believe that a better balance between urban and rural population is necessary for the best interests of the country—the marked trend toward the dissolving of concentrated industries as shown in this paper is most heartening for the future.

W. S. Giele.[16] The writer has had the opportunity to observe at close range certain phases of the process of the dissolving of concentrated industries and also to contrast the operation of plants in large cities and in small communities. Points bearing directly on the paper can be illustrated by data and observations from Lebanon, Pa. The population of Lebanon county in 1930 was: Total, 67,103; urban, 32,513; rural (non-farm), 23,590; rural (farm), 10,982; employed, 26,735. The industries were:

	Manufacturing	Agricultural
Value of products	$43,276,700	$6,308,560
Capital invested	$26,041,500	$19,341,406
Persons employed	10,705	....
Compensation	$10,032,400	....

Among the plants brought to Lebanon by the Chamber of Commerce from congested metropolitan districts were plants manufacturing or processing cigars, clothing (men's), clothing (women's), handkerchiefs, hosiery, rayon, shirts, shoes, and silk.

Among the industries which originated in a semi-rural environment are the Hershey Chocolate Company, the Lebanon Paper Box Company, and the Lebanon Steel Foundry, producing small steel castings of special alloys. All of these industries found that with modern transportation and communications facilities they could remain in sufficiently close touch with their markets. The Lebanon Steel Foundry, which does its melting in electric furnaces, is an example of what the distribution of power has made possible. All of these industries find material savings in many of those items in the cost of doing business that are incidental to congestion.

However, the greatest single advantage gained by all lies in the greater economic security, better living conditions, and general well being of the workers. Improved roads and cheap automobiles have increased the radius from which workers can be drawn, and many of these plants collect workers in buses. These people have been rooted in the soil for many generations. Shorter factory hours or lack of factory employment give them opportunity for agricultural activities. They have shelter and fuel and food.

An important factor, and one seldom mentioned, is that the wider distribution of power and of mechanization has affected the farm as well as the factory. The great reduction in man-hours (and in woman-hours) required to do farm work constitutes a reservoir of potential labor supply so located that industry will have to go out to meet it. This is a labor supply that can maintain itself in times of industrial depression and relieve in some measure a part of the farm problem in that these people eat the food where it is produced.

The Hershey Chocolate Company has developed a combination of industry with agriculture that seems nearly ideal. The farms surrounding its industrial community produce the milk that goes into the product, and the farm boys and girls work in the factory.

The relatively lower cost of workingmen's homes, with the relatively greater stability of real-estate values, contributes to the sense of economic security in the smaller communities.

Lastly, the laid-off worker comes in contact with opportunity for new employment much more readily in the smaller communities than he does in the more concentrated areas—another factor in economic security.

Harry Arthur Hopf.[17] The concentration of industrial plants in cities of this country has already exceeded all limits dictated by a decent regard for living conditions, wage levels, transportation facilities, health, law observance, and the rearing of children. In the mad desire for profits the welfare of human beings has only too often been subordinated to monetary considerations. While it is of course absurd to suppose that complete decentralization of all manufacturing enterprise is an objective to the attainment of which our best energies and abilities should be dedicated, we may in all fairness assume that the highest dictates of enlightened social policy support acceleration of the trend which is discussed in the paper.

Lewis Mumford, critic of architecture, stated in one of his books substantially as follows: "We cut a steel gash in the bowels of the earth and call it a subway. Every morning we hurl from the Harlem to the Battery, within a concentrated period of time, several hundred thousands of people under disgraceful conditions of crowding. Every evening we repeat the process in the reverse direction, and that is what we are pleased to call civilization."

In the background of the entire discussion to which this paper is devoted is the challenging question, "How big is too big?" In all human undertakings we must strive to find the optimum of size, and that applies to cities, states, and governments, to industries, schools, and churches, and even to many other things. If the trend to decentralization of industry will ultimately result in more normal conditions of human life and happiness, it is a secondary consideration whether or not the highest profit possibilities may be achieved under such circumstances.

H. R. Taube.[18] The author has analyzed the economic forces and considerations which in many cases justify the transfer of existing industrial plants from congested cities into smaller communities or the establishment in them of branch plants or new enterprises. His arguments in regard to the sociological advantages of the dissolution of concentrated industries seem, however, less convincing. He mentions, for instance, "the hazards of plant-operation interruption due to concentration of large aggregations of labor," alluding evidently to labor disputes, strikes, etc. As a matter of fact, the last ten years, comprising periods of intensive industrial activity and of unprecedented depression, have been, as far as the highly concentrated large-scale industries are concerned, conspicuously free from such disturbances—a result that must be credited primarily to the far-seeing, progressive management of these industries. The news of serious labor conflicts that during these years appeared on the front pages of the metropolitan press did not come from Pittsburgh, Cleveland, or Chicago, but from the mill towns of North Carolina and from the mining towns of the soft-coal districts of Kentucky and Illinois, where they, theoretically, ought to be least expected. These labor disturbances, characterized by violence and bloodshed, have more than once required the calling out of troops and found an aftermath in sensational murder trials.

[16] Mechanical Engineer, Philadelphia, Pa. Mem. A.S.M.E.

[17] H. A. Hopf & Co., New York, N. Y. Mem. A.S.M.E.

[18] New York, N. Y. Mem. A.S.M.E.

Reports of these events in the New York "Times," a paper that can hardly be suspected of radical tendencies, have revealed most shocking living and working conditions prevailing in those small half-rural communities—conditions which compare unfavorably with those in large industrial centers.

The author, pointing out the declining rate of employment in the basic established industries, remarks that "we have in so far as spreading employment is concerned, a real problem to solve. To what extent we can solve this problem by the partnership of the land and the factory . . . is not yet clear." The possibilities of solving the unemployment problem in this way seem very remote indeed. Industries that can be set up in strictly rural districts and can rely upon the employment of farm labor available during the winter months are rare exceptions rather than the rule. This is borne out by the statistics on the fourth page of the paper. The number of industrial plants gained in the years 1926 and 1927 by communities having a population of less than 2500 is only 6 per cent, as against 60 per cent in cities of over 50,000. The benefits to be derived from gardens made available to industrial workers in smaller communities are also very limited. Taking care of these truck gardens might promote the health and morale of the workers and the produce they grow might reduce their grocery bills, but they cannot provide economic security, particularly so if the worker does not own the land and might be dispossessed at any time.

It is therefore difficult to see how the transfer of industrial plants into small or rural communities can protect the wage earner from the results of cyclical or casual unemployment, whereas in regard to technological unemployment he is decidedly worse off than the worker in industrial centers offering opportunities for employment either in other industries or in other occupations. The plight of the miners in the soft-coal districts is a good illustration. While unable to earn a living wage in an overmanned and disorganized industry, they cannot get away from it, lacking the knowledge, the mobility, and the opportunity to make their way into other occupations.

There is still another side to this question. As a rule, when a plant is transferred to a new location, only a limited number of indispensable highly skilled men is taken along, and the rest of the new labor force is recruited from local labor, part of which is drawn from non-industrial occupations. Thus the supply of industrial labor is increased, whereas industrial employment, since 1919, has been steadily declining. The final result will be an increase in the number of the permanently unemployed industrial wage earners. A speeding up of the process of dissolution of concentrated industries, advocated by the author, might therefore noticeably aggravate the already serious unemployment situation.

From the facts presented so impressively by the author and from those mentioned herein, three conclusions might be drawn:

1 The dissolution of concentrated industries offers in many cases to the individual manufacturer distinct economic advantages; we do not, however, possess at the present time sufficient data to judge how this process, applied on a large scale and carried out in a brief period, might affect industry and national economic life as a whole—for instance, to what extent the lowering of wages, which is one of the main factors in favor of decentralization and plant relocation, might narrow the market for industrial products.

2 The possibilities for sociological improvements inherent in decentralization do not come into effect automatically; better living conditions of the worker and industrial peace do not depend so much on the size of the community as on the type of management and the character of its policies.

3 Dissolution of concentrated industries does not provide a palliative for, let alone a solution of, the most pressing industrial problem society is confronted with—that of unemployment.

JOHN A. PIQUET.[19] The decentralization of manufacturing is not a theory nor an attempt at utopia. Facts show, first, that it is going on quite rapidly; second, that decentralization should increase more rapidly than ever, and, third, that its effects are beneficial to all interests. The paper brings out two of the most important factors in industrial decentralization:

1 That the small or medium-size industry can compete favorably with the large corporation. This is important, for the cause of the concentration of industry and population in large cities and in one small section of the country, the Northeastern States, has been largely the control under few hands of the technical knowledge, capital, and industrial facilities of the nation. These relatively few industrialists and financiers have by force of the limitations of time and distance desired to keep the factories near their offices—in short, in relatively few places and sections. Furthermore, large or mass-production plants have favored a location in large centers where mass labor supply, nearness to large banks, the best water and rail facilities, etc., are present. The argument that real decentralization of industry can come about through large corporations establishing myriad branches throughout the nation is not sound, as there is a limit to such branches due to the serious fluctuations in sales suffered by large corporations, the difficulty of keeping track of many branches, and other handicaps that big manufacturers suffer.

2 That the small city, with the extension to it of power, truck transport, educated people, telephone, radio, amusement, and in short most or all of the business and living conveniences once monopolized by the large cities, is now in a position to handle industrial production efficiently, while having lower land, tax, and living costs than the big cities, and hence better for labor morale and factory costs.

The concentration of manufactures in this country is very great, and has led directly to high costs in shipping raw materials long distances from every part of the nation to the Northeast factories, in shipping finished goods long distances, in the necessity for armies of salesmen, branch depots, jobbers, and swollen retail sections in every town; in huge advertising totals; in the high cost of financing this long-distance and highly competitive distribution; and finally, in labor troubles, crime-breeding slums, inefficient factories and layouts, high land and tax costs, and other evils of crowded industry in large or one-industry cities.

The Northeast, the section extending from Boston to Chicago and north of the Mason and Dixon line and the Ohio River, has held its own in industrial concentration, although its share of the nation's population has decreased, thus:

	Per cent of United States Population	Per cent of United States Manufactures (value)
1900	63	83
1927	49	71

This indicates an approximate increase of about 5 per cent in concentration, allowing for the higher value of the finished products made in the Northeast over the large percentage of semi-finished products made elsewhere. This increase in concentration is directly due to war and boom production and expansion since 1916, the quick demand for more goods being accommodated largely in the Northeast with its already large and numerous factories, and the control of war and boom orders by companies operating mainly in the Northeast.

The other side of the picture—decentralization—is shown in many figures and many trends, of which a few are as follows:

First, within the Northeast itself, highly concentrated industrial centers and state are losing production to smaller places or other states in the region. The decentralization of centers of

[19] Metropolitan Information Service, New York, N. Y.

three basic manufacturing industries in percentage of national production is shown as follows:

	State	1900, per cent	1927, per cent
Iron and steel.....	Pennsylvania	54	39
Cotton goods.....	Massachusetts	32	18
Meat packing.....	Illinois	40	22

The story back of these figures is well known. Blast-furnace and rolling-mill production gradually concentrated around Pittsburgh, then decentralized toward markets, at Chicago, Cleveland, Baltimore, Detroit, with small centers developing in Utah, on the Coast, at Lake Champlain, Troy, Boston, Newark, and of course the raw-material and market combination of Birmingham. Ala. Cotton goods spread throughout the South, for labor reasons. Meat packing extended from Chicago to Kansas City, Cincinnati, and now into every section of the country where cattle and hogs can be conveniently gathered from a radius of 200 miles, and particularly where growing markets abound. Cement, once made near Allentown, Pa., and shipped as far as Texas, is now produced in every section of the country. The same story goes for the coarser grades of paper, once concentrated in New England. Sugar refining, originally a New York monopoly, is scattered in every principal port from Boston to New Orleans. Cracker bakeries, once concentrated, are now found in all chief centers. Tin cans are now made in every large center, or near every important source of sea foods, fruits, or vegetables. The same goes for canneries of all sorts, their location being dependent on the raw material rather than on the market. Flour mills have extended from Minneapolis through to Buffalo. Minor trends which are not important yet, but of interest, are the home flour mills sold to families following the movement for using the original flour with all its food values which commercial white flour eliminates in part, and the barter movements of the present business depression, where farmers in many sections are having their own flour ground at local flour mills of small size.

The clothing and garment industries, once so highly concentrated in New York, have established over a thousand plants in small towns in the Northeast since the war, for labor reasons. Fine silk underwear, a style product, is slowly leading the way in the trend from Broadway locations (so essential to style products because of frequent changes in style and desires of visiting store buyers) to smaller cities, for labor reasons. Standard goods, like men's clothing, shirts, rayon and cotton underwear, and work garments, are well decentralized. Printing of books, monthly magazines, and large jobs is becoming a small-city proposition due to the strong unions of the large centers. The postal rates on printed matter by mileage zones have also tended to move much Eastern printing and paper patterns and other printed matter to factories in the Middle West, mainly in small cities—a central location for national distribution. Shoes and slippers are steadily extending from large cities to small towns, for labor reasons, and from their original New England home to Missouri, Pennsylvania, and many other states. Candy making follows population so closely that in addition to large plants in large cities everywhere, we also have myriad small plants, located in every town. Freshness is a factor, while the capital required to start is small. Silk weaving, tied closely to New York by selling and financing reasons, is nevertheless now a small-town industry, due to labor costs, and is scattered in localities from Massachusetts to Maryland, with Pennsylvania the leader. Full-fashioned silk hosiery is slowly scattering in small cities throughout the East, for labor reasons, while the still concentrated carpet and upholstery firms of Philadelphia are making surveys of country towns in the East and South with a view of eventually escaping the combination of high labor and high taxes. Narrow fabrics and lace curtains are other Philadelphia industries that are tending toward smaller cities.

Finally, almost every large center has extended its definitely suburban areas of industry until some of them reach as much as 60 miles out. In New York City the total gain in number of industrial employees since 1914 has been in New Jersey, adjacent to the metropolis; the actual number in the city has remained the same. Despite this decentralization, New York City and its area still remains so powerful an example of the extreme concentration of industry that here are manufactured more goods, in value, than in all New England or in all the South east of the Mississippi River.

Because industry is tending to follow population so closely, it is interesting to note that the Northeast, with its 49 per cent of population and 71 per cent of manufactures, occupies only about 11 per cent of the land area of the United States.

The South, east of the Mississippi River, has about 25 per cent of the population and 14 per cent of the manufactures in an area roughly 22 per cent of the nation, and the vast open country west of the Mississippi contains more than 66 per cent of the land area, but only 25 per cent of the population and 16 per cent of the manufactures.

There would be nothing significant in these figures if our business organization ran smoothly. However, the basic cause for industrial decentralization is the fact that there is too much productive capacity and that there are too many manufacturers each trying to sell the whole nation at once, coupled with a less rapidly increasing population and consumer demand (to say nothing of financial parasites and uncontrolled machine fecundity displacing employees), and this has led to such intensive competition for customers that most industries can no longer compete successfully for customers that are too far away nor produce efficiently in factories in congested centers. The average manufacturer today and from now on, therefore, realizes that the fundamentals of successful manufacture must be as follows:

Distribution: Knowing customers' needs, and changing products to suit; quick and cheap deliveries to customers; personal contacts with customers, and therefore customer loyalty.

Production: Employee and shop teamwork, high efficiency per man; basic low costs in land, plant, taxes, transportation facilities, and living and amusement costs for workers.

So intense is the competition, since production overtook consumption in 1925, that every one of these fundamentals is necessary for every manufacturer except those with highly superior or patented product or management and those concerns, such as newsprint, whose location is determined purely by nearby materials. These fundamentals, it is clearly seen, can be obtained only by plant locations in cities that are not congested and where labor conditions and costs are favorable; and in locations as near to the chief markets as possible. To be near customers is to have the great advantage today. There is a limit to selling and shipping radius from the plant beyond which customers are lost track of, deliveries are slow or costly, and more favorably located competitors have the inside track. In short, the strong tendency now is medium-size or small plants each covering a surrounding market region of from 100 to 1000 miles radius, depending on the line of industry, the competition geographically, the transportation facilities, etc., with a rough average of 500 miles radius.

The overdevelopment and intense competition in every line of industry will keep on increasing and will force the average manufacturer to concentrate on the market within truck or fast rail delivery of his plant, to know that market well, and to hold on to it. Steady sales, and full-time operation of a smaller plant, with capital, overhead, and expense adjusted in proportion, will enable him to make more economies, maintain full-time employment at higher wages, give more quality at lower prices, and enjoy more stable profits, than the present highly fluctuating sales volume has ever given him a chance to do.

JOHN N. JITKOW.[20] With the difficulties of the present times it has become quite fashionable to lay all blame at the door of the concentration of industry. And the proposed remedy is largely very simple: the evil lies in concentration; therefore let us deconcentrate or dissolve the concentration, and everything will be all right. Some advisers proposed to create a substantia tariff wall and to return to the status at about the time of the Civil War; others would pattern after the balanced state of industry and agriculture in France; and, finally, there are those with whom the author agrees, who are satisfied to rely upon a general dissolution of concentration.

It makes no difference how far the proposed regressive march has to go, for it is quite obvious that the advisers do not clearly understand the significance and role of the normal industrialization process, the correlation of the present-day culture with concentrated industry.

The industrialization process has been developed by all nations in similar, parallel lines—not all at the same time, but the difference being only in the tempo and rate of industrialization.

The true independent measure of industrialization can be derived by superimposing two forms of industry: one which has produced goods for immediate consumption; the other, capital or productive goods, which are now available for the future. The first embraces foodstuffs, textiles, the clothing and leather industries; the last, iron and steel, the production of machinery, and different metallic goods, including the transportation media and the chemical industry.

In every industrialization process everywhere there existed the consumption industries at the beginning, and only when capital, credit institutions, and trained workmanship attained sufficient size and strength did the capital goods industries appear on the national stage. The output of the consumption goods industries at first was 100 per cent of the total; then with the increase of the capital goods industries, there came a time when 50 per cent was allotted to the consumption and 50 per cent to the capital goods industries. This moment in the United States fell in 1909, in Sweden in 1914, and in Belgium in 1927. If we present in ratio form the output of both types of industries, we shall secure the most significant characteristic for a given moment and a given nation.

If we divide these capital/consumption ratios in the four groups[21] of: higher than 1/2.8, of 1/2.79 to 1/1.7, of 1/1.69 to 1/1, and of less than 1/1, we shall have for corresponding periods: the industrialization start, maturity, building of heavy industries, highest concentration.

Every period as an independent variable has many dependent functional values: The corresponding increase of wealth of the nations, the percentage of the working and town population to the total, and, what is most important, the situation of organized labor. In the first period we see the beginning efforts to organize local trade unions; in the second, organization on a national basis; in the third, efforts to amalgamate; and in the fourth, organized labor not yet satisfactorily expressing itself. For the most part, most concentrated capital goods industries actually are not unionized. Craft unionism supported mainly by the American Federation of Labor does not satisfy either employer or employee, nor is it adjusted to the social and economic order of the country, as opposed to the structure of highly concentrated industries. The automotive industry is the best example of this.

Without any reference as to what route the economic evolution reflected in the coming legislation will take, it is clear that a soundly organized labor movement, normally developed and corresponding to the concentrated form of industry, will greatly help to induce any possible kinds of changes intelligently and smoothly.

Observing the relation of the industrial and functional factors corresponding to each phase of industrialization, we see nearly the same picture in every country; this correlation is not the consequence of occasional causes, but only of the play of economic forces expressed on the stage of industrialization. In his present solution, the author, like most interpreters of our present-day difficulties, has mixed up the original cause (the coincidence of cyclical low point and the war aftermath[22]) with concentration which has nothing abnormal about it and on which the whole present culture is exclusively based. To deconcentrate industry and the whole social and economic structure is more impossible than to bring back the flow of a river.

One point in the paper is of great value—the statement that "the engineer must become economic minded if he is to play his part well and serve society as it should be served." In this categoric "must" we see the desire to bring together two most important branches of modern life, engineering and economics. There lies the center of gravity of the future order or disorder.

Another man, a great engineer and scientist, the last winner of the Nobel prize, Dr. Langmuir, gives more precisely the reason for our difficulties. It does not lie in concentration, but simply in the fact that "applications of physical sciences have gone ahead faster than the development of economic sciences. Our economic system does not enable us to profit efficiently from the gains contributed to industry by the discoveries and inventions of science." This is the actual difficulty. The physical and engineering sciences are already in an established maturity, while economic sciences are still in their infancy. In the nineteenth century we saw a most efficient growth of the first and an abnormal development of the second. Physical and engineering sciences took a decidedly quantitative-experimental course, while the parallel development of economics lagged under the strong influence of ethics and the fluctuations of philosophic theories, occupied with qualitative determinations, with only rare quantitative excursions in the field of statistics. Since the example of Adam Smith, whoi ntroduced a "homo oeconomicus," there have been only abstract concepts such as "owner," "employer," "worker," and the like; very seldom actual figures and relations. Any marked change can be observed only after the words of A. Marshall, who, twenty-five years ago, in the Royal Economics Society, said that "qualitative analysis has done the greater part of the work in economics, and the higher and more difficult task of quantitative analysis must wait upon the slow growth of thorough, realistic statistics."

The task has not been an easy one, but such men as Prof. W. Mitchell have greatly advanced the behavioristic, quantitative approach in the economic sciences, which produced general and irrevocable change in some of the most important factors of legislation of all the civilized world. But the gap between economics and engineering is still a great one, and there, and only there, are located all of our present troubles.

In all false reasoning there always exists a trick, a jump in logical developments of argumentation, usually not seen in perspective at the first glance. This leap one may find easily in the author's thinking. He sees "light and happiness" in the dissolving of industrial concentration executed in the form of many smaller plants, the location of which "will be chosen with a scientific regard for the factor of distribution." He does not open the secret of "scientific regard," explaining only that this will be done in connection with "scientific marketing." In submitting the program of dissolution and organization of the smaller "scientific"

[20] Mechanical Engineer, New York, N. Y.

[21] By such division we may easily find the real location of the much-praised balanced French industry. Its present situation, with the ratio 1/1.5, corresponds to the American of 1866.

[22] Which in the more primitive state of post-Napoleonic wars desynchronized and threw out of rhythm the economic system for more than 20 years.

units, it is understood if not expressed that the dissected industry has to have static stability.

This presumption is a major hiatus in the author's reasoning. Life is motion, and industry is its most dynamic expression. Life and industry are not a pond, but a real, flowing river, with necessary adaptations of form, speed, temperature, amount of flow. The large city with its resources and reserves gives opportunities for permanent reconstruction, for competition of ideas, and a change of production methods. Most of the inventions that have become the closest attribute of our present-day life and of our high standard of living, even the human material vividly expressed in the type of Edison, Langmuir, and Ford himself, are the direct product of concentration.

If we will analyze the relation of the proposed dissection of the various types of industries, we may see that the consumption goods industries could be dissolved according to the author's plan because—

1 The methods of production there do not undergo such revolutionary changes as in the capital goods industries
2 They are not the attribute of exclusively large units (if we take the European example, we see that most of the consumption goods industries exist on a most decentralized basis)
3 There is the comparative stability of distribution.

As to the capital goods industries, they are not able to exist or even to survive in dissected form.

The normal way lies not in the dissolution of capital goods industries, but in higher concentration. "More machines and increased application of science are inevitable and highly desirable, provided the set-up is such as to enable all to share the benefits," says Langmuir.

While the engineering side gives us a complete account of what we have and what we can expect, the economic is comparatively obscure, undermined, and befogged. The dissolution of industry, even in bringing much lower efficiency and higher cost of production, will never solve the problem. The factor of greatest confusion and complexity in our situation lies exclusively in oversupply of human material and commodities.

Of the two oversupplies, the human is of greatest concern and is closely connected to the possibility of grave consequences. There is no doubt that the workers will benefit by the bettering of his standard of living by the introduction of gardens and the like. The occasional application of the Ford plan ("one foot on the soil and one foot planted on industry") can be very beneficial, but there rises a very serious question: To what extent can this measure be practically and not theoretically introduced on a large scale? It is unreasonable to expect to find the solution of our problems in the half industrial worker, half farmer. It can be a local palliative, but not the general cure.

Author's Closure

The author is indebted to those who have brought out in their discussion the various phases and aspects of the subject that could not be included in a paper of this character, primarily by reason of space restrictions, and also for calling attention to matters which the author overlooked. He is indebted to Dean Kimball for bringing out so clearly the operation of the law of diminishing returns; to Mr. Heckman for pointing out the operation of the change in emphasis, the relative importance and the realignment of the primary factors and forces influencing industrial-plant location—he is quite right in calling attention to the fact that as a practical matter it is difficult to clearly distinguish at all times between economic and sociologic forces; to Mr. Giele for pointing to the greater economic security of the workers that has been developed by several industries in Lebanon County, Pennsylvania; to Mr. Hopf for calling attention to the necessity for a proper balance between the desire for profit and the welfare of the employed; to Mr. Taube for rightly calling attention to the fact that we do not now have at our command the necessary basic data to forecast what the results might be if a large-scale redistribution of industrial units was to take place, and furthermore that the dissolving of concentrated industries of itself will or can supplant the effects of sound policies and righteous dealing on the part of management; to Mr. Piquet for pointing out that sound marketing conditions will in the future be the predominant force in industrial-plant location; to Mr. Jitkow for emphasizing the fundamental economic differences between the capital-goods and the consumptive-goods industries, and that the latter are probably more readily amenable to forces of redistribution than the former.

The author has been credited with advancing a plan for the dissolution of concentrated industries. Such is not the case. He has endeavored to collect and analyze much of the available material, so as to point out the trends and the forces at work; and he hopes that, with this paper and discussion as a base, a sound literature on the subject will be built up so that management can compare their individual situations with the general trend and thereby keep in step and in line with the times.

MAN-55-4

Planning and Scheduling

By WALTER N. POLAKOV,[1] NEW YORK, N. Y.

The process of planning presupposes an aim to be attained, and therefore it is a purposeful action. It may or may not have a clearly conceived means for this achievement or a clearly determined order of procedure. These two elements, the means or facilities for the achievement of the aim and the order of their utilization, determine the quality of planning. The author analyzes the requirements, cites the work of Taylor and of Gantt, lists the preferred methods in use, and reaches definite conclusions, summing up with: Planning is a reversible process tending to reduce the element of random.



THE process of planning necessarily presupposes an aim to be attained; therefore it is a purposeful action. But it may or may not have a clearly conceived means for the achievement of the aim, and it may or may not have a clearly determined order of procedure. These two elements, the means or facilities for the achievement of an aim and the order of their utilization, determine the quality of planning.

If the means or facilities for securing the desired result are undetermined and the order of procedure is not defined, the probability of its attainment is extremely remote, depending on the chance offered by the uncontrolled circumstances.

If the means for securing the desired results are determined, but the order of their exercise is indefinite—i.e., left to chance and not subject to the influence of planning—the degree of probability of securing the desired results is higher.

If the means for securing the desired results are determined and the order in which these means are brought into play are likewise determined, then the probability of attainment is very high.

If the order of events is determined as to the place and time (sequence), but the selection of means is left indeterminate, the obtaining of the desired results depends upon a chance offering the opportune means.

If the order is indeterminate, but the necessary and sufficient means or facilities for securing the desired results are present, the successful outcome of such a planning is very improbable

If the order of procedure is determined as to the place and time (sequence), but the means or facilities are only partly determined—i.e., if the means selected are necessary but not sufficient, or one or more elements are beyond our control—the degree of probability of attaining the aim depends upon the factors not fully under control.

Thus the perfection of a plan increases with the decrease of variables or indeterminate factors, and analysis of sufficient conditions is the essential requirement of planning. Since, however, in the complexity of life, to foresee all possible influential variables is practically impossible, most planning is typified by cases wherein either the order of procedure or some of the means are left either to the uncontrollable economic forces or to the choice of an executive. Hence most of the industrial planning is subject to the readjustments during the execution, and therefore must necessarily be augmented by continuous and graphic records of progress in order to observe the deviations and decide upon the corrective actions.

This interdependence of planning and executive work is of extreme importance in practical life. The full appreciation of this fact will establish a proper relation between a plan and the executive action. Therefore the technique of planning and the method of presenting data for the guidance of the executive actions should form a united whole in any system of management. The process of industrial, or more specifically production management is therefore neither merely planning of industrial activities nor merely execution of plans, but it is a continuous adaptation of a plan to the variable circumstances and influencing the circumstances to meet the requirements of the plan.

Hence, planning of the means or facilities for efficient production alone is inadequate for any degree of certainty in management. For instance, a well-planned maintenance and the carefully devised "trademarked" systems like "Manit," "Bedaux," etc. could produce a measure of good only if the order of work is well taken care of either by the nature of the process or by the supplementary efforts of the management.

Likewise, if the planning be limited to the establishment only of the order of procedure, such as may be found in conveyor-type assembly shops or continuous operations or elaborate semi-mechanical and semi-clerical planning boards, there could be no assurance of efficient execution if the means or facilities are receiving inadequate or spasmodic attention instead of being subject to a well-planned care.

Taylor's earliest description[2] of planning enumerates 17 leading functions. From the perusal of his list it is apparent that the stress was laid on the conditions or facilities for a planned production, while the order of the planned procedure assumed a secondary position. Of all these 17 "leading functions," only one directly refers to time of delivery. The emphasis was placed on time studies on the assumption that, if every piece of work is done in a specified time, the whole will be done as expeditiously as possible. That this is not the case is evident from the consideration that some parts may be manufactured rapidly, but in larger or smaller quantities than others, thus producing an unbalanced flow of parts into the assembly.

As far back as 1888, Gantt realized the paramount importance of the element of order in the planning, and in the steel-casting

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Note: Statements and opinions advanced in papers are to be considered as individual expressions of their authors, and not those of the Society.

[2] F. W. Taylor, "Shop Management," p. 111.

department of the Midvale Steel Company he developed the graphic daily balance, which he described years later as applied by him to the Schenectady shops of the American Locomotive Works.[3]

Time keeping and timing the workman had been done long before the industrial era. As men were paid by the day, it was important to keep track of the time they worked and of the amount of work done.

With the speeding up of our industrial tempo, the time for completion of an order began to play a dominating part in the process of planning. In the past a shop order usually stated only what was to be done, but Gantt realized that under the changed economic conditions the final date alone would not suffice. Planning had therefore to include not only the means of doing the work, but also the time order (sequence) of doing it, thus:

1 What is to be done
2 When it is to be done
3 Where it is to be done
4 How it is to be done
5 How long each detail may take.

To inaugurate such a system, it was necessary to have a running balance of finished parts and to follow daily the progress of parts in the course of manufacture as they passed through the shop. Such "planning as a whole" has a structure of order in time. The functions of a manufacturing (planning) office, according to Gantt, may be enumerated, in general, as follows:

Handling material: (1) Purchase of material, (2) custody of materials and stores, (3) what shall be done to the material (drawing and specifications), (4) when it shall be done (schedules), (5) movement of the material through the works (routes), (6) care of finished product.

Handling men: (7) How the operations shall be performed on the material (use of equipment, instructions to workmen), (8) what compensation shall be awarded for the work.

Since it is obvious that the last two functions cannot be properly performed unless functions (4) and (5) are well planned and properly executed, the attention of planning must center around the order of progress in time (schedules) and space (routes).

The order in which the various operations are to be performed being determined as a part of planning, the next important function is to see whether the actual progress of work follows the route on time. The early device was Gantt's graphical daily balance in manufacture. It consists of a sheet of paper with vertical columns listing the parts to be manufactured and the operations to be performed on each. The horizontal lines represent the consecutive dates. We thus have jobs on abscissa and time on ordinate. In each column are entered the number of pieces finished daily and a cumulative number of pieces finished to date. The date on which all parts on order are to be finished is underscored beforehand in red, and thus offers a visual warning. If the numbers continue to appear below the line, it indicates that the production is behind the schedule.

Gantt later devised what is now called the "Gantt chart." This new device for planning considers time, space, and matter as "an event" and uses one scale indicating at once by a single line the sequence of operations (order), the quantity of product, and the time of separate operations as well as the dated completion of an order. The construction of the Gantt chart is well known; its typical features are:

1 The multiple scale, such as: Time and matter, time and energy, time and space, time and ratios of any kind, etc.

2 Comparison of the preplanned task with the record of its performance.

[3] "A Graphical Daily Balance in Manufacture," by H. L. Gantt, Trans. A.S.M.E., 1903, vol. 24, p. 1322.

Unless both of these features are incorporated in the chart, it is a fallacy to term it a Gantt chart.

Planning of order is of little use unless it is organically tied with the records of performance. Once this is accomplished, however, the process of planning ceases to be a theoretical problem which, once solved, becomes a law unconcerned with its enforcement and having no part in the judgment of transgressions. To the contrary, it becomes a basis for judgment and it determines corrective actions.

Planning, as a whole, involves the responsibility for the fulfilment of the plans. This fundamental principle of management makes planning a dynamic process instead of a static set of demands. To make the planning process a dynamic force and an active tool of management, there must be an inseparable interconnection established between the task set by a plan and the record of the progress made and the handicaps encountered, as shown by charts.

The mechanism of this interconnection, as offered by the modified method of chart analysis, permits a continuous (daily or for every shift) control of the progress of the work and the correction of troubles. The simplest form of this mechanism consists of a progress chart on an output-time scale, supplemented by man-record and machine-record charts. As the short bars lagging behind the date of observation indicate delays in the performance and the causes of these shortcomings, these charts afford the opportunity to readjust the means or rearrange the order in time to avoid any serious disruption of plans.

In the typical industrial establishments the planning of production must coordinate and plan the following means of getting work done:

1 Material procurement
2 Storeroom service (raw, semi-finished stock)
3 Toolroom service (tools, jigs, templets, dies)
4 Maintenance department (equipments and building)
5 Power department
6 Safety, sanitation, welfare
7 Employment and personnel department
8 Training department (job studies, tasks, instructions)
9 Dispatching and transportation
10 Payroll and cost keeping
11 Engineering department (drawings, specifications)
12 Schedules (sequence of processes and operations, routes, process charts, time allotment)
13 Shipping
14 Record and chart keeping.

These means, in order to be of full value to production, should be used at a definite time, in a definite place, and in definite quantities. The establishment of an order in which these means should be used to obtain the optimum results is the function of planning proper.

It is obvious that the planning may be started before all the means are correctly organized and exhaustively studied. Indeed, until the planning of order is installed, there is no certainty as to which of these functions impede production. On the other hand, some planning as to sequence of operations, means required, time needed, etc. can be started on the basis of the available common knowledge and past experiences.

It is not disputed that job standardization and various forms of incentive payment have their place in the practice of management. Yet these means do not constitute a system of management, and under no condition should they be installed before planning of the work order is installed and is functioning well. If, however, the order of management installation is reversed and the tasks are set or piece rates are fixed before the planning of the work order and Gantt charts have their effect, then it will soon

become necessary to revise the tasks and methods of payment again. Besides the duplication of expense, such a topsy-turvy procedure is likely to produce demoralization and lead to the restriction of work output by keeping the day pay within the old limits.

In order to find out which of the means of production are improperly utilized or poorly functioning, it is necessary to map out the sequence and the order of procedure in time, and then to enter on this map daily the course covered. In this manner the attention of an executive is immediately fixed to those elements which create delay or cause some trouble. The progress chart will indicate whether the parts needed are delivered on time and in required quantities, and if some deliveries are behind, the symbols will indicate the reason why and lead to the correction of it. The machine-record chart will show whether the equipment is properly utilized, and if not, why not. Both these charts will locate the more serious troubles and point out the remedies. Such a study will picture the activities of the management in making the proper use of the facilities and means at its disposal. Later on a third chart, a man-record chart, should be started to bring within the scope of vision of each workman and his foreman the troubles that handicap his work and thus finally "remove the obstacles which confront the workman."

When at least three basic forms of charts are kept for some time, and the most important weaknesses or functional disorders of an organization are thus disclosed, it is possible to install a more inclusive planning. While the elements subject to planning are essentially the same in every type of industrial establishment, the principles underlying the planning functions are independent from the nature of the product or the scope of production. In every case, in order to have the product ready in time, production planning must see that:

(*a*) Needed material and supplies are available when wanted, where wanted, and in quantities required
(*b*) Needed equipment (machines, tools, power, transportation) is available in the best working conditions and fully utilized
(*c*) Needed men are available and provided with necessary instructions and rates. Then, according to the specifications and sequence of technological processes
(*d*) Schedules of work for men and machines and movement of material are worked out.

The nature of the product and the quantity in which it is produced will determine the character of planning organization. There are two types of industrial enterprises:

1 Manufacturing one or few types of product in large quantities (mass production)
2 Manufacturing on individual specifications in relatively small quantities (production on order).

The typical instances of the first type of manufacturing establishments would be textile or paper mills, large automobile plants, sugar mills, etc. The typical instances of the second type of manufacturing establishments would be a locomotive shop, a printing plant, tailoring establishments, etc.

The organization of a planning department in an establishment for mass production of a single product will be concerned chiefly with the control of materials and men. On the other hand, the organization of a planning department of a jobbing or individual-order plant will be chiefly concerned with routing and dispatching of separate orders and planning of the load for equipment, in addition to the control of supply of material and labor. This shifting of the emphasis either on means or on order will determine the structure of the planning office and the divisions of functions.

Evidently a large number of manufacturing establishments will not fall definitely into either one of these two classes, as for instance a machine-tool-building plant manufacturing a standard type of tool as well as taking orders on special machinery, or a garment factory manufacturing only two types of overalls, etc. In other words, of the two features of a manufacturing establishment, volume and variety, dominating importance may be shifted, presenting four types:

1 Large volume production of a single article
2 Large volume production of a variety of articles
3 Small volume production of a single article
4 Small volume production of a variety of articles.

Correspondingly, the structure of planning will tend to give attention to the planning of the means with increased volume and to the planning of the order with increased variety of products. In an extreme case of mass production of a single type and size article, the planning of an order will become automatic.

A mechanism of preliminary planning of an order may be clearly exemplified in a case of a machine shop manufacturing machine tools in series of widely different types and designs. In order to reduce to a possible minimum the variety of elements entering the finished machine, the parts, such as gears, cranks, pumps, flexible couplings, bearings, etc., were standardized so that they may be used for any type or size of machines. Other parts were made in as few as possible "normal" designs to fit likewise most of the machines. A designer, upon receipt of an order, works out a set of detailed working drawings, incorporating nothing but normal and standard parts of suitable dimensions, and designing individually only such special parts as probably cast frame or base.

Next, the production superintendent, together with the designer and in consultation with the various foremen, works out the sequence of processing as to planning, drilling, milling, polishing, etc., designating the class of factory equipment best fitted for the work and indicating also the possible substitute machines in case no machines of the preferred class are available at the time in the shops. The list of "normal" parts, a set of working drawings, and the manufacturing schedule with the classes of machines are then referred to the planning department.

The material-planning clerk or the balance-of-stores clerk compiles the cards for each individual part to be manufactured, specifying the following data, on a material card of a suitable design, for each element:

1 Part number and name
2 Stores symbol
3 Grade of material
4 Size
5 Weight per piece
6 Requirement (number of pieces, weight)
7 Balance on hand
8 Amount to be ordered
9 Expected delivery (name of supplier, quantity ordered, date of delivery)
10 Note surplus or deficiency for order.

From these material cards a summary of "material wanted" is prepared, tabulating each kind of material needed for the order, and is sent in triplicate to the purchasing agent and to the balance-of-stores clerk in the manufacturing office, so that they can make the necessary arrangements to avoid shortage. The third copy goes to the production superintendent or foreman.

The machine-load clerk at the same time prepares the cards for each part necessary for the order from the data furnished by the production superintendent and engineering department, showing on each card:

1 Part's number or symbol
2 Numbers of operations, consecutively
3 Manufacturing department
4 Shop
5 Numbers of machines to be used
6 Time allowed on each machine
7 Total time needed (all machines or men)
8 Notes (substitute machine or ready parts in store).

The personnel clerk, upon receipt of these cards, makes a summary of labor hours needed for the different classes of work, estimates the labor cost, and makes the necessary reports as to adequacy, excess, or shortage of the present force on the payroll.

The cost estimator, upon receipt of both material cards and machine cards, works out the unit cost from them for each part in every department. Thus the estimate, for instance, for part 74 (100 needed) will be:

Material	$0.03	$ 3.00
Labor, Dept. 4	0.01	1.00
Machine, Dept. 4	0.03	3.00
Labor, Dept. 6	0.025	2.50
Machine, Dept. 6	0.02	2.00
Total	0.115	$11.50

(Stores, $3.00; Dept. 4, $4.00; Dept. 6, $4.50.)

Finally, the manufacturing office, which is the planning department proper, receives the worked-out preliminary requirements for materials, machines, and men needed for the order (cards and summaries), together with the form order, drawings, and specifications. The first duty, then, is to consult the load chart and to reserve the time for the use of the machines needed. The date of delivery to the assembly floor is thus determined, and from it the delivery date is fixed. If it is a rush job and the date thus determined is later than wanted, the departmental layout charts or machine-record charts are consulted, and, in cooperation with foremen, readjustments are made to meet the promised date. Once the final date of shipment is thus fixed, it is entered on the progress chart by placing a closing bracket two days ahead of the promised shipment, allowing time for the final inspection, testing, and boxing.

Inasmuch as the assembly and erection of machines is made on the schedule of so many units per day, the finished parts are drawn by the assembly department from the finished-parts store. This is the most vital point for constant supervision, and the input-output chart is kept indicating the daily receipt of finished parts in the store and their issue to the assembly, seeing that a safe balance is always on hand.

It was found that detail planning and despatching can be best made, not in the central planning office, but in the shops by their foremen. Every afternoon the foreman ascertains whether new orders were received and what work is ahead. Consulting his machine record chart and layout chart, he determines what work he can do tomorrow and, accordingly, prepares the "daily order of work," listing all the work he is prepared to do opposite each machine in the shop, thus:

Machine number
Order number
Description of part
Tools, dies, templets, etc.
Operation
Quantity (or weight)
Start
Finish
Running
Cause of delay
Postponement.

Upon approval by the superintendent of production of his plan to run the machines as indicated, the foreman passes the "order of work" to the respective dispatchers in his shop, who accordingly make the necessary entries on their layout charts, and also secure material and assign men. The stores-issue slips and time cards are prepared beforehand by a clerk in the foreman's office.

The shop dispatcher, having his desk and bins located centrally in the shop, continually observes whether the work issued to the men is returned within the time allotted as marked on the layout chart, and if not, he makes inquiries as to the reason why, enters the symbol on the chart, and refers the matter to the foreman; then he extends the time for the completion of the job if it does not interfere with another job previously scheduled, and if it does interfere, he consults the foreman. When the work is completed, the dispatcher issues the next job, stamps the time card, and makes the necessary entries on the layout chart.

Toward the end of the shift, the foreman examines the layout charts and notes the progress made and the work that is behind the schedule. This enables him to prepare the daily work order for the next day with the full knowledge of the available facilities. If some of the work cannot be done as planned and the foreman is unable to remedy the trouble himself, he notifies the superintendent of production by sending a "report of delayed work," requesting his executive action.

In different shops it will be found that the layout charts are to be made on different bases. For example, in a machine shop or forge shop the work will be laid out for each machine, while in a plating department it will be preferable to lay out the work by orders, since each part must pass through several operations such as soldering, plating, relieving, polishing, etc. In every case, however, the aim of shop dispatching is to schedule the flow of work so that:

1 Every machine has work assigned to it
2 Every machine is provided with materials and tools
3 Every machine is provided with an operator
4 Every order is started on time
5 Every order is finished on time
6 Every part is moved from operation to operation
7 Every part can be located at any time
8 Every defect or delay is accounted for
9 Time taken is accounted for
10 Time lost is accounted for.

The proper use of the layout charts, based on the foreman's daily order of work, is the principal guide for this work.

In this manner planning is no longer centered in the main office, divorced from the shops, but permeates the organization. The central planning department does not presume to know as intimately as the foreman does all the minute conditions in the shop, but gives the foreman authority (within the general requirements) to do his own planning and makes him assume the full responsibility for the work over which he has control, instead of laying the blame on the main office for asking him to do impossible things. The workers, likewise, are given an opportunity to do the planning within their scope of competence, aided by the information contained in the instruction cards and limited by the time allowed. The use of the man-record charts both stimulates the efforts to excel and reflects the ability to plan the use of the means and facilities under their command. They also reflect the efficiency of the foreman in planning the sequence and conditions of work for his men and removing the obstacles that are in their way.

Budgeting is not a substitute for planning, but is an interpretation in financial terms of a plan already prepared. When the manufacturing schedules are laid out and the required capital investment, obsolescence reserve, and fixed charges are deter-

mined, the operating, maintenance, and administrative expenses can readily be expressed in monetary terms and balanced with the revenues anticipated for the given output and time taken as a frame of reference. The budget progress chart admits the same technique of graphic presentation and control as is established for the production control. Operating control of budget may be laid out on a departmental basis if a single product is manufactured or on the unit of product basis if a variety of articles are planned for production.

Similarly, national economic planning depends on the same principles and technique. It begins with an estimate of the total bulk of consumption anticipated for a period under consideration. Then the stocks are balanced to the period needed for the new product to reach the market, thus establishing the beginning of new deliveries to consumers. Production volume is then allotted among the manufacturing establishments, considering the centers of consumption, material sources, and productive capacities of the respective plants, and readjusted, giving preference to the more efficient productive units. Budgeting, coordination with outer sources of supply and export, building up of reserves, and control of performance are then handled by a graphic method on the basis of weekly returns.

Planning and scheduling conceived along these general lines presents not only a logical structure, tying into one harmonious whole (*a*) mapping the procedure, (*b*) recording the progress, and (*c*) executive direction of the deviating forces, but also a structure that functionally corresponds with our nervous system. In the old conception of planning: "All possible brain work should be removed from the shop and centered in the planning or laying-out department, leaving for the foremen and gang bosses work strictly executive in its nature. Their duties should be to see that the operations planned and directed from the planning room are promptly carried out in the shops."[4] This creates an undesirable domination of a static planning room over the dynamic shops.

In the human nervous system it is found that the central part of the brain (thalamus) has a direct connection with the sense organs and locomotion, while the "gray matter" of our brains (cortex) has no direct contact with the outside world, but receives all the impulses through the thalamus. Hence, these are well-coordinated activities of a normal person.[5] In our conception of planning there is structurally a similar organization; the dynamics of production through the records and charts (sense organs) has a direct contact with the lower centers of planning, the higher centers of planning receive the informations and then coordinate and influence the executive reactions. There is no gap between the "thinking" and "doing without thinking," hence the organization works smoothly, responds precisely, and does not create dissatisfaction or resentment against the "task masters."

Conclusions

(*a*) Planning is a process of determing the structural order of actions in a four-dimensional space (time-space).

(*b*) The process of planning aims to specify the conditions that are necessary and sufficient for performing the desired task.

(*c*) The analysis of conditions differentiates between "facilities" and "order."

(*d*) An established "order" determines the necessary and sufficient conditions for the efficient use of "facilities."

(*e*) The established "facilities" do not determine the "order" of procedure, neither are they sufficient for securing the desired result.

(*f*) An analysis of the elements of planning depends upon continuous information as to the progress of the previous plans.

(*g*) The instrument of analysis and planning is a chart which places the elements in order structurally corresponding to the reality (establishing what should be and plotting and measuring the deviations of what is, similar to the navigator's map that pictures the course to follow and records the progress and deviations made).

(*h*) Therefore, planning is a reversible process tending to reduce the element of random.

[4] F. W. Taylor, "Shop Management," pp. 98–99.

[5] "Science and Sanity, an Introduction to Non-Aristotlian Systems," by Alfred Korzybsk, Science Press, 1933, especially supplement 3, offers the most valuable method for solving problems of structure.

MAN-55-5

Management Policies and Practice in a Continuous-Production Gray-Iron Foundry

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In a continuous-production gray-iron foundry with a capacity of 250 tons daily, chiefly making small castings, the management policies and practice have been directed toward obtaining maximum results by the continuous molding-pouring method with the minimum expenditure of time and effort. The operation of this continuous foundry is described in detail, through all the processes from handling the raw materials to the final inspection of the product.



IT CAN BE considered an axiom that the success of an industrial enterprise depends on the coordination of its management, mechanical, and technical branches, so that maximum results are obtained with a minimum expenditure of time and physical effort. There are few branches of industry in which this is more evident than in the foundry, where the repeated handling of the materials and the product forms the major cost of production. It is the purpose of this paper to discuss how a foundry with a maximum capacity of 250 tons daily, engaged chiefly in the manufacture of small castings, has attempted to solve the problem by the continuous molding-pouring method and to mention some of the details that have been found necessary to assure the success of that method.

DESCRIPTION OF THE FOUNDRY

In order to understand clearly the management policies employed in this foundry, it will be necessary, first, to describe briefly the foundry itself and its operation, and later to coordinate this with the management practice. The foundry as a whole is divided into four departments: the jobbing foundry, where castings for machine tools and equipment used throughout the factory are made, the hand-molding foundry, the non-ferrous foundry, and the continuous foundry. This latter foundry is the one that will principally be discussed, and the one where approximately 85 per cent of the total tonnage is cast.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The melting equipment consists of two cupolas of a capacity of 250 tons each, two of a capacity of 180 tons, and two of 90 tons. One cupola of the size necessary for the tonnage required is used each day, the second one of the same size being prepared for operation on the following day. Each cupola is served by two charging machines located opposite each other, and the coke, pig iron, scrap, and sprues are charged in the order indicated throughout the working day.

The continuous foundry is a concrete structure, designed for this specific purpose and for the machinery to be installed, and is conveniently located to receive raw materials by rail or boat. The building may be considered as two abutting buildings, which for the purpose of identification will be called the east section and the west section. The east section is three stories high, while the west is four, although the ground floor and the upper floor of each section are on the same level. The molding and shaking out of the castings, as well as the sand preparation, are performed in the east section, while the cleaning, filing, grinding, and polishing are accomplished in the west section. In a northerly southerly direction the east section may be considered as divided into six sections designed for installation of six independent mold-conveying and metal-pouring units. Five of these units have been installed, and the framework is in place for the sixth, should it be required. Each unit is complete in itself and consists of a battery of eight hydraulic-molding presses, with its own mold-conveying, sand-preparation, and distributing machinery.

The molding is done on the ground floor, the mold is conveyed to the pouring station, also on the ground floor, and is then conveyed to the west end of the upper third floor of the east section, where it is shaken out, the empty flasks being replaced on the conveyor and returned to the molding floor.

HANDLING RAW MATERIALS

Raw materials are received by boat or rail. Cranes, equipped with magnets and clamshell buckets, unload all incoming material, transferring the pig iron, scrap, coke, and limestone to a designated section of the raw-material yard, while the sand is delivered to inside storage bins. The raw-material yard is exceptionally flexible, being well supplied with industrial tracks in order to facilitate the loading of the materials as quickly as possible and with a minimum amount of handling.

Definite weights of coke and iron are made up on the ground floor and are delivered by industrial charging cars to the charging platform by elevator. Sprues and gates are continually being returned from the shake-out stations of the continuous foundry via chutes and conveyors to the charging cars, and later weighed and delivered to the charging platform, which is located on the same floor level. In this way the hot sprues are returned to the cupola less than one hour after the molten iron has been poured.

The blower room is located on a mezzanine floor adjacent to the cupolas and between the charging floor and the foundry floor proper.

For continuous operation it is important that the metal be delivered regularly to the mechanical systems. The molten iron, in 500-lb ladles, is delivered to each molding unit by monorail cranes that pass directly in front of the cupola, delivering continuously throughout the day. The metal is then emptied into 50-lb

hand ladles, and the molds are poured, while the operator stands on a secondary conveyor synchronized to the speed of the mold-conveyor unit.

The slag from the cupola is granulated in order to handle it effectively. Water under pressure granulates the slag as it flows from the slag spout, carrying it along cast-iron ducts leading to a reservoir. At the end of each operating day a clamshell bucket removes the slag to open-top cars.

Sand Preparation and Distribution

The preparation of sand for 20,000 molds each day is an important item, especially when it is considered that the castings must have a very smooth surface. The operation of the sand-handling and distributing system is continuous during the working day, and all its units are coordinated so as to obtain maximum efficiency. The sand is reclaimed, screened, tempered, cooled, aerated, and delivered to hoppers directly over the molding machines.

Replacement sand, either in the form of reclaimed sand or natural sand, is fed to the mechanical systems through a gravity-discharge bucket elevator, delivering it from the foundry floor to the system sand storage located on the top floor of the building.

Sand from the shake-out is cleaned by passing over a reciprocating screen and magnetic separator, which removes all iron shot, sand pellets, and other foreign matter before it falls to the distributing belt. The backing sand is distributed uniformly on the mixing and tempering floor below by the aid of an automatically moving plow located over the distributing belt, and, after tempering, the sand is mixed and delivered to the backing-sand hoppers located directly over the molding machines.

Facing sand is made from mulled backing sand with the addition of a small amount of either reclaimed or natural sand. A battery of three mullers is used for this purpose, each equipped with suitable bucket loaders and discharging doors for quick action. The backing sand used for facing is delivered to a sand storage over the mullers by a cross-belt conveyor and bucket elevator from any one of the main distributing belts. After tempering and mulling, the sand passes through a centrifugal aerating machine to storage hoppers for delivery to the various facing-sand chutes.

Green core sand is mulled and delivered to the core machines in exactly the same manner as the facing sand.

Mold-Conveying System

It is important in a continuous system of this nature that the speeds of the various conveyors are properly synchronized. In the system described, the speed of the mold conveyor is the controlling factor, and is adjusted to suit the mold requirements by reducing or increasing it as the work varies.

The mold-conveying equipment consists of a series of trays suspended from an overhead conveying system. Four molding machines are located on each side of the mold conveyor, with a total of eight machines to each unit.

After pouring, the molds are conveyed to the shake-out, which is located on the third floor of the building. Here the castings are removed from the sand, and the empty flasks are returned to their respective trays. The sprues and gates are removed from the castings and delivered to industrial charging cars by a cross-belt conveying system; the sand falling through the shake-out grate bars is returned to the tempering floor as previously indicated. The average time cycle for a complete revolution of this unit is 20 min, with an elapsed time of 10 min from the pouring to the shake-out stations.

The jarring of the sprues and gates in falling to the conveyor loosens the sand on them, and this passes through a screen to the sprue-sand receiving storage for reclamation.

The operations of pouring, shaking out of the molds, removal of the castings, and the return of the empty flasks are accomplished while the mold-conveyor system is in motion, all operations being performed at a uniform rate so that no time is lost.

The fact that the various units of the system can be operated independently and that the speed of each unit can be controlled within reasonable limits adds the desirable amount of flexibility to the system.

The sand that accumulates in the sprue-sand storage is reclaimed by passing it over an electric vibrating screen. This separates the sand into two grades—that retained on the screen is discarded, while the other, which is the larger part, is collected and delivered to the system storage, and later bonded with finely pulverized clay.

Core Handling

The cores made in the foundry are of two kinds: green-sand cores and dry-sand cores.

Green-sand cores are used wherever possible because of the greater economy secured from the standpoint of labor and material costs. The core machines are located adjacent to the molding machines, and the sand is delivered to them directly from the floor above with a minimum amount of handling.

Miscellaneous and complex cores are baked. In order to minimize handling, the core ovens and benches are located only a short distance from the molding machines. The silica sand is delivered to the core room in skid boxes from an inside storage bin, and here it is mixed with the proper amount of core binder and delivered to the core makers. After baking, the cores are placed on portable racks and delivered to the molding machines on specially constructed hand trucks.

Cleaning Section

The design, size, and mass of castings used in the manufacture of sewing machines are readily adapted to tumbling, and for this purpose metallic shot and stars are used as abrasives for removing the molding sand in the cleaning operations.

The cleaning room is located in the west section of the building and on the fourth floor, which is on the same level as the third floor of the east section, and is adjacent to the shake-out stations of the mold-conveyor systems. Instead of carrying the castings to the tumbling barrels, which are of the friction-drive type, the tumbling barrels are brought to the shake-out stations, the shake-out operators merely hooking the hot castings from the shake-out gratings into the tumbling barrel. When the tumbling barrels are loaded and closed, overhead traveling cranes pick up the barrel and deposit it on the run.

This section has a total of 80 tumblers, which are divided into a series of eight individual lines of 10 each, and the construction is such that any tumbler may be run independently of all other tumblers.

When the cleaning operations have been completed, the crane delivers the barrels and castings through a hatchway to the finishing section located on the third and second floors.

The loose sand resulting from the tumbling is delivered by a belt conveyor to a refuse-storage bin and dust collector at one end of the building and later removed to open-top cars without further rehandling.

Finishing Section

In this department the tumbling barrels are unloaded and the castings delivered in skid boxes to the filers, grinders, and polishers, and after the work has been completed it is delivered to the first floor ready for shipment. Some castings are shipped to the machinery departments directly after cleaning; this depends on the use of the particular casting.

Electric Trucks

In handling materials an ideal condition is obtained only when the system is properly organized and the materials are moved as rapidly as they are ready for succeeding operations. For this purpose electric trucks are coordinated with the mechanical units for the routing of all materials.

Concrete and cast-iron floors throughout the foundry permit the extensive use of electric trucks and are an important factor in reducing the maintenance cost.

Electric trucks are used because of their flexibility, and cover a wide range of handling problems. Specially constructed electric trucks equipped with large built-up hopper bodies and bottom-discharge doors are used for delivering facing sand. Electric lift trucks are used extensively for delivering castings and in disposing of general refuse accumulations, and when not in use for routing work may be applied to other handling activities. Each truck is scheduled and the drivers know exactly what is expected of them.

FOUNDRY ORGANIZATION

In studying the management organization in a continuous foundry of this nature, it must be remembered that the foundry is but one of a number of departments in the plant and is governed by the general factory policies.

The foundry and pattern departments are closely associated with each other and are under the general supervision of an assistant factory superintendent, who acts in an executive capacity, directing the activities of both departments. Each department is under the direct supervision of a manager, but there must be close cooperation between the two in order to develop the greatest efficiency.

Other members of the organization include the general and maintenance foremen, bookkeeping and stock-section staff, and other sectional foremen in charge of productive labor, such as melting, molding, core making, cleaning, and finishing.

The foundry is operated as if it were an independent business. It is charged with factory overhead such as insurance, taxes, interest, rent, repairs, depreciation, and power; it is debited with all raw materials and receives credit for the finished castings from all receiving departments.

MANAGEMENT POLICIES

During the building and organization of the continuous foundry it became more and more apparent that certain fundamental policies must be laid down in order that a high quality of product be maintained, the efficiency of the machinery be kept at a maximum, and the costs of production at a minimum. Some of these policies that may be of general interest will be outlined.

Foundry Executives' Weekly Conference

A weekly meeting of the foundry executive organization, together with the members of the factory engineering staff directly interested in foundry work, is held for the purpose of discussing current foundry problems of quality, operation, and new development. The minutes of these meetings, a copy of which is sent to the works manager for his information, are filed as progress reports. This plan has proved very valuable in broadening the vision of the entire organization and in emphasizing the more important foundry problems. The organization is kept informed of all new developments in foundry practice outside of our organization by convention reports and technical papers which are discussed at these meetings.

Pattern-Equipment Conferences

The importance of properly designed and constructed patterns cannot be too strongly emphasized, and to insure this result, close cooperation between the foundry and pattern department is absolutely essential. Conferences between the managers of the two departments are held daily, and all matters of general policy are referred to the assistant factory superintendent.

The assistant factory superintendent is in constant touch with the factory production office to enable him to provide in advance for all new requirements.

Engineering Advice and Development

Engineering control and development is an important factor in the scientific development of the foundry, and while the latter has its own organization of trained men, it receives the benefits of engineering from the factory engineering departments and laboratories, consisting of mechanical, electrical, civil, chemical, and metallurgical departments.

The chemical and metallurgical departments are closely associated with the foundry, both in routine control and in development work. In these departments routine analyses of all coke, limestone, and pig iron are made, and the quality of the iron is further controlled by Brinell hardness tests and microscopic examinations. The detailed sand analysis is made in the chemical laboratory, but a sand-testing laboratory operates in the foundry to control the moisture, permeability, and strength of the sand in daily use. By instituting a control of this nature the quality of the sand can be corrected before it is used, and excessive foundry losses can easily be prevented.

Development work is continuously in progress in the metallurgical department and is converted into practical results by cooperation with the foundry organization.

Production Control

Production control in a continuous foundry resolves itself mainly into a properly planned schedule and routing system, shipping-section cost system, purchasing division, and stock room. Records and statistics are essential in operating a continuous foundry of this character and are of much more importance than in the ordinary foundry.

Scheduling and Routing. The proper routing of parts through any department is obtained only after a careful analysis of the requirements and by having them travel in a straight line to the shipping points or stock room accompanied by some form of identification.

The factory production department regulates the flow of work from the foundry by means of an order system, and governs these orders according to the demand of the product, from figures obtained through the factory management. In routing the castings through the various departments they are assigned part numbers in order to facilitate clerical labor.

The foundry orders controller is responsible for production, and it is his duty to acquaint himself with stock records and weekly and bulk orders, so that he is in a position to have parts made either for production or for stock requirements.

Orders entering the department are generally issued on two forms: a weekly order, which is a standing order for a definite number of castings to be delivered each week, and bulk or quantity orders. Bulk orders are given a due date, and the work is arranged in strict accordance with the schedule.

In starting the work through the department the orders controller issues a production order in triplicate: one to the foreman that has charge of manufacturing the castings, one to the pattern storage, thereby applying for the pattern, and one to the shipping clerk. After receiving the pattern the foreman produces the castings, which automatically pass through the successive operations until they reach the shipping station. The shipping clerk, having received notice of the castings, is responsible for their

proper destination. After the pattern has been returned to storage and the parts have been shipped, these three production orders are returned to the orders controller, who destroys the duplicates, but retains the original for reference.

As most of the operations in the continuous foundry are paid for on a piecework basis, it became necessary to devise some method for crediting the molders and other piecework operators with the actual work they had performed. The molding machine and shake-out operators receive credit for their work through a system of checks which are numbered corresponding to the molding machines, the operators placing one check on each mold made. After the unit has moved to a different section of the foundry, the checks are removed, sorted, and later delivered to the office for counting and recording. These results are entered daily by the foreman on the payroll work sheets indicating the part numbers, operations, quantity of castings, and piecework rate of the work performed by each operator. At the end of each week a summary is made deducting the total defective castings from the total number of castings produced. The payroll is determined from these work sheets, which are delivered to the factory payroll department, where the extensions are entered on the factory payroll sheets, and later returned to the foundry office for their permanent record.

All operators receive their pay checks weekly, from the factory payroll department through an identification ticket issued by each foreman under whose direction the work was performed.

A ticket system is used in the cleaning department for checking the tumbling operations. Tickets are placed on each tumbling barrel indicating the part number, tumbling time, and also the packer's number. Aside from indicating the tumbling time, it shows the operator who is responsible for the packing operation. After the castings have been delivered to the operators in the finishing department, they are counted, and a duplicate foundry inspection ticket is placed with each lot of completed castings, one acting as a follow ticket, which remains with the castings throughout the successive operations until they are ready for shipment, while the other acts as a stock office ticket, which is retained by the foreman and is checked against the record of the daily work slips of each operator. After the foreman has checked both the stock and work tickets, they are entered on his payroll work sheet after the piecework rates are checked in the foundry office. These work sheets are then routed in the same manner as the work sheets for molding operations.

Shipment of Castings. The foundry is connected with all receiving departments, factory production, factory cost, and factory bookkeeping departments through the large amount of detail work necessary in the shipping and receiving of castings. An example of this work is as follows:

After the castings reach the shipping department, the clerk removes the follow ticket for his record and substitutes two shipping tickets, which are delivered to the receiving department with the castings; one duplicate, however, is retained by the clerk. The receiving department, after checking the quantities and part numbers on the tickets, signs the original and returns it to the foundry office. At the end of each week all tickets are summarized and delivered to the receiving department for their signature. This summary is made in triplicate: the original is delivered to the receiving department, one copy goes to the factory production department, and one copy is retained in the foundry shipping department. The receiving department, after signing the original, delivers it to the production department, and the latter sends its copy to the receiving department. In this way errors due to the quantity of the castings delivered are less likely to occur. Entries are made of shipments by each of the departments concerned; the receiving department does this to show the number of castings received, the production department to show the movement of stock, and the foundry department to eventually receive credit for the castings delivered. Bills containing the part numbers and quantities of the castings delivered are sent to the production department periodically for checking, and later, after pricing and extending, are delivered to the cost and bookkeeping departments for credit. In this way a careful check is maintained on the production, movement, and cost of all castings.

Stock Room. It is obvious that with a variety of 1500 to 2000 different castings made in this continuous foundry—some of which are required in large quantities weekly, while others are in smaller demand, and particularly as some of the castings are small, perhaps as many as 30 or 40 castings being produced per minute while the demand calls for perhaps only 10,000 per week—some latitude of actual production must be allowed. To allow for this intermittent demand for certain individual castings and for the overmolding capacity of others, a foundry stock room was inaugurated for the storage of surplus castings. As a general policy we have established maximum and minimum quantities of the various castings that may be carried. It is our aim to keep close to the minimum quantity set in order to have as little rehandling as possible, and also to keep the carrying charges as low as possible.

It is true that greater efficiency can very often be obtained if the castings are shipped to the various receiving departments directly after manufacturing, and it is our intention to do this whenever possible, placing only the surplus in stock.

The stock room is located adjacent to the foundry and with sufficient aisle space for receiving and delivering castings. The large-size castings are placed in storage bins, while the smaller castings require a properly labeled rack and shelf layout. Each casting has a certain rate of demand, and the most active classes of work are given the most desirable and advantageous positions.

A complete list of the quantity of each item is kept at both the stock room and the foundry office. In this way it is possible to keep an accurate record and perpetual inventory of all stock and also to reduce errors due to mixed stock, mislaid stock, and the receiving of incorrect stock. The office receives notification of all stock movements through department shipping tickets.

Corrosion prevention is a most important item when stock has accumulated, and we aim to keep the temperature of the room at 70 F in order to minimize rusting of the parts.

Inspection of Castings

The quality of the casting produced is considered paramount, and for this quality demanded the unit foreman is held responsible. He is provided with an assistant whose duty it is to constantly inspect castings for defects, and by reporting to his superior, cause the remedy to be applied. This assistant foreman from time to time takes molds from each molding press immediately after pouring, not waiting for them to take the usual time and course through the shake-out, has them immediately sand-blasted, and performs a rigid inspection.

The operators at the shake-out also give the castings as close an inspection as possible, reporting immediately to the unit foreman such defects as they find, but naturally the main inspection comes after the castings are tumbled, as defects are then more easily discerned. We have found that the cheapest and most effective method of inspection is to pay for all operations on castings subsequent to the tumbling operation, on a basis of good castings produced. Thus the grinder removing fins and gates does not get paid for grinding defective castings, nor the polisher for polishing defective filed castings. While this policy may have the effect of making the operators overcautious and causing them to set aside work that later may be saved, nevertheless we find it better to err on the side of caution than to allow castings to pass

to subsequent costly machining operations and later to be discarded.

In addition to the foregoing inspections, a constant check on inspection is being made by the foreman of the cleaning and finishing departments. A final inspection by the foundry shipping department is made before the castings leave the foundry.

The foreman and operators are constantly informed of defective work. Foundry discount sheets, which constitute a combined record of the inspections, are distributed to the molding foreman each day, with a record of the part numbers, amounts, and defects, classified in detail. The relation between the percentage of defects and the moisture, permeability, and strength is charted on a bulletin board, for the benefit of both the foreman and the management. In order to acquaint the operator with his particular defects, a discount record is kept on the blackboard of each molding machine.

We believe that this type of inspection is necessary in mass production, particularly, for the type of labor employed in the foundry.

Maintenance

In order to obtain long life and uninterrupted service in operating foundry mechanical units and in the mechanical transportation of materials, the maintenance of the machinery is of supreme importance. In the foundry described, approximately 5 per cent of the organization is employed in this activity, and the main function of the maintenance section is to inspect and repair all mechanical and electrical equipment.

The complete foundry molding system, as previously described, is divided into five independent units, the maintenance of each unit being in charge of its foreman. In addition to the foreman there is a unit operator, whose duty is to watch the mechanical-conveying and sand-handling system of his unit, to see that all moving parts are constantly lubricated, and to report promptly to the foreman any sign of weakness or excessive wear that has developed.

In addition to the servicing of the system units during the operating hours of the day, a careful weekly inspection is maintained on all mechanical and most of the electrical equipment.

The foundry is equipped with a machine shop for the maintenance and immediate repair of all minor equipment. This is essential because the shutting down of important handling units for any length of time means a loss in both production and in time-work productive labor. All minor electrical and mechanical repairs are made in this section, while other repairs that are too large to be handled here are made by the factory electrical department and tool room.

Aside from the unit operator the foundry maintenance section employs a pipe fitter, an electrician, and a number of general repair men. The general repair men repair or replace all worn and broken machinery and make all other minor miscellaneous repairs, such as resurfacing flasks and repairing tumbling barrels. All air and hydraulic lines are serviced by the pipe fitter, while the electrician makes all minor electrical repairs, inspecting and lubricating all electrical equipment such as motors, electric trucks, and safety switches.

The cranes and elevators are under the direct supervision of the factory electrical department. The equipment receives a continuous inspection by the operators, as well as receiving a careful weekly inspection and servicing, and a weekly report is made on the condition of all such equipment.

The machine operators in the molding and finishing departments are to a large extent responsible for the condition of their machines, but in case of failure the foundry maintenance section is notified immediately.

The number of spare parts necessary for the maintenance of the mechanical units and other handling machinery has been reduced to a minimum, by producing standard interchangeable parts.

Suitable exhaust equipment is an important feature in reducing the dust present in all foundries. A cyclone dust-collecting system confines the dust from the cleaning and finishing sections for easy disposal and materially reduces the maintenance cost of the machinery and at the same time reduces the hazard to health.

Standardization

In mass production, standardization is important, because it eliminates waste and results in economies through a better and more uniform product at a lower cost. For this reason it is the general policy to standardize parts as far as possible.

Castings, equipment, and raw materials have been standardized. The most essential feature in the standardization of the product is accuracy and interchangeability in order that the parts shall fit properly, but the limits of any part will vary according to the requirements. In this respect, the shrinkage of the metal must be considered and the pattern increased in size accordingly, if the castings are to be produced within proper limits.

Flasks, pattern plates, and a large number of parts required for the various units of the continuous system are interchangeable. All raw materials such as coke, limestone, pig iron, and scrap are purchased according to standard specifications.

Cost-Accounting System

A cost-accounting system is of vital importance as a means of visualizing conditions and can be secured only by records from a comparison of standard and actual costs. Due to the enormous quantities of materials consumed in large foundries, items that ordinarily might be considered insignificant assume importance in the final analysis of costs. Therefore a system that keeps accurate records is an absolute necessity in producing efficient management.

The cost system instituted in this foundry shows, essentially, the cost of the castings and the consumption of materials, and it measures the efficiency of labor by comparison and acts as a guide for correcting faulty methods of manufacture. The accounting method is separated into three main divisions, as follows:

1 The foundry operations are classified into sections depending upon the type of work done
2 Each section is subdivided into separate accounts
3 A method is used for applying indirect and overhead expenses directly to the section involved.

In order to differentiate between the types of work, the production costs are classified as follows:

1 Melting section
2 Molding section
3 Coremaking section
4 Cleaning section
5 Filing, grinding, and polishing section
6 General expense
7 Special charges.

A summary of the cost of production is prepared periodically for comparison and the accounts are condensed, supplementing the summary with additional records that give costs in greater detail. If there is an abnormal fluctuation, the cause for this is determined and in many cases corrected.

A yearly report on the cost of production is also made for the benefit of the factory management in order to determine the efficiency of operation and also to determine the advisability of making fundamental adjustments and improvements.

Cost of Individual Castings. The average cost of manufacturing castings is insufficient, and it is essential that a further analysis

be made by studying the cost of producing castings from specific patterns. By the aid of job costs the management has a guide to aid in producing castings at a minimum. A record sheet is kept of the cost of each different part, giving detailed information of operations, prices, and cost per 100 lb.

Labor is important, because it is one of the most variable factors in contributing toward success or failure. Since the costs of individual castings are largely dependent on labor costs, incorrect results may lead to disastrous cost finding. Rate setting for molding and core-molding operations acts as a means for controlling direct labor costs. The molding labor and molding piecework rates are established by the department head after conferring with the foreman, and these are based on experience with similar work, the volume of sand, size of the flask, number and size of the cores, core-setting time, type of patterns, and the number of castings that can be produced per day.

In a continuous system it ordinarily is difficult to obtain accurate results on time-work productive labor and to connect this directly to the individual castings, due to the fact that one operator distributes his time over a variety of patterns. We have found that a convenient and accurate cost can be established for this type of labor from the production of both the individual and total number of machines operating on each unit. This is called the molding-press day and is defined as the number of molds produced each day on one molding machine. This is the basis of all time-work productive-labor costs from molding to shake-out. In this way an accurate cost and cost comparison can be established for any particular casting; also there is less clerical effort necessary in order to obtain accurate results.

The rates in the filing, grinding, and polishing section are determined by a combination of rate setting and the results of previous records.

The volume of detail necessary for an accurate determination of costs is also valuable for statistical purposes, especially for comparison between the different factories operated by the company.

Purchasing. All materials are purchased through a central purchasing department, which keeps in close contact with the organization. Technical specifications have probably been the most notable advancement in the purchasing of equipment and supplies and are used wherever possible.

It is the intention of this department to purchase products in the most suitable quantities and in the most economical amounts.

Records

In large organizations records are of major importance in presenting the facts. A detailed cupola report is made daily for the benefit of the management, and it is also used as an office record for compiling the detailed weekly summary reports. The daily cupola report shows the iron mixtures, analyses, physical tests, and the melting ratio, while the weekly summary reports show the consumption of all raw materials, raw material stock on hand, and cupola data, as well as a detailed tabulation of all defective castings throughout the factory.

Safety Methods

The nature of foundry work in general, and particularly where machinery is employed, requires that a large amount of attention be given to safety methods in order to minimize accidents and personal injury.

Throughout the system safety has been carried out in detail by the application of mechanical safety devices consisting of safety control switches, automatic stop switches, safety clutches, and safety appliances. Each unit can be stopped immediately by manually operated safety-switch controls located at four different sections equidistant from each other and at points where workmen are stationed. Automatic safety stop switches have been installed to prevent possible accidents from falling flasks on that section of the system where the mold-conveying unit returns to the first floor. The stop switches are placed at definite points so that flasks that project from any of the trays on the conveyor unit, constituting a hazard, will contact with the switches, automatically breaking the circuit and stopping the system. Each sand-distributing unit is provided with a safety clutch in order to disconnect it from the sand-conveying belt whenever necessary. Metal protecting guards are used extensively for protection against moving machinery such as gears and grinding wheels. Operators in all sections of the foundry are provided with other safety appliances such as leggings, goggles, and respirators.

Because the iron is poured at definite stations and the molds are conducted away as quickly as possible, the smoke and gas are removed before they can have any effect on the men. In addition, the operators at the shake-out have been further protected by the installation of fan equipment in order to afford more ventilation, particularly in the summer.

Education in safety is important, and we believe that bulletin boards, properly located, carrying photographs with legends of what may occur, have a psychological effect on the men, tending to make them more cautious in their work.

A branch of the factory hospital is located in the foundry, for the immediate treatment of the more common accidents. A nurse is in charge, but is under the supervision of the factory physician. The factory management is kept informed of this work through records and reports of the injuries sustained by the workers.

ESSENTIALS FOR EFFICIENT MANAGEMENT

Maximum efficiency from a continuous-production foundry can be obtained only by a careful consideration of certain essential facts, such as these:

1. The work must flow in straight lines to the shipping points
2. Responsibility must be fixed, particularly with reference to subordinates
3. Engineering research and development are necessary for improving the quality and reducing the cost of the product
4. The grade of work produced must be considered paramount
5. A well-designed production-control and cost system is fundamental; it should show by records the location and the quantity of work in the department as well as accurate detailed cost data
6. Proper maintenance and safety devices are essential.

CONCLUSION

Maximum efficiency from a continuous system, as measured by the cost of production, can be obtained only through scientific management.

MAN-55-6

Fundamentals of Training

By G. GUY VIA,[1] NEWPORT NEWS, VA.

Fundamental principles involved in a modern training program are outlined. Beginning with an accepted definition of training, certain postulates are set up: (1) The careful selection of the learner. (2) The careful selection of the instructor. (3) The development of a suitable system of grading. (4) The analysis of the business for its training content. (5) The objectives of the training program. In selecting the candidate for training, attention must be focused on (*a*) physical, (*b*) manual, (*c*) mental, and (*d*) moral qualifications. A certain minimum of each capacity should be evident. The instructor must be a craftsman, a teacher, a psychologist. He must be a leader, an adviser, and a friend. A method of grading somewhat out of the ordinary is developed from Richard's formula for effective training. This system uses the performance of the average worker as a standard, rather than that of an imaginary ideal worker who has never existed. The instructor should analyze the business thoroughly for its instructional content, regrouping the units according to the learning difficulty from simple to complex. This analysis may often be turned to useful account by the thoughtful executive in comparing the worth of various employees, reviewing the processes of the business, etc. The paper is concluded with the reminder that the perpetuation of a business depends upon the continual absorption of new blood, which gives to the study of training an unusual significance.



TRAINING has been defined as the process of developing through a series of progressive experiences an ability on the part of a learner to do something that he could not do before. This definition embraces every field of human endeavor, each with its manifold activities, and offers a secure footing on which to erect a most satisfactory training program, whether simple or elaborate. The newly employed cook in the kitchen must receive training in the dietary habits and peculiarities of her employer's household; the new bookkeeper must needs be trained in the routine of the debits and credits of the business; the new machinist must become familiar with the methods, tolerances, etc. of the shop; even the newly appointed manager must himself learn to render wise decisions in the complex situations that are continually arising in the discharge of his duties. This business, then, of "developing ability through a series of progressive experiences" would seem to have a wide application, and an understanding of the fundamental principles involved should go far toward the elimination of wasted time, effort, and material in this increasingly important field of engineering activity—training.

Two phrases in the definition cited are worthy of consideration: the first, "developing an ability," and the other, "through a series of progressive experiences." The development of ability implies, first of all, careful selection of the novitiates to weed out those who by reason of mental or physical imperfections may be unable to profit fully from the training offered. Better yet, if choice can be limited to those who are equipped to profit most from the opportunity offered, greatest efficiency is secured from both a training and a performance standpoint. Waste of time and effort may be averted here. It implies as well the careful selection of those who are to be assigned to the duty of instruction. Poor instruction is profitless instruction, to both beginner and employer, and teachers of talent and ability are not common in the world of business and industry. It implies also a carefully determined method of grading, fair to both employer and employee, in order that each may know whether actual and satisfactory results are being secured.

The phrase "through a series of progressive experiences" sounds innocent enough when spoken and looks equally simple in print, but what a useful and complex range it unfolds to the patient investigator as he undertakes to apply it to the practical phases of business or industry! "Series"—one after another, covering the entire field of the business or industry; "progressive"—graded in difficulty from simplest to most complex; "experiences"—the multitudinous tasks, jobs, or operations that form the warp and woof of the business fabric. Here, then, is involved the complete analysis of the industry, its processes, the danger hazards, the trade terms used, the mathematical computations, the applications of the principles of science, perhaps an understanding of human behavior. These and many other fields of knowledge must be explored, charted, and rearranged before the text book which constitutes the "series of progressive experiences" can be put into the hands of the learner. The task is stupendous, but both the direct and collateral benefits of determining first the differentials of the industrial equation and the subsequent reintegration of those refined differentials will be instantly apparent to the engineering or managerial mind.

Unfortunately, this definition of training is not complete. The beginner's mind may be well stocked with carefully presented and well-digested knowledge; his hand may possess the last refinement in skill and cunning; but unless there is created in him an ever-present urge to perform gladly, willingly, and pridefully at his daily task, a most important training fundamental has been neglected. The development of morale must have prominence in any training program, real morale as distinguished from misdirected and frequently overdone paternalism. The machine may emancipate the mind and the hand of the worker, but nothing can substitute for his heart. With this thought in mind, the definition might be revised to read: "Training is the process of developing through a series of progressive experiences an ability on the part of a learner to do willingly and pridefully something that he could not do before."

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Objectives of Training

From these introductory remarks on the definition of training, there may be taken the following interesting phases for more detailed study:

1 The careful selection of the learner
2 The careful selection of the instructor
3 The development of a system of grading which is founded in fact and not on guesswork
4 The analysis of the business to determine what shall be incorporated in the program of training
5 The process of developing: (a) the hand, (b) the head, (c) the heart.

These last three items—the hand, the head, and the heart—may be considered the three objectives of the training program, for every activity associated with training will be consciously directed at the development of one or another of this important trio. Does this or that project further the development of the hand, the head, or the heart of the learner? There is here a convenient yardstick by which one may measure the worth of any training proposal. Can this or that instructor reach effectively the hand, the head, and the heart of the learner? The length of shadow which the instructor casts on this objective screen determines his fitness for the assignment. Does this or that applicant possess the necessary requisites of hand, head, and heart to profit most from such a program? What an effective mesh to separate desirable wheat from the human chaff pile! From every angle the excellence of the program may be measured by the degree to which it approaches the utmost fulfilment of these three desired objectives.

Selecting the Learner

a Physical. Every human being represents the integration of an infinite variety of tendencies and capabilities, good, bad, and indifferent. Clamoring at the gates of business and industry are dull individuals and intelligent, educated and ignorant, fat and lean, short and tall, handsome and ugly, with every possible variation in between. Judicious choice is indicated at every turn. Every task demands certain physical and mental qualifications of the operator which cannot be ignored except with unsatisfactory or even disastrous results. Why waste time training a crosseyed girl to sell millinery? On the other hand, the same girl might become an expert designer in the back room and be most happy at her vocation. The sight of a 200-lb electrician clinging to the top of a tall, slender ladder will cause even the most perfunctory safety engineer to look the other way, while the same man snaking a heavy cable through a conduit makes a pleasing industrial picture. To be sure, physical limitations are in evidence on every hand and must assuredly be given consideration in selecting the learner.

b Manual. Manual dexterity cannot always be acquired. Harry B. comes from a long line of forebears who were mechanics and has every appearance of becoming a satisfactory craft apprentice, but he is not. Somewhere between Harry's mind and his finger tips there is faulty synapsis, and Harry's fingers will not obey Harry's brain. The results are deplorable, both to Harry and to his employer. Arthur N.'s father is a nationally known expert in machinery operation, as were his other ancestors. Arthur also has brothers who are gifted artizans, but he will never acquire craft skill. Arthur's fingers obey his will, but reluctantly! Arthur's motor-mental hook-up is awry.

These defections may often be brought to light by a study of the applicant's previous school record considered in connection with certain simple manual tests for determining digito-muscular control. Such an applicant's medical history will sometimes show epilepsy, fainting spells, etc. all of which information is concealed by the applicant himself and is discovered only after painstaking investigation.

c Mental. Does the task require the frequent exercise of judgment and intelligence? The mental capacity of the applicant then assumes prime importance, and a careful investigation of this faculty is indicated. If available, the applicant's previous school record, obtainable from the school principal, is illuminating. It may sometimes be advisable for the prospect to be given an intelligence test on the spot to measure this important factor in his qualifications, and while the results are not always infallible, they may at least be relied upon to eliminate from consideration those who are hopelessly inferior mentally.

Some years ago a certain large industry made a study of the correlation between the intelligence of its apprentices and their training progress. Groups of 40 boys, each in two widely different crafts, were chosen for study. Each apprentice was given an intelligence test and classified in one or the other of four divisions, according to the results obtained.

The ten highest scores were segregated in group I, the ten next highest in group II, and so on. The training averages of these same boys over a period of several months were treated in the same manner, the ten best apprentices being segregated in group A, the ten next best in group B, etc. It was expected if correlation existed that the majority of the brightest boys would be found among the leaders in the shop, while the dullest boys would evidence retarded progress in craft training. How well these expectations were borne out may be seen from a study of the distribution chart, Table 1.

TABLE 1 SKILL-INTELLIGENCE CORRELATION

Shop group	Intelligence group I	II	III	IV
M— Trade				
A	5	1	3	1
B	3	4	1	2
C	3	3	0	4
D	1	1	4	4
F— Trade				
A	6	3	1	0
B	1	4	4	1
C	2	2	2	4
D	1	1	3	5

This industry has since methodically eliminated from consideration those applicants for apprenticeship who are unable to pass a satisfactory intelligence test, and has chosen only those whose scores were average or better.

d Moral. No longer can business or industry afford to ignore the habits of its employees, both during and after work hours. Conduct is a sure index to character, and human behavior mirrors the inner man. Industry is interested in character quality, if for no other reason than the maintenance of morale. A better reason is that men of character build products of character. A closer scrutiny of the habits and morals of the applicant is indicated. There is here an apt application of the old adage concerning the bent twig and the matured tree. Due consideration for the third objective, heart, will necessitate the elimination of the "rotten apple in the barrel," for those qualities of loyalty, punctuality, regularity, and service which industry demands today are not consistent with weak moral fiber. The risks involved in the employment of individuals of faulty character far outweigh their probable ultimate usefulness.

Selecting the Instructor

For years it has been the custom of industry to delegate the duties of training to aging employees who were giving ground in the production battle, the first step toward their eventual retirement; and while these men have done the best they could with a task which is now recognized to have been far beyond their capacities in most instances, industry today demands far superior

results from the training dollar. In the first place, manners and customs vary widely from generation to generation, especially the manners and customs of youth, and an understanding of the impulses and reactions of youth from whose ranks will be drafted the greatest amount of raw material for training is imperative. It is most unlikely that the veteran about to be retired has maintained a sufficiently close contact with young people to appreciate the psychology of modern youth. Even though a parent, he has probably long since thrown up his hands in despair over this same problem in his own home; and yet youth today is on the whole as clean and wholesome as ever. They have simply discarded the art of dissembling, which was so prevalent in days gone by, and yield little to the cloak of modesty. To this extent the problem of the psychologist in training is simplified, and the successful instructor must of necessity comprehend at least the fundamentals of this science. Teaching is both a science and an art, and the veteran undertakes a new field of endeavor too late. Years of conscientious effort and study are necessary if one is to become even a passably proficient instructor, and the science of pedagogics is not to be mastered overnight.

It is axiomatic in training that skill cannot be transferred; it must be acquired. Hence the years spent by the veteran in acquiring adeptness will be of no avail unless combined with the other qualifications for successful instructional ability, a rare natural combination. It would seem unfair then to delegate such a responsibility to elderly employees, though it must be borne in mind that exceptions, and ofttimes notable exceptions, do occur. As a rule, however, the young and ambitious craftsman willing to begin the study of psychology, pedagogics, and such necessary kindred subjects offers the greater probability of success in this work. He or she is not so far removed from youth itself as to have lost contact. An intensive course in teacher training (in which either the State or Federal Board for Vocational Education is willing to participate) will provide the necessary fundamental experience for the new instructor, and results thereafter will depend on his inherent skill, his patience, his diligence, and his intelligence.

A final word about the instructor: It should be borne in mind that he must not only be able to tell about his craft in striking manner, but many times daily he will find it necessary to demonstrate his ability as a craftsman. To do this he must himself be a skilled artizan and a master of his vocation. Another would be hopelessly lost the first day. His influence over the learner will be far-reaching. Is the proposed instructor the kind of person to influence youth? Is his head right? His hand? His heart?

System of Grading

The measurement of human effort presents a problem which is most baffling to solution. No two people ever see a definite occurrence alike, as millions of jurors in the courts can testify. Since daily work performance is merely a long series of definite occurrences, it is most improbable that there shall ever be found two people whose opinions of a beginner's progress will coincide. And yet, in all fairness the learner is entitled to know that his effort is not being spent in vain, and management should always be relieved of incapacity promptly. The rate at which training is proceeding is then of importance.

Training Curve

The study of thousands of performance records of learners shows a remarkable approximation of similarity; in fact, a fairly definite training curve. Beginning at or about zero, the curve mounts fairly rapidly at first, gradually flattening out in the midsection, and finally resuming its upward trend at an accelerated pace to completion. (See Fig. 1.) The reason for the plateau between the two slopes still remains a matter of conjecture, although some ingenious solutions have been suggested. Nevertheless, during the training period of every worker there comes a time when progress seems to cease regardless of effort and when despair possesses the learner. Time, patience, and encouragement in this crisis bring renewed hope to the discouraged worker, and eventually satisfactory progress is resumed. Whatever system of grading is chosen should be sufficiently accurate to reflect the true condition of affairs for the guidance of the instructor. As the usual 0–100 per cent method is seldom more than guesswork, the following method is offered as an improvement.

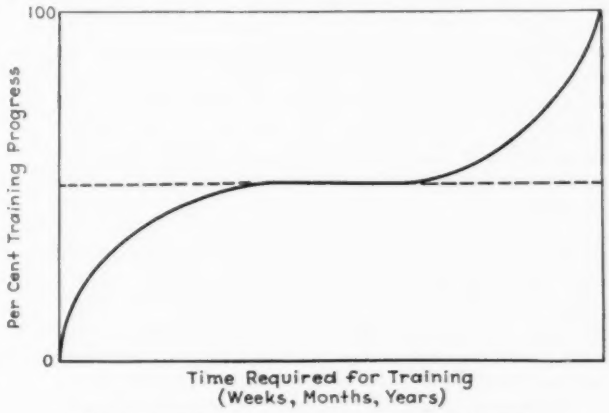


Fig. 1 Theoretical Training-Progress Curve

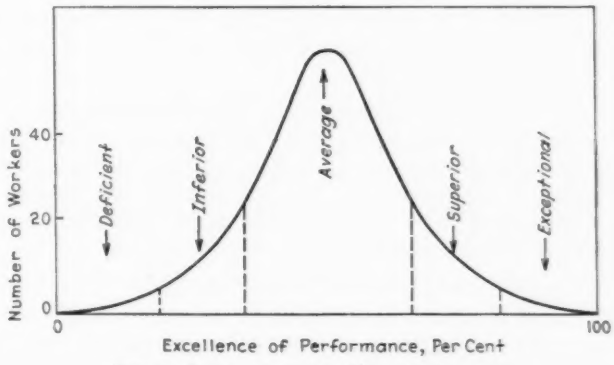


Fig. 2 Distribution of Worker Ability

Distribution of Ability

The backbone of any industrial organization is the average worker. Two-thirds of those who present themselves at the pay window will come within this classification. Half of the remaining third is made up of a few workers who are superior to him in general, and fewer still exceptional men who seem gifted with talent for continuously excellent performance. The workers in these three classifications constitute industry's money-makers.

Moving downward on the scale from the average worker, the remaining sixth may be classified under two headings of inferior and deficient. The inferior worker lacks something, job knowledge, manipulative skill, application, which renders his service somewhat less satisfactory, while the deficient employee lacks these or other qualifications in sufficient measure to make his retention on the pay roll questionable. Their effort is far less profitable. This distribution of abilities may be represented as in Fig. 2, and are classified as (1) exceptional, (2) superior, (3) average, (4) inferior, and (5) deficient. This table, then, provides a convenient yardstick by which one may scale the productive effort of the worker, and the adjectives chosen to describe

his several qualifications should be selected and arranged with these differences in mind.

Make-up of the Worker

A certain worker is spoken of as being a "good mechanic." Another is a "poor mechanic." Just what is meant by this distinction? Obviously both descriptions are generalizations, and each needs further analysis to determine what the qualifications are which cause one to consider the first worker "good" and the other "poor." Every human being represents the integration of a multitude of traits and characteristics—some prominent, others negligible; some inherited, others acquired. If one could accurately single out and evaluate each characteristic in an individual, he would acquire a true picture of that person. Of course this is impossible. At best one can only choose certain outstanding traits of aptitude and attitude which are desired in the worker and estimate the extent to which these traits predominate in a given individual. For instance, by observation it is known whether a man is lazy, slow, moderate, fast, or diligent, thus gaging his muscular effort. The results of his labor will evidence crude, mediocre, satisfactory, skilled, or expert workmanship—an index to his psycho-motor coordination. Is he enthusiastic, earnest, sincere, indifferent, or antagonistic toward the company's aims and policies? His conduct will reveal even the innermost impulses of his heart to the trained observer.

It only remains then to select certain indexes reflecting those traits which every worker is expected to possess in greater or less degree, and then to measure the amounts of these indexes which the individual in question actually has in his make-up. A fairly accurate picture will result.

Choosing and Evaluating Characteristics

In singling out these indexes, no better starting point can be found than that provided by Richard's formula, which states that effective training is the result of properly combining M, manipulative skill; T, technical knowledge; I, auxiliary information; J, trade judgment; and Mo, craft morale.

$$E = M + T + I + J + Mo$$

These five elements are evidently not all equal in weight, and these may be evaluated as one sees fit. For discussion, however, let it be assumed that a total number of 10 indexes is wanted ($E = 10$). The following is suggested as a fairly satisfactory distribution of weights:

M	4
T	1
I	2
J	1
Mo	2
Total (E)	10

Then select four indexes which reflect the worker's M, or manipulative skill. Five possible choices are given, although many others may be included:

Workmanship: "Quality of work produced"
Dexterity: "Readiness and skill in using hands"
Speed: "Time interval consumed in completing task"
Craftsmanship: "Reputation for trade ability"
Accuracy: "Ability to work within limiting measurements."

Technical knowledge, T, is the knowledge which the worker needs to perform satisfactorily the immediate task in hand. The carpenter cannot cut the joist unless he can read his rule. Reading his rule, then, is necessary to the immediate task, and this knowledge would be classified as T. The machinist must read a micrometer to work in thousandths. A knowledge of decimals, then, would be classified as technical knowledge for a machinist. "Confidence" certainly reflects the possession of this information and may be used as an index. If dimensional errors are made because of lack of this information, then "accuracy" may be listed here as well as under M, with the distinction that accuracy in M reflects muscular control and the errors would be small, whereas an inaccuracy in T would probably be an error of real proportions and would arise from ignorance. Other indexes will suggest themselves.

I, or auxiliary information, is that broad field of knowledge which provides a general background for craft accomplishment. Pure science, mathematics, economics, and sociology all contribute to a proper concept of industrial duties and obligations and provide a reservoir of information from which the worker draws from time to time in the performance of his duties. For employed intelligence, three indexes are suggested:

Comprehension: "Manner of grasping mentally"
Curiosity: "Anxiety for knowledge"
Memory: "Faculty for retention of knowledge."

Trade judgment, J, is ingenuity at task performance accumulated through the experience and observation of years. Some workers seem particularly gifted in this direction; others not at all. Two indexes are given for consideration:

Resourcefulness: "Capacity for finding or adopting means"
Originality: "Capacity for original conception."

Craft morale, Mo, is the ambition to excel. Beginning with the individual, it can be expanded to include group, shop, and company. It is reflected in both word and deed. Five indexes are noted:

Reaction: "Response to stimulus"
Attitude: "Indication of opinion"
Expressiveness: "Evidence of opinion"
Interest: "Attention with a sense of concern"
Initiative: "Ability for original conception and independent action."

Building Up the Report

The suggestions that have been offered are neither complete nor perfect. They are intended to point out a method by which certain characteristics in individuals may be chosen according to a definite plan and ratings made on these characteristics which will fairly represent the worker's worth. Selecting the proper number of indexes under each element of Richard's formula, a report might be built up as follows:

M (4 indexes)	1	Workmanship
	2	Dexterity
	3	Speed
	4	Accuracy
T (1 index)	1	Confidence
I (2 indexes)	1	Curiosity
	2	Memory
J (1 index)	1	Resourcefulness
Mo (2 indexes)	1	Interest
	2	Initiative

To complete the report, it is only necessary to select five adjectives to apply under each of the ten indexes, each group of five adjectives having a descriptive range from deficient to exceptional and weighted accordingly. To illustrate:

Index	Range	Weight
Workmanship	Expert	5
	Skilled	4
	Satisfactory	3
	Mediocre	2
	Crude	1

The completed report might then appear as shown in Fig. 3.

Comparing the Workers

It is to be noted that the middle adjectives, satisfactory and handy, are fairly descriptive of the workmanship and dexterity of the average worker, and would apply to approximately two-thirds of all employees. There are also two higher and two lower classifications in every instance.

In actual use the adjective is selected which best seems to describe each of the ten characteristics present in the individual, and the assigned weights are added to give a total sum. This sum in turn could be multiplied by the constant (2), to translate the score to a basis of 100 if desired.

A further refinement is recommended. If the average score for a group of workers be ascertained, and this average be divided

NAME.....

TRADE ABILITY (Aptitude)	CRAFT MORALE (Attitude)
WORKMANSHIP (M)	CURIOSITY (I)
Expert	Eager
Skilled	Inquiring
Satisfactory	Average
Mediocre	Passive
Crude	Indifferent
DEXTERITY (M)	MEMORY (I)
Adept	Exceptional
Clever	Retentive
Handy	Average
Clumsy	Fair
Bungling	Defective
SPEED (M)	RESOURCEFULNESS (J)
Swift	Ingenious
Fast	Clever
Standard	Reliable
Moderate	Ineffective
Slow	Helpless
ACCURACY (M)	INTEREST (Mo)
Exact	Intense
Accurate	High
Adequate	Normal
Faulty	Moderate
Inadequate	Low
CONFIDENCE (T)	INITIATIVE (Mo)
Inspiring	Diligent
Positive	Industrious
Certain	Willing
Uncertain	Listless
Lacking	Indolent

CHECK YOUR CANDID OPINION UNDER EACH HEADING

Fig. 3 Form Used for Reporting Learner Progress

into each individual score, a percentage will result reflecting the percentage of superiority or inferiority, according to the deviation from 100. Suppose the average score for a group of workers after multiplying by the constant 2 is 63.2. John Doe's score is 68; Richard Roe's score is 52.

$$68 \div 63.2 = 107.59, \text{ John Doe's percentage}$$
$$52 \div 63.2 = 82.27, \text{ Richard Roe's percentage}$$

Doe is 7.59 per cent better than average, while Roe is 17.73 per cent below, both intelligible results.

The chief advantage of ascertaining averages by groups lies in the fact that adjustment is thus made for the tendency of one grader to mark high, while another grades low. On a percentage basis all workers are finally equitably compared.

Extending the Scope

Richard's formula also lends itself readily to developing reports to suit different classes of employees. A balanced report can be worked out covering every worker from office boy to president by varying the weights given the different elements to suit the responsibilities of the individual and ascertaining suitable indexes to conform in number to the weightings assigned. The process would be identical to that already worked out. Such a table is suggested in Table 2.

TABLE 2 WEIGHTS ASSIGNED TO ELEMENTS OF RICHARD'S FORMULA FOR DIFFERENT CLASSES OF EMPLOYEES

	M	*T*	*I*	*J*	*Mo*	Total
Worker	4	1	2	1	2	10
Foreman	2	2	3	1	2	10
Superintendent	.	2	4	2	2	10
Manager	.	1	4	3	2	10
Director	.	.	4	4	2	10

Following the Learner's Progress

If the month-to-month percentage which the learner rates is plotted against the theoretical training curve, an interesting and illuminating record is created which far surpasses in usefulness the average record of this nature. Predicted performance is set against actual performance, and the interpretation of deviation is both enlightening and helpful to the instructor or the employer and of distinct moment to the learner. For every abnormal drop or rise there is a reason which challenges discovery by the instructor, and the correct interpretation of the causes of these variations is quite essential to the progress which the individual is making.

Three actual records are given for illustration and study. The first is the record of a normal student in electricity whose graph follows closely the theoretical curve. The second is the record of an unusually capable learner at plumbing whose curve averages 25 per cent better than normal throughout. The third is the picture of a failure at carpentry whose interest did not survive a bare six months, the reasons being lack of home discipline (mother dead), bad companionship, with the usual corollary of dissipation.

Analysis for Training Content

In the light of modern training methods, it seems incredible that for so many years beginners should have been placed in shops and factories to pick up as best they could such information and skill as they might by observing others—"training by absorption." It would be just as reasonable to turn loose a youngster in a university without a definite course of study and expect him to return as a competent lawyer, engineer, or doctor four years hence. Yet that is precisely what has been done with the boy in industry for generations. Industry's course of study is the careful and thorough analysis of the business, trade, or job, broken down into digestible teaching units and regrouped from simple to complex in order to facilitate learning.

Little need be said here about analysis other than to point out that it is essential to the instructor and ofttimes most useful to the executive. The whole panorama of a business—showing the introduction of a new trade term at this point, a danger hazard at that, acquaintance with a new tool or machine here, a necessary mathematical computation or a direct application of a science there, with reasons for everything—offers the thoughtful manager an unexcelled opportunity to review his processes and perhaps to make improvements which might otherwise escape notice. It may also be used to compare the abilities of workmen one with another and to judge their real worth to a business on a definite accomplishment scale.

For the learner the analysis provides the "text book" which must needs be mastered in detail for the information which it contains; it is likewise his "course of study," advancing him

from experience to experience, each progressively more difficult, leading him eventually to the desired goal of competence. The diligent instructor will lose no time in developing his analysis, if he is truly to bear the honored title of instructor.

A Brief Specimen Analysis

Each business demands an analysis to suit, but fundamentally all are alike in that similar operations are grouped together in what is termed a block, the total number of blocks depending on the business. For instance, in a machine shop there would be a lathe block, a planer block, a bench block, a drill-press block, etc. Having determined the blocks involved, each is further broken down into every possible operation, from the simplest to the most difficult and complex job. These will be known as the block units or lessons. These units are further analyzed for their instructional content as to trade terms, safety, science, knowledge of stock, etc. as may be seen from a study of the skeleton lesson plan shown below.

In rearranging the blocks for instructional purposes, no attention is paid to production order. The last production operation will be placed at the beginning if that operation happens to be the most easily learned. The idea held constantly in mind is to facilitate learning; production will follow in due time.

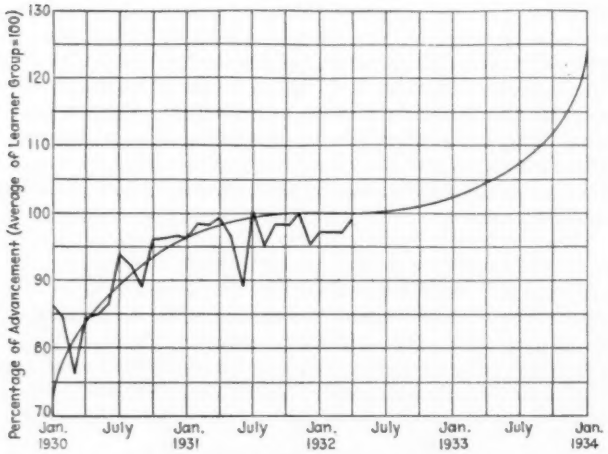


Fig. 4 Training-Progress Graph for Learner of Average Ability

(Showing close approximation to theoretical training curve.)

A sample analysis of patternmaking may be helpful. The patternmaker performs four separate and distinct sets of operations in completing a pattern, as follows:

1 Layout
2 Form and fit
3 Assembly
4 Finish.

Layout (1) and finish (4) are comparatively small blocks. The bulk of his effort is distributed between form and fit (2) and assembly (3). He continually alternates between them until he has built up an object which is called a pattern; hence it is here that one may expect and find the greatest number and variety of learning difficulties.

1 *Layout.* The first step in pattern construction is the preparation of the layout board, on which the object is laid down, full scale, with shrinkage in casting and finish in machining properly provided for trade drawing, pure and simple, with a judicious admixture of trade judgment.

2 *Form and Fit.* From this layout the patternmaker proceeds to the construction of the components making up the pattern. These components take the form of well-known geometrical solids, such as prisms, pyramids, cylinders, cones, etc. each with its frustums; in certain cases, solids of some particular form hand-shaped with the gouge, chisel, etc. are necessary.

This block also brings in most of the woodworking machinery found in the pattern shop, practically all of which is used by the patternmaker in turning out the average product, and it here becomes necessary to provide training on each machine as a part of this block. These are: (1) Band saw, (2) rip saw, (3) cut-off saw, (4) jig saw, (5) jointer, (6) planer, thickness, (7) disk sander,

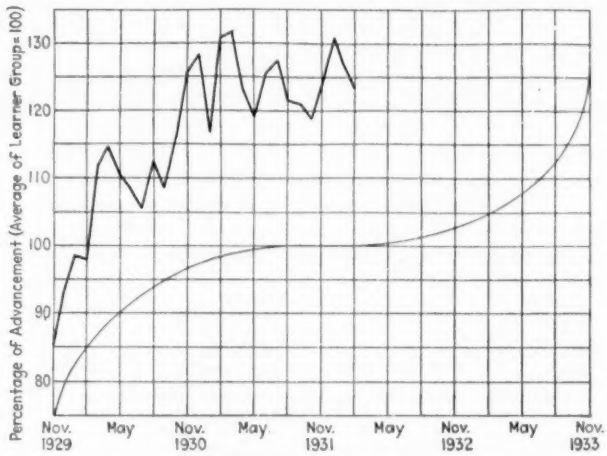


Fig. 5 Training-Progress Graph for Learner of Unusual Ability

(This is 25 per cent better than average throughout.)

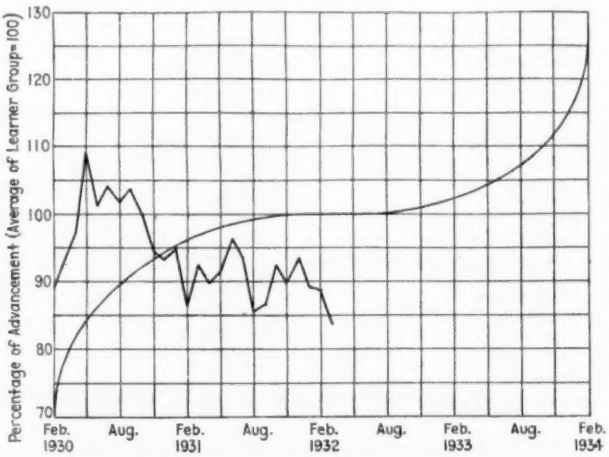


Fig. 6 Training-Progress Curve for Boy Who Failed to Make Good at Carpentry

(Major-trend lines intersect at February, 1931, when boy should have been released.)

(8) spindle sander, (9) turning lathe, (10) Daniels' planer, (11) boring machine.

3 *Assembly.* This is a simple block involving the use of nails, screws, dowels, and glue, and is continuously alternated with the previous block by the patternmaker in making the average product.

4 *Finish.* The reasons for and the proper methods of using sandpaper and shellac are developed.

Rearranged for effective instructional purposes the following order will in general be followed, reverting whenever necessary to blocks already taught to complete the pattern:

Block No. II SKELETON LESSON PLAN Lesson No. 40

TEACHING UNIT: TURNING LATHE: Between Centers, Material Solid, Grain Parallel to Centers

Primary or Secondary: Primary

	STEP 1—PREPARATION:
Idea 1	A piece of wood can be planed to a cylinder
2	It is very difficult to plane it to a true cylinder
3	
4	
5	

J. O. P.: A piece of wood can rapidly be turned to a true cylinder in the lathe

	STEP 2—PRESENTATION:
Point 1	Place material with excess end allowance between centers, screwing up tail stock
2	Turn down to within 1/8" of required diameter with turning gouge
3	Notch to correct diameter with 1/4" chisel, using outside calipers to measure
4	Turn down to notched diameter with flat chisel, using straight edge and calipers for measurements
5	Sandpaper smooth and shellac
6	Remove from lathe
7	Saw to proper length
8	Finish pattern

TYPE JOBS
1. Gate Stick
2. Riser
3. Weight
4. Hammer
5. Bushing
6. Socket

Block No. II AUXILIARY INFORMATION Lesson No. 40

(STEP 2—PRESENTATION)

Teaching Point	TRADE TERMS	SAFETY	CARE OF TOOLS, ETC.	SCIENCE	STOCK
1	Centers Tail stock Parallel			Amount of end allowance	Do not tighten too much or wood will split
2	Turning gouge Rest Calipers	Not to take too heavy a cut	Keep tools sharp.	Proper lathe speed Use of calipers while turning	
3	Notch Small turning chisel	Careful tool does not catch Position of gouge			
4	Large flat turning chisel	Cut must be very light	Keep tool very sharp	If tool is not sharp it will not cut but rub wood off	If tool is not sharp job will be ruined
5		Move rest back to avoid bruising fingers		Very light pressure on sandpaper to avoid changing shape or size	
6					
7		Clamp or brace to avoid rolling when sawing			
8					

FIG. 7 SAMPLE OF A SKELETON LESSON PLAN IN PATTERNMAKING

(Showing complete analysis of unit for instructional purposes and list of jobs typical of operation.)

Block I, finish
Block II, elementary forming
Block III, layout
Block IV, assembly
Block V, assembly forming.

The course of training will then be as follows:

1 Finishing patterns
2 Sharpening tools
3 Boring machine
4 Disk sander
5 Spindle sander
6 Band saw
7 Jig saw
8 Pony or thickness planer
9 Jointer
10 Cut-off saw
11 Power rip saw
12 Daniels' planer
13 Layout
- (*a*) Blueprints
- (*b*) Construction
- (*c*) Shrinkage allowance
- (*d*) Finish allowance
- (*e*) Draft allowance

14 Single-planed objects, no cores
15 Assembly
- (*a*) Nails and screws
- (*b*) Glue
- (*c*) Fillets
- (*d*) Dowel pins

16 Simple assembly of single-planed objects, no cores
17 Cores
- (*a*) Core prints
- (*b*) Core boxes

18 Simple assembly of single-planed objects, cored
19 Turning lathe
- (*a*) Between centers, material solid, grain parallel to centers
- (*b*) Between centers, material split, grain parallel to centers
- (*c*) Assembly between centers, grain at right angle to centers
- (*d*) Assembly, turned to turned, intersection plane or curved
- (*e*) Face plate, material solid
- (*f*) Face plate, material split
- (*g*) Face plate, material built up segmentally
- (*h*) Between centers, material built up in layers
- (*i*) Between centers, material built up in staves
- (*j*) Between centers, long objects, small diameter

20 Single-shaped objects
21 Boxed patterns
22 Skeleton patterns
23 Sweep patterns
24 General shop production.

Experience has shown that two years is sufficient time for the average apprentice to cover thoroughly this entire course, paying his way with salable production all along. Speed follows quickly thereafter, depending upon his natural ability. During the next two years his training curve should turn sharply upward, particularly if the boy has applied himself diligently to the related studies (mathematics, drawing, etc.) in the meanwhile. At completion he stands revealed a craftsman, skilled, intelligent, full of pride in his work and his company, the product of planned training.

CONCLUSION

Thus there has been given a hasty glance at an absorbing field of engineering, that of building with human material. Lack of time and space have precluded a thorough discussion of any of its interesting phases, and many important details have of necessity been omitted altogether. It is hoped, however, that enough has been given to emphasize that a prime necessity of any business is the perpetuation of itself through its personnel, which fact lends to the study of training an ever new and significant aspect.

Discussion

FRANK CUSHMAN.[2] Some of the fundamental considerations for efficiency in a training program which are emphasized in the paper are: (*a*) the proper selection of apprentices and instructors, (*b*) a progressive and systematic series of experiences for apprentices based upon analyses of the trades involved, and (*c*) a practical method of checking the progress of learners and, indirectly, the results of the program.

Perhaps the outstanding characteristic of the scientific method of dealing with any problem is to first secure all of the facts that have a bearing upon it and then to deal with these facts as intelligently as possible. The method outlined in the paper of dealing with a training program and its manifold problems is clearly in harmony with the definition of the scientific method. Many industrial organizations which pride themselves on the applied science reflected in their operations and processes incident to a turning out of their product, fail to appreciate the fact that it is also possible to deal with the problems bearing on the training of workers for their industry, by methods which are equally scientific.

The utilization of Richard's formula in connection with the system of rating apprentices and checking their progress is of particular interest. The writer would question, however, the correctness of assigning fixed weights to all of the factors throughout the period of apprenticeship. In the early stages—say, for the first year or two—the *M* and *T* factors should have maximum emphasis.

While there is no definite agreement as to the significance of the *I* factor in the formula, it is quite widely considered as indicating such things as the ability to comprehend the job in its larger aspects and in relationship to the industry as a whole. That is, at least, one phase of job intelligence. Job intelligence is also, in part, the result of the recognition of the inter-relationships of specific, heterogeneous, and often unorganized technical knowledge which an apprentice acquires on the job as he progresses in his trade. Such being the case, the building up of job intelligence is an important characteristic of the last years of apprenticeship, and it should continue to increase as long as the individual remains alive "from his neck up." This is true for all occupations.

Job judgment is often regarded as the result of the acquirement of skill plus extended experience in utilizing that skill in specific situations. If this idea is sound, the *J* element would have an increasingly larger value as the individual made more and more progress in learning or practicing his trade. For these reasons it would therefore seem that a more nearly correct assignment of weights would place more emphasis upon *M* and *T* in the beginning and an increasing emphasis on *I* and *J* as an individual became more and more expert in the doing of his work. It would seem to be entirely logical, also, that the weights assigned to the *I* and *J* factors would be even higher for a foreman than for a worker.

The morale factor *Mo* is probably one which is of equal importance throughout the career of an individual. It is intended to represent his attitude toward his job and toward his responsibilities as a worker, as well as an individual citizen.

C. F. BAILEY.[3] This matter of hand, head, and heart, this trinity in any young man's personality, is one to which attention is especially called by the author. His emphasis upon it in daily contact is a very practical part of his training program.

This learners' progress curve illustrated in Fig. 1 is also applicable to many of us as we doubtless recall passing through similar experiences, although we probably did not at the time recognize it. It would not be surprising if the author himself, while preparing this paper, might not have struggled along the plateau part of this curve when little progress seemed visible or apparent. The law itself is old even if we may have only lately discerned it. The discovery of this form of the curve applies not only to the students in early training, but it also holds in many of our own later efforts. All the instructors should understand this tendency in order that they may help and encourage the student rather than to yield to the temptation to blame him when at times his efficiency appears to drop.

If any lazy man thinks that he can set up and operate such a training system, he will be woefully disillusioned. It takes intense application, study, work, and perspiration, just as it does for a manager or any other effective executive to carry on in the highest responsible place.

BEN S. MOFFATT.[4] The paper is an excellent statement of some of the fundamentals of training and of a systematic means of measuring its character and progress. Nevertheless, there is one vital fundamental to any program of training with which this author, in common with all of the authors contributing to the subject during the last two decades, has failed to deal, and this is the engineering problem of securing the training operations from the existing work organization within a plant.

It is one thing to establish principles and to delineate fundamentals of an effective program of training, and quite another thing to secure their establishment as an efficient operating function in an industrial plant. A program of training, however carefully and scientifically planned, has still in the long run to prove itself not only effective in training, but also efficient in costs in accordance with the economic conditions regulating the particular plant in which it operates.

During the last decade, many of our boys have been inducted into so-called systems of apprentice training under industrial conditions wherein there did not exist the remotest possibility of securing the fundamentals to training as planned. The conditions existing in some of our plants may in many cases contain all of the elements to the fundamentals as enumerated by the author, and still be unavailable to a training program, owing to the character of their function in the work of the plant and where any attempt to incorporate them into the training program must result in excessive costs and disruption to production schedules, with the inevitable result that in times of economic difficulties this work is the first to be discontinued and wherein in normal times it will be permitted only through an expensive paternalism.

The difficult problem in the field of apprentice training is that of establishing just where, when, how, and to what extent it can be applied effectively in terms of training and efficiently in terms of costs. These two fundamentals to training are purely problems of production engineering.

RENSIS LIKERT.[5] The desirability of a broader training for engineers is being emphasized today in many quarters. The paper emphasizes the use and value of psychology to the engineer, and involves one aspect of the application of psychology to industrial problems.

The "theoretical training-progress curve," which is presented in Fig. 1, is not supported by the curves the author himself

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[3] Engineering Director, Newport News Shipbuilding and Dry Dock Company, Newport News, Va. Mem. A.S.M.E.

[4] Supervisor of Plant Training, Caterpillar Tractor Company, Peoria, Ill.

[5] Department of Psychology, College of Engineering, New York University, New York, N. Y.

presents and most flagrantly violates an enormous mass of experimental evidence. An almost countless number of experimental studies on the shape of the learning curve have shown that improvement tends to be quite rapid at first, but that with increasing practice less and less improvement occurs, until finally the curve approaches an asymptote which is spoken of as the physiological limit. We all recognize, for example, that in running the 100-yard dash it is relatively easy at first to cut 1/2 second or more off our time, but after we are able to run it in 10 seconds or less it takes a great amount of effort and refinement to reduce our time by as much as one-tenth of a second.

The curve which is generally regarded as best fitting the data from learning experiments is a logarithmic function,[6] although Thurstone[7] claims that it is best fitted by an hyperbolic function.

It is true that in certain instances a plateau does occur in the learning curve. Such a curve is shown in Fig. 10, on p. 125 of Woodworth's "Psychology," revised edition (Henry Holt & Co.). It will be noticed, however, that the plateau is not followed by the type of accelerated improvement shown by Mr. Via's curve, but by another rise showing negative acceleration.

The causes of these plateaus vary. A very frequent cause is a change in the unit with which the learner is dealing. In learning reception in telegraphy the novice first learns to receive letters. After reaching approximately his maximum rate for letters, he then starts receiving in "higher units" or words, and shortly his rate of receiving rises again as he learns to work with these larger units. In addition to "higher units," plateaus may be caused by fatigue on the part of the learner or a shift in interest.

The use of psychological techniques for developing and improving methods for selecting superior or exceptional men for a given job, as the author suggests, is no doubt going to be increasingly used.[8]

The best man for a given job may be, as the author stated, the most intelligent applicant; however, such is not always the case. In many kinds of work too much intelligence is a liability. Experimental studies have shown that in work involving simple, repetitive tasks, intelligent persons often become irritated by the monotony, and they then become a contagious source of dissatisfaction which may spread to other employees. As the author would say, their hearts are unsatisfactory. It is only by the experimental study of each kind of work using proper techniques that we can determine what are the optimum qualifications for that job.

The author's insistence upon "the development of a system of grading which is founded in fact and not on guesswork" is to be commended. It is regrettable, however, that he does not go further in removing the subjective element on the part of the instructor in obtaining the grades. His statement that "no better starting point can be found than that provided by Richards' formula" is to be seriously challenged. Certainly the use of what are commonly known as objective tests are much to be preferred. The latter can be used to reduce considerably, if not entirely eliminate, the subjective element, as represented by the instructor's whims, fancies, and prejudices, in the giving of grades. Even the attitudes of workers toward their work, the company, etc., can be measured quantitatively by the use of recently developed techniques.[9] In judging quality of work and various other factors which must be done by the instructor, the error due to the subjective element may be considerably reduced by the use of carefully constructed rating scales. The techniques used in judging mechanical ability in the Minnesota mechanical ability test study[10] well illustrate the construction and use of devices capable of reducing errors of judgment to a minimum. Such techniques might well be emulated.

If one wishes to analyze the different factors entering into achievement and to determine the relative importance of each factor, this can be done by recently developed statistical techniques. Space does not permit of the discussion of this subject here.

The author's suggestion of plotting the student's progress is to be heartily endorsed. Such use of the learning curve not only informs the instructor of the student's progress, but it is also a powerful motivating device for the student to observe his progress and encourages him to increase his rate of learning.[11]

Author's Closure

The criticism which Mr. Cushman offers to the use of fixed weights in the application of Richard's formula to the progress report has much merit. Undoubtedly each factor has a different rate of development at different points during the training period, and proper allowance should be made therefor. The conclusions which he offers in his discussion of the different factors will enable one to develop a series of reports which should give an accurate picture of learner progress from start to finish.

One of the prime requisites for the success of any training program is the understanding and sympathetic interest of the management. Behind Mr. Bailey's kindly remarks may be discerned the type of managerial attitude which provides inspiration for the successful prosecution of any personnel work, especially training. Without that continued support and encouragement, the best-planned training program must sooner or later fail.

Mr. Moffatt points out that this paper, like others of its kind, does not deal with the problem of securing training operations from the existing work organization within a plant. He submits that apprentices have sometimes been exploited in the past, while on the contrary other training programs have blossomed forth at excessive cost of operation only to wilt at the first economic frost. In these statements the author concurs. A program should not only train, but pay its way as well if it is to be justified.

Perhaps the most potent single influence in a successful and profitable program is the attitude of the management, without whose whole-hearted backing a thousand provocations and obstacles will arise, eventually crushing the sincerest and ablest effort. But when it is understood all along down the supervisory line that the boss expects real cooperation for his training program, it is surprising how smoothly matters will progress. Then if the proved principles of effective training have been adhered to in carrying on the work, and honest purpose has been manifest throughout, the program will prove to be a profitable investment from every angle, including the dollars-and-cents angle. The auditor's cost returns will invariably tell an interesting story. As to meeting production schedules, the interested learner will find a positive delight in holding up his own, as the author can testify from many years of just this type of experience.

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[6] See chapter on learning, by Walter S. Hunter in Murchison, C. (ed.), "Foundations of Experimental Psychology," Clark University Press, 1929.

[7] Thurstone, L. L., "The Learning Curve Equation," Psychological Monographs, No. 26.

[8] See, for example, Bingham, W. V., and Freyd, M., "Procedures in Employment Psychology," McGraw-Hill, 1926, or Viteles, M. S., "Industrial Psychology," W. W. Norton & Co., Inc., New York, 1932.

[9] Thurstone, L. L., and Chave, E. J., "The Measurement of Attitude," University of Chicago Press, 1929. Likert, R., "A Technique for the Measurement of Attitude," Archives of Psychology, 1932.

[10] Paterson, D. G., Elliott, R. M., Anderson, L. D., Toops, H. A., and Heidbreder, Edna, "Minnesota Mechanical Ability Tests," The University of Minnesota Press, Minneapolis, 1930.

[11] Book, W. F., "Psychology of Skill," New York, 1925.

The "theoretical training-progress curve" presented in Fig. 1 is offensive to Mr. Likert, who contends that the preponderance of experimental evidence leans to a hyperbolic or a logarithmic curve instead. In view of the conclusions presented by Mr. L. P. Alford in his excellent paper, "Laws of Manufacturing Management,"[12] there would seem to be a divergence of opinion on this interesting subject, each view being apparently supported by a mass of experimental data. The author confesses to an open mind on the question, although years of observation would incline him to support Messrs. Alford, Bigelow, and other industrial investigators who offer the "plateau between slopes" theory.

It is an accepted principle in psychology that in the process of learning many different connections are established simultaneously. The learner develops aptitudes and attitudes together, and a great deal more is involved in industrial training than the development of mechanical ability alone. This important fact must be taken into account in recording the learner's progress, and will considerably affect the technique of assembling data of this type. This may account for the divergence of opinion which has been noted.

The objection raised to the use of Richard's formula for measuring the progress of the learner is difficult to understand. A grade on workmanship, accuracy, etc., certainly necessitates a critical examination of the day-by-day production of the worker, besides requiring a study of the worker's attitudes as expressed in resourcefulness, interest, etc. The author has not found the use of the objective test method very practical in a busy shop where production schedules must be met regardless, and most assuredly he does not prefer them. Here again there is room for a division of opinion.

It is to be borne in mind that every modern training program reflects an effort to explore and chart that mysterious realm, the human mind, in its relation to physical reaction. Years of patient and careful investigation by our psychologists have thus far produced but few conclusions which may invariably be accepted in toto. Two individuals receive identical stimuli under identical conditions; their responses are altogether different. Why? Conversely, the same response is often observed as a result of different stimuli. Thus it would seem that human behavior yet defies accurate prediction, and beyond a few broad generalities we are still looking hopefully to the psychological laboratories for more light. Much fine work has been accomplished, but much more awaits the future.

It is indeed gratifying to have had these interesting and suggestive discussions offered, and the author is much indebted to each contributor. The wholesome differences of opinion which have arisen should serve to emphasize the need for continued research by both school and shop in this absorbing subject of training.

[12] Trans. A.S.M.E., vol. 48, 1926, p. 411.

MAN-55-7

Applications of the Kmh Method of Analyzing Manufacturing Operation

By L. P. ALFORD[1] and J. E. HANNUM,[2] NEW YORK, N. Y.

The kmh method of analyzing industrial operation, first presented by the authors in a paper before this Society in 1928, is used to show variations in size of plants, in length of work periods, and in kmh rates of productivity, value of product, wages, and salaries for four industries in 1929. The trends of kmh operating rates from 1923 to 1931 for 110 concerns are also studied. Important findings are: High wages and high expenditures for supervision accompany high operating performances; the kmh factor is the most convenient unit for measuring plant capacity and a more satisfactory unit for measuring employment and unemployment than a census of workers; the optimum size of plant for most effective production is small; the optimum range of length of work period is considerably shorter than the longest periods that have prevailed currently; changes in productivity are substantially independent of the business cycle; and, due to continuously increasing productivity, the same quantity of product turned out by 50 concerns in a 51-hour week in 1923, could have been produced in a 35-hour week in 1931.

IN A PAPER entitled "A Basis for Evaluating Manufacturing Operation" presented at the December, 1928, Annual Meeting of The American Society of Mechanical Engineers, the authors of this paper developed the kilo man-hour (kmh) method of analyzing industrial operation. The original search was for a base factor which should be: "Measurable by an established unit; universal in use in industry; easy to handle in computation; influential in its effect on industrial operation." The exploration yielded the kmh factor and the method of analysis based upon it. In our 1928 paper we said:



L. P. Alford



J. E. Hannum

> From this base factor, the kilo man-hour, numerous factors of industrial operation can be developed.....It is self-evident that many of these compound factors can be separated into component elements for the purposes of comparison and analysis, but all on the base of the kilo man-hour.
>
> It is evident, likewise, that the results of manufacturing operation expressed in terms of these compound units can be used to compare the period-to-period performance of a single industrial establishment, or to compare the relative achievements of two or more concerns in the same line of industry, or, even as regards certain selected factors, to compare the relative performance of concerns in different lines of manufacturing.

Work on applications of the method has been carried forward during the past four years. A sufficient number of data have now been accumulated to warrant the preparation of this paper, to show further details of the method, and to give certain results obtained and conclusions reached.

The material available has had two origins:

1. In a research into the factors in, and measurement of, technological unemployment made by the authors for the American Engineering Council
2. In a statistical analysis of certain data for four industries presented in a special industrial study of the United States Bureau of the Census.

These data in each instance represent a substantial assay of the fields investigated. The number of employees in the manufacturing concerns reporting to the American Engineering Council is $^1/_{63}$ of the total number employed in 1929, while the number of railroad employees is $^1/_{33}$ of the total number employed in 1929; the data in the industrial study of the Bureau of the Census were reported for 95 per cent of all the value of products for three industries, and for 25 per cent in the fourth. The extent of the assays and the origin of the data give them a high degree of credibility.

Several general findings are supported by the facts brought out by the studies:

1. The ranges of productivity, selling value of product, and value added by manufacture, are wider than those revealed by the study of 1928
2. The operating factors are of the same order as those revealed by the study of 1928

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[2] Editor, The Engineering Index, The American Society of Mechanical Engineers. Assoc-Mem. A.S.M.E. Mr. Hannum received degrees of B.S. (1915) and M.E. (1920) from Pennsylvania State College. He was machine-shop foreman, Skinner Engine Co., 1915–1916; instructor in industrial management, Purdue University, 1916–1918; industrial engineer, Diamond Chain and Manufacturing Co., 1918–1919; in charge of industrial education of service men blinded in the World War, Red Cross Institute for the Blind, 1919–1922; research engineer, Eyesight Conservation Council, 1923–1929; director, Safety and Production Study, American Engineering Council, 1926–1927; editor of The Engineering Index since 1927, and office manager of A.S.M.E. since 1931. He is also a member of the American Academy of Political and Social Science, the A.A.A.S., American Management Association, Illuminating Engineering Society, and the Institute of Management.

Contributed by the Management Division and presented at the session on Thursday, December 8, of the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

3 High wages are shown to accompany high operating performance

4 High expenditures for supervision are shown to accompany high operating performance

5 The kmh factor is shown to be a more satisfactory unit to use in measuring employment and unemployment than the data from a census of workers

6 The kmh factor is shown to be the most convenient unit to use in measuring and comparing plant capacities

7 The optimum size of plant for most effective production is small when indicated by kmh worked per year or number of workers. The ranges for four industries studied follow. The relation between kmh and number of workers is on the basis of 2000 working hours per man per year.

Industry	Optimum number of kmh	Optimum number of workers	Percentage of total kmh in optimum range
Blast-furnace operation..	270–640	135–320	33.8
Machine-tool building....	30– 80	15– 40	4.5
Petroleum refining.....	40–140	20– 70	7.7
Lumber manufacturing...	50–300	25–150	15.2

8 The optimum range of length of work period for most effective operating performance varies considerably among the four industries studied. The upper limit in each range is substantially lower than the highest values that have prevailed currently. The lower limit for three industries is substantially the 35-hour-per-week schedule referred to in item 11 following. The ranges for this factor are given in a tabulation below.

Industry	Optimum range of length of year in hours	Optimum range of length of week in hours	Percentage of total kmh in optimum range
Blast-furnace operation...	1810–2610	34.8–50.2	26.5
Machine-tool building.....	1930–2460	37.1–47.3	40.9
Petroleum refining.....	2450–2810	47.0–54.0	72.2
Lumber manufacturing....	1600–2640	30.8–50.8	64.0

9 Changes in productivity are substantially independent of conditions established by the expansion or recession of business activity, or of the operation of the business cycle

10 A majority of the manufacturing concerns reporting, 58 per cent, had a higher rate of productivity in 1931 than in 1928. The increase in productivity for 1928 over 1925 ranges from 1.3 to 83.5 per cent; and for 1931 over 1928 is 0.5 to 112.2 per cent

11 The concerns which showed a continuous increase in the kmh rate of production from 1923 to 1931 reduced, on an average, the number of kmh per unit of product by 31 per cent. That is, if the working force of 1923 had remained unchanged, the amount of product turned out in a 35-hour week in 1931 would have been the same as was produced in a 51-hour week in 1923

12 The railroads that reported showed a reduction in kmh per unit of service of 33 per cent from 1923 to 1931.

Measurement of Operating Performance

In a report submitted to the Committee on Technological Employment of the United States Department of Labor by the Committee[3] on Man-Hour Information of the American Engineering Council, in September, 1931, there were proposed certain units for measuring operating performance, each being a rate computed from some two of four records of industrial operation. The four quantities are:

A Average Number of Workers Employed
B Total Number of Man-Hours Worked
C Total Quantity of Product Output
D Total Number of Workers Employed.

Eight significant units of measurement may be derived from these four quantities:

1 Employment Unit: Number of kilo man-hours worked per year ($B \div 1000$). The trend of this unit indicates directly an increase or a decrease in time employment, and is therefore a quantitative measure of employment

2 Productivity Unit: Quantity of product output per kilo man-hour ($C \div$ kmh worked). The trend of this unit indicates an improvement or recession in industrial effectiveness, inasmuch as it is an integration of the energies affecting operating performance

3 Working-Force-Requirement Unit: Average number of workers per kmh ($A \div$ kmh worked). The trend of this unit indicates an increase or a decrease in the number of workers required to perform 1000 hours of work in a year

4 Labor-Utilization Unit: Average number of workers per unit of product output ($A \div C$). The trend of this unit indicates the influence of labor-saving facilities in the saving or wasting of human energy

5 Labor-Performance Unit: Quantity of product output per worker ($C \div A$). The trend of this unit indicates the effectiveness of the working force

6 Labor-Time Unit: Number of man-hours worked per worker ($B \div A$). This unit measures the length of the working period, and the trend indicates fluctuations in the length of the working period

7 Time-Utilization Unit: Number of man-hours worked per unit of product output ($B \div C$). The trend of this unit indicates the influence of time-saving facilities in saving or wasting time

8 Labor-Stability Unit: Total numbers of workers per kmh ($D \div$ kmh worked). The trend of this unit indicates the effectiveness of managerial skill in handling personnel.

Productivity and Employment

The production and man-hour data gathered by the American Engineering Council for the Safety and Production Study in 1926 are so admirably suitable for analysis by the kmh system that the records of 110 companies have been brought up to date by the accumulation of additional data through 1931. These data have been used by the authors in the preparation of a report on technological unemployment recently submitted to the American Engineering Council showing the trends of the eight units for the 110 companies. The value of the kmh system in analyzing operating results is indicated by the discussion of employment, productivity, and working hours that follows.

Measure of Employment. The authors have contended that the number of kmh worked and not the average number of workers employed is the correct measure of the trend of employment.[4] Reports from 110 concerns support this contention. The aggre-

[3] L. P. Alford, Chairman, Glenn A. Bowers, Ernest F. Du Brul, William B. Ferguson, Joshua E. Hannum, Kenneth C. Hogate, Joseph W. Roe, Laurence H. Sloan, and Charles P. Tolman.

[4] Report of Committee on Man-Hour Information of the American Engineering Council on "Proposed Units for Measuring Technological Advance and Technological Unemployment" to Committee on Technological Employment of United States Department of Labor.

gate number of 414,671 kmh worked in 1928 was reduced by 129,533 kmh to 285,138 kmh in 1931, a reduction of 31.3 per cent. Wide differences exist between the percentages of the decreases in the number of kmh worked and the percentages of the decreases in the number of workers employed. The percentages of decrease in the number of kmh worked are, for the majority of companies, much greater than the percentages of decrease in the number of workers employed. The latter are greater than the former for but seven companies; the two decreases are the same for but three companies. The frequency of the ratios, shown in Fig. 1, indicates the wide range of differences between the decreases of the two quantities, the ratio being more than 4 to 1 for eleven companies.

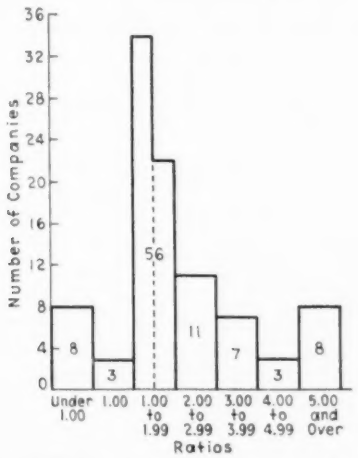


FIG. 1 RATIOS OF DECREASES IN THE NUMBER OF KILO MAN-HOURS WORKED TO DECREASES IN THE NUMBER OF WORKERS EMPLOYED BY 96 COMPANIES, FROM 1928 TO 1931

The character of the quantitative values and relationships is indicative of the unreliability of the number of workers employed as a measure of employment. Variations in the length of the work day and the work week, which are entirely ignored by using the number of workers, are correctly accounted for when the number of kmh worked is used as the evaluator of employment.

Decrease in Employment. During the three-year period from a peak year of prosperity, 1928, to a low year of depression, 1931, decreases in the number of kmh worked range from 1.4 to 86.5 per cent. Only 13 of the 110 companies increased employment; 72 per cent decreased employment from 20 to 70 per cent; the decrease for 9 companies has been from 70 to 90 per cent. (See Fig. 2.)

Increase in Productivity. During a period of depression when employment is being decreased by laying off the less efficient workers, productivity increases at high rates in well-managed plants. Production records of 110 companies from 1925 to 1931 are used to indicate quantitatively the accuracy of this statement.

In this group productivity increased extensively during the period from 1925 to 1928, inasmuch as 72 per cent of the companies effected increases ranging from 1.3 to 88 per cent, whereas only 28 per cent of the companies experienced decreases ranging from 0.5 to 51.5 per cent.[5] Productivity was less than 10 per cent higher for 19 companies, from 10 per cent to 20 per cent higher for 21 companies, from 20 to 30 per cent higher for 18 companies, and more than 30 per cent higher for 21 companies; it was less than 10 per cent lower for 16 companies, and more than 10 per cent lower for 15 companies.

Likewise productivity increased extensively during the three-year period from 1928 to 1931, inasmuch as 58.1 per cent of the companies effected increases ranging from 0.5 to 112.2 per cent, and 41.9 per cent of the companies experienced decreases ranging from 1.1 to 57.6 per cent.

More than half the companies that had a higher productivity in 1928 had a still higher productivity in 1931; more than half that had a lower productivity in 1928 had a higher productivity in 1931. A summary follows.

	Number of companies	Percentage of total
Total number of companies	110	100.0
A—Increase in Productivity:		
1 Productivity in 1928 from 1.3 to 83.5 per cent higher than in 1925	79	71.9
2 Productivity in 1931 from 0.5 to 112.2 per cent higher than in 1928	64	58.1
3 Higher productivity in 1928 followed by higher productivity in 1931	46	41.9
4 Higher productivity in 1928 followed by lower productivity in 1931	33	30.0
B—Decrease in Productivity:		
1 Productivity in 1928 from 0.5 to 5.5 per cent lower than in 1925	31	28.2
2 Productivity in 1931 from 1.1 to 57.6 per cent lower than in 1928	46	41.9
3 Lower productivity in 1928 followed by higher productivity in 1931	18	16.4
4 Lower productivity in 1928 followed by lower productivity in 1931	13	11.8

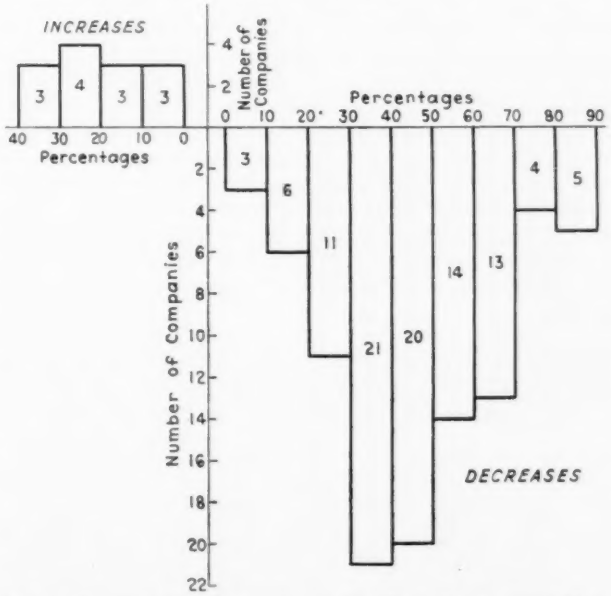


FIG. 2 PERCENTAGES OF INCREASES AND DECREASES IN THE NUMBER OF KILO MAN-HOURS WORKED BY 110 COMPANIES, FROM 1928 TO 1931

Reduction in Kilo Man-Hours Required. The marked advance in technological developments and managerial technique during recent years has had such a pronounced influence upon industrial operation that the changing aspects of the business cycle seem to have little, if any, effect upon continuing progress.

[5] The percentages are derived by comparing the rates of the terminal years of each period, and not by the use of the least square method of determining straight line trends.

When measured by the integrated factor of productivity, a steady improvement may be noted throughout three phases of business conditions: (*a*) a period of stability from 1922 to 1925; (*b*) a period of expansion from 1925 to 1928; and (*c*) a period of recession from 1928 to 1931. In all three periods productivity, as indicated by our studies, has steadily increased. During the period of stable conditions the productivity of 359 companies, employing 254,529 workers, increased 14.4 per cent, the range of increase being from 0.1 to 46.1 per cent for 34 industries.[6] The experience of 110 companies, employing 169,458 workers, already presented, indicates that productivity likewise increases during periods of expansion and recession.

The results of continuously increasing productivity are, either wholly or partially:

a A greater quantity of product will be turned out by the same working force in working periods of the same length

b A constant quantity of product can be turned out by a reduced working force in working periods of the same length

c A constant quantity of product can be turned out by the same working force in shorter working periods.

The greater quantity of product that was turned out in 1928 as compared with that turned out in 1925 indicates the first result during a period of expansion. The previous discussion of the decrease in employment from 1928 to 1931 shows the second result during a period of depression.

The following kmh analysis indicates what the reduced number of kmh required each year would have been during an eight-year period, from 1923 to 1931, if a constant quantity of product output had been maintained by 50 manufacturing companies employing 67,824 workers in 1923, and by two railroad companies employing 51,787 workers in 1923.

The number of kmh required by the 50 manufacturing companies in 1931 would have been individually from 2 to 69 per cent less than the requirement in 1923. In 1923 the 50 companies required 180,767 kmh, which would have been reduced by 56,070 kmh to 124,697 kmh in 1931, a reduction of 31 per cent. Consequently if these companies had maintained a constant output during the eight-year period without shortening the length of the work day or work week, approximately one-third of the 67,824 workers, or 21,127, would have been released. What actually happened was that the employment decreased with some fluctuation from 180,767 kmh in 1925 to 178,498 kmh in 1929, after which it decreased rapidly by 33.6 per cent to 118,515 kmh in 1931.

TABLE 1 ACTUAL AND REQUIRED NUMBER OF WORKERS, NUMBER OF KILO MAN-HOURS, AND LENGTH OF WORK WEEK FOR 50 MANUFACTURING COMPANIES FOR EACH YEAR FROM 1923 TO 1931 TO MAINTAIN CONSTANT PRODUCTION AND TO FURNISH STEADY EMPLOYMENT TO 67,824 WORKERS

Year	Average number of workers actually employed	Number of workers required to maintain constant production in constant length of work week	Actual number of kilo man-hours worked per year	Number of kilo man-hours required per year to maintain constant production	Actual number of man-hours worked per worker per year	Number of man-hours required per worker per year to maintain constant production and steady employment	Average length of actual work week in hours	Average length of work week in hours required to maintain constant production and steady employment
1923	67,824	67,824	180,767	180,767	2670	2670	51.2	51.2
1924	64,295	66,200	164,398	176,433	2560	2610	49.2	50.2
1925	66,789	61,000	171,855	162,962	2570	2400	49.5	46.0
1926	68,629	56,500	166,274	150,350	2420	2220	46.6	42.6
1927	66,492	55,300	156,455	147,142	2360	2180	45.3	42.0
1928	66,621	51,000	163,538	135,894	2460	2000	47.1	38.4
1929	73,300	49,100	178,498	130,819	2440	1930	46.8	37.1
1930	69,808	49,000	155,710	130,554	2220	1920	42.6	36.9
1931	58,211	46,700	118,515	124,697	2040	1840	39.2	35.4

[6] "Safety and Production," A Report by American Engineering Council, New York, 1928, p. 76.

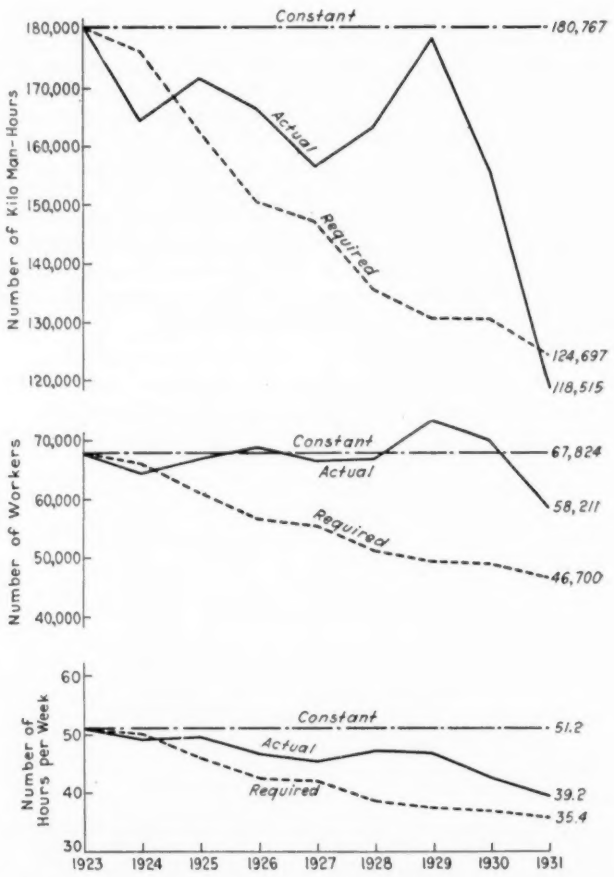


FIG. 3 TRENDS OF ACTUAL AND REQUIRED NUMBER OF WORKERS, NUMBER OF KILO MAN-HOURS, AND LENGTH OF WORK WEEK FOR 50 MANUFACTURING COMPANIES FOR EACH YEAR FROM 1923 TO 1931 TO MAINTAIN CONSTANT PRODUCTION AND TO FURNISH STEADY EMPLOYMENT TO 67,824 WORKERS

The average number of workers employed increased by 8.1 per cent from 67,824 workers in 1923 to 73,300 in 1929, and afterward decreased by 20.6 per cent to 58,211 in 1931.

The values for the actual and required number of workers and the actual and required number of kmh are given in the first four columns of Table 1. Trend curves for the values are presented in Fig. 3.

A similar reduction of 33 per cent in the number of kmh required to maintain a constant service could have been made by the two railroad companies which reported during the same eight-year period from 1923 to 1931. The reduction would have been from 134,131 kmh in 1923 to 90,540 kmh in 1931.

Work Period. The third result of continuously increasing productivity, viz., a constant quantity of product turned out by the same working force in a shorter work period, might have been brought about by the 50 manufacturing companies if the average length of the work week had been shortened from 51.2 hours in 1923 to 35.4 hours in 1931. The actual length of the work week had been shortened from 51.2 hours in 1923 to 39.2 hours in 1931.

Likewise a steady working force could have been maintained for the 51,787 workers employed by the two railroad companies if they had shortened the average length of the work week from 49.8 hours in 1923 to 33.6 hours in 1931.

A summary of the 1923–1931 comparisons for the 50 manufacturing companies follows:

	Per cent
Number of kmh Worked:	
Actual decrease	34.5
Decrease required to maintain constant production	31.0
Number of Workers:	
Actual decrease	14.2
Decrease required to maintain constant production in unvarying length of work period	31.0
Length of Work Period:	
Actual decrease	23.4
Decrease required to maintain constant production and constant working force	31.0

Optimum Operating Performance

At the request of The American Society of Mechanical Engineers transmitted through the American Engineering Council, the Bureau of the Census in the Census of Manufactures for 1929 obtained reports from four manufacturing industries on a basis permitting a kmh analysis of several factors of operation. The industries that reported are:

Blast-Furnace Operation
Machine-Tool Building
Petroleum Refining
Lumber and Timber-Product Manufacturing

The data for the following comparison have been taken by permission from the study and report of the Bureau of the Census, which report is in process of publication by the Government. The degree of coverage for the first three industries is 95 per cent. For lumber manufacturing it is only about 25 per cent, as only those concerns reporting products valued at $100,000 or more were requested to fill in the supplementary man-hour schedule; and further, of those that replied to the supplementary schedule a number showed that the inquiry was not understood, and in consequence their returns were excluded in the tabulations.

The Census report on the four industries is too extensive to review here, therefore comments are restricted to six items:

Physical Volume of Product
Selling Value of Product
Wages
Salaries
Optimum Length of Work Period
Optimum Size of Plant.

Physical Volume of Product. Values for this quantity were reported by three of the industries studied, the exception being machine tools. The data support the conclusion in our previous paper that there is a wide spread of physical volume of production per kmh among plants in the same line of manufacture.

Productivity of Blast Furnaces. The data for 86 blast furnaces are grouped according to number of tons of pig iron produced per 1000 hours of labor by wage earners. Attention is called to the final column, which gives the quantity of product per kmh for each of six groups of plants. The ratio of the lowest rate to the highest rate of productivity is 1:9.

Limits of groups, tons per kmh	Number of establishments in the group	Quantity of product, tons per kmh
Under 200	9	145
200 to 399	22	341
400 to 599	18	504
600 to 799	17	665
800 to 999	13	869
1000 and over	7	1313

Productivity of Petroleum Refineries. The rates of productivity of plants refining crude petroleum as revealed by the same method of analysis of returns from 306 refineries had a ratio of 1:224 from lowest to highest. The data are:

Limits of groups, barrels of crude petroleum refined per kmh	Number of establishments in the group	Quantity of product processed, barrels of crude petroleum per kmh
Under 1,000	9	633
1,000 to 2,499	59	2,008
2,500 to 4,999	97	3,734
5,000 to 9,999	72	5,810
10,000 to 24,999	49	12,709
25,000 to 49,999	13	29,972
50,000 to 99,999	4	56,570
100,000 and over	3	141,829

Productivity of Lumber and Timber-Product Manufacturing. The productivity rates as revealed by the same method of analysis of return from 305 concerns had a ratio of 1:13 from lowest to highest. The data are:

Limits of groups, M fbm per kmh	Number of establishments in the group	Quantity of product, M fbm per kmh
1 to 19	40	13
20 to 29	56	25
30 to 39	54	32
40 to 49	35	45
50 to 74	56	61
75 to 99	20	85
100 to 149	28	127
150 and over	16	173

Value of Products. The paper of 1928 showed wide differences in the selling value of product per kmh as turned out by different firms in the same industry and in different industries. The same situation is revealed in the data from the four industries included in this study. The limits are given in the following tabulation:

Industry	Lowest selling value of product per kmh	Highest selling value of product per kmh	Ratio of lowest to highest value
Blast-furnace operation	$2449	$24,087	1:10
Machine-tool building	918	10,673	1:11
Petroleum refining	4563	192,908	1:42
Lumber manufacturing	468	5,481	1:11

The figures for the four industries showing values of product per kmh by groups are now given.

Value of Product per Kmh

Limits of groups, average value of product per kmh	Number of establishments in the group	Value of product per kmh
Blast-Furnace Operation:		
Under $4,000	7	$2,449
$4,000 to $7,999	25	6,620
8,000 to 11,999	26	10,268
12,000 to 15,999	15	13,804
16,000 to 19,999	9	18,111
20,000 and over	4	24,087
Machine-Tool Building:		
Under $1,000	5	$ 918
$1,000 to $1,499	105	1,568
1,500 to 1,999	82	2,465
2,000 to 2,499	25	3,383
2,500 to 2,999	6	4,623
3,000 to 3,999	4	5,308
4,000 and over	4	10,673
Petroleum Refining:		
Under $5,000	15	$4,563
$5,000 to $7,499	35	6,218
7,500 to 9,999	60	8,671
10,000 to 14,999	88	12,276
15,000 to 24,999	52	17,364
25,000 to 49,999	43	29,539
50,000 to 99,999	9	70,975
100,000 and over	4	192,908

Lumber Manufacturing:		
Under $500	6	$ 468
$500 to $999	70	820
1,000 to 1,499	116	1,244
1,500 to 1,999	80	1,668
2,000 to 2,499	51	2,303
2,500 to 2,999	43	2,709
3,000 to 3,999	26	3,492
4,000 to 4,999	9	4,445
5,000 and over	4	5,481

Wages. The differences in wages paid in the four industries under investigation, as well as the range for each industry, are shown in the following tabulation:

Industry	Lowest wage rate per kmh	Highest wage rate per kmh	Ratio of lowest to highest
Blast-furnace operation	$198	$1,016	1:5
Machine-tool building	464	927	1:2
Petroleum refining	242	1,067	1:4
Lumber manufacturing	149	1,143	1:8

It will be noticed that the highest values are in substantial agreement for all industries, being respectively on an hourly rate basis: $1.02, $0.93, $1.07, and $1.14. The data for the four industries showing wages per kmh by groups are given in the following tabulation:

WAGES PER KMH

Limits of groups, average cost of labor per kmh	Number of establishments in the group	Wages per kmh	Hourly rate
Blast-Furnace Operation:			
Under $300	5	$198	$0.20
$300 to $399	8	355	0.36
400 to 499	15	446	0.45
500 to 599	31	551	0.55
600 to 699	14	636	0.64
700 to 799	9	738	0.74
800 and over	4	1016	1.02
Machine-Tool Building:			
Under $500	28	$464	$0.46
$500 to $599	55	552	0.55
600 to 699	90	654	0.65
700 to 799	39	737	0.74
800 to 899	15	844	0.84
900 and over	4	927	0.93
Petroleum Refining:			
Under $300	7	$242	$0.24
$300 to $399	19	393	0.39
400 to 499	44	469	0.47
500 to 599	109	547	0.55
600 to 699	70	631	0.63
700 to 799	36	745	0.75
800 to 899	18	826	0.83
900 and over	3	1087	1.09
Lumber Manufacturing:			
$100 to $199	5	$149	$0.15
200 to 299	57	263	0.26
300 to 399	84	351	0.35
400 to 499	68	454	0.45
500 to 599	71	552	0.55
600 to 699	72	644	0.64
700 to 799	28	739	0.74
800 to 899	15	856	0.86
900 and over	5	1143	1.14

Salaries. The cost of salaries paid to officers, executives, supervisors, and other salaried people can be taken as an indication of the cost of supervision. There are wide differences in this item in the various industries studied, the spread being:

Industry	Lowest salary rate per kmh	Highest salary rate per kmh	Ratio of lowest to highest
Blast-furnace operation	$32	$186	1:6
Machine-tool building	63	844	1:13
Petroleum refining	40	1056	1:26
Lumber manufacturing	18	339	1:18

The complete data on salaries for the four industries are given in the following tabulation:

SALARIES PER KMH

Limits of groups, average compensation of salaried persons per kmh	Number of establishments in the group	Salaries per kmh	Hourly rate
Blast-Furnace Operation:			
Under $50	15	$32	$0.03
$50 to $74	19	61	0.06
75 to 99	17	86	0.09
100 to 124	17	108	0.11
125 to 149	8	138	0.14
150 and over	10	186	0.19
Machine-Tool Building:			
Under $100	26	$63	$0.06
$100 to $199	78	143	0.14
200 to 299	51	246	0.25
300 to 399	32	304	0.30
400 to 499	16	431	0.43
500 to 599	14	527	0.53
600 and over	14	844	0.84
Petroleum Refining:			
Under $50	18	$40	$0.04
$50 to 99	46	76	0.08
100 to 199	110	147	0.15
200 to 299	68	224	0.22
300 to 399	28	338	0.34
400 to 499	14	470	0.47
500 to 749	16	593	0.59
750 and over	6	1056	1.06
Lumber Manufacturing:			
Under $25	24	$18	$0.02
$25 to $49	89	39	0.04
50 to 74	104	60	0.06
75 to 99	68	84	0.08
100 to 124	37	111	0.11
125 to 149	25	141	0.14
150 to 199	35	171	0.17
200 to 249	13	231	0.23
250 and over	10	339	0.34

Length of Work Period. A comparison of length of work year for the four industries studied is given in the following tabulation:

Industry	Shortest work year, in hours	Longest work year, in hours	Ratio of shortest to longest work year
Blast-furnace operation	1760	4340	1:2.5
Machine-tool building	1940	2970	1:1.5
Petroleum refining	1650	3460	1:2.1
Lumber manufacturing	1450	3000	1:2.1

The spread from shortest to longest is considerable in each industry. The longest for machine-tool building and lumber manufacturing is on the basis of 60 hours per week, for petroleum refining on the basis of 70 hours per week, and for blast-furnace operation on the basis of 84 hours per week. By way of comparison, the shortest are 40, 50, and 60 per cent of the maximum figures. The data for the four industries follow:

VARIATION IN LENGTH OF WORK PERIOD

Limits of groups, average length of work year in hours per worker	Number of establishments in the group	Length of work year in hours	Length of work week in hours
Blast-Furnace Operation:			
Under 2000	3	1760	33.8
2000 to 2249	3	2120	40.7
2250 to 2499	3	2400	46.1
2500 to 2859	16	2680	51.5
2860 to 3332	36	3070	59.0
3333 to 3999	18	3520	67.6
4000 and over	7	4340	83.4

Machine-Tool Building:			
Under 2000	12	1940	37.3
2000 to 2099	5	2090	40.1
2100 to 2249	9	2190	42.0
2250 to 2349	18	2310	44.5
2350 to 2499	64	2440	47.0
2500 to 2674	64	2560	49.1
2675 to 2859	47	2760	53.0
2860 and over	12	2970	57.0
Petroleum Refining:			
Under 2250	5	1650	31.7
2250 to 2349	13	2310	44.5
2350 to 2499	47	2450	47.0
2500 to 2674	50	2560	49.1
2675 to 2859	64	2810	54.0
2860 to 3079	94	2960	57.0
3080 to 3339	20	3160	60.8
3340 and over	13	3460	66.5
Lumber Manufacturing:			
Under 1667	15	1450	27.9
1667 to 1999	25	1910	36.7
2000 to 2099	24	2060	39.6
2100 to 2249	28	2160	41.5
2250 to 2349	51	2280	43.9
2350 to 2499	69	2420	46.5
2500 to 2674	73	2570	49.5
2675 to 2859	66	2780	53.5
2860 and over	54	3000	57.6

Optimum Size of Plants. In our paper of 1928 we presented a comparison on the kmh basis of the physical volume of production for the smallest and largest concerns in 53 industries. The criterion of size was the number of employees. That comparison brought out, among others, these relationships:

The smallest company has a higher rate of production than the largest in 35 industries

The smallest company has a lower rate of production than the largest company in 18 industries

The smallest company has the highest rate of production in 16 industries and a high rate in 5 industries

The smallest company has the lowest rate of production in 6 industries and a low rate in 16 industries.

These comparisons indicate that the optimum size of plant, from the productivity point of view—the one that produces most effectively—may not be the largest. Study of the data upon which this part of our paper is based showed that the most satisfactory unit of measurement to indicate the optimum plant is the number of kmh worked per year, and not the number of workers employed, and that the factors for any industry can be expressed as a range.

This optimum range is obtained by taking into consideration the kmh per year corresponding to the highest values of product, the addition made by manufacture, and the physical volume of product for each of a group of operating factors. Tables 2 to 5 present these data for Blast-Furnace Operation, Machine-Tool Building, Petroleum Refining, and Lumber and Timber-Product Manufacturing.

The relationship of wages to optimum operation is indicated by Tables 6 to 9. In general, the highest wage rate, measured in dollars per kmh, accompanies the highest kmh operating rate, and the lowest wage rate accompanies the lowest operating rate.

The relationship of supervision to optimum operation is indicated by Tables 10 to 13. In general, the highest salary rate per kmh accompanies the highest operating rate, and the lowest salary rate per kmh accompanies the lowest operating rate.

The relationship of length of work period to optimum operation is indicated in Tables 14 to 17. In general, the length of work week, for highest operating rate, reflects prevailing schedules, being, when arranged, about 44 hours for blast furnaces, machine tools, and lumber, and about 50 hours for petroleum refining.

TABLE 2 OPTIMUM SIZE OF PLANT—BLAST-FURNACE OPERATION

Factor per kmh	Range of factor	Value of products	Value added by manufacture	Quantity of products, tons	Kmh per year
Cost of fuel and electric energy	$6000 and over	$28,228	$8211	1514	273.1
Value added by manufacture	5000 and over	18,534	5287	984	346.4
Wage earners	0.400 to 0.449	22,060	5859	1229	393.6
Quantity of products	1000 tons and over	21,922	3393	1313	487.7
Value of products	$20,000 and over	20,811	4329	1357	499.4
Cost of materials	14,000 and over	20,822	3880	1366	512.1
Salaries	150 and over	14,421	2617	756	597.3
Prime cost	15,000 and over	20,672	3532	1137	611.5
Wages	800 and over	17,649	2753	1026	637.3

TABLE 3 OPTIMUM SIZE OF PLANT—MACHINE-TOOL BUILDING

Factor per kmh	Range of factor	Value of products	Value added by manufacture	Kmh per year
Value of product	$6000 and over	$10,673	$6114	29.1
Salaries	600 and over	4,373	2665	41.7
Value added by manufacture	4000 and over	6,874	4887	49.9
Wage earners	0.475 to 0.499	4,186	2733	52.2
Prime cost	$4000 and over	6,665	3139	56.6
Wages	900 and over	4,569	3161	59.9
Cost of fuel and electric energy	60 and over	2,563	1617	77.7
Cost of materials	2000 and over	5,566	2650	80.8

TABLE 4 OPTIMUM SIZE OF PLANT—PETROLEUM REFINING

Factor per kmh	Range of factor	Value of products	Value added by manufacture	Quantity of products, bbl	Kmh per year
Salaries	$750 and over	$67,989	$14,632	29,296	39.3
Wage earners	0.450 and over	34,098	8,354	16,504	44.5
Value of products	$100,000 and over	192,908	15,330	129,010	59.3
Prime cost	100,000 and over	208,159	10,866	141,829	67.3
Quantity of products, bbl	100,000 and over	208,159	10,866	141,829	67.3
Cost of fuel and electric energy	$750 and over	104,055	6,811	68,336	71.3
Cost of materials	50,000 and over	117,751	17,594	79,021	88.5
Wages	900 and over	64,386	14,591	40,899	97.8
Value added by manufacture	20,000 and over	79,990	24,251	40,704	137.1

TABLE 5 OPTIMUM SIZE OF PLANT—LUMBER MANUFACTURING

Factor per kmh	Range of factor	Value of products	Value added by manufacture	Quantity of products, M fbm	Kmh per year
Value added by manufacture	$3000 and over	$4911	$3377	71	47.5
Value of products	5000 and over	5481	2982	23	84.4
Wage earners	0.600 and over	2513	1519	72	219.8
Wages	$900 and over	2668	2208	37	232.2
Cost of materials	2000 and over	3891	1357	110	244.4
Prime cost	3000 and over	3981	1374	115	267.9
Quantity of products	150 M fbm and over	3624	1460	173	276.7
Salaries	$250 and over	2710	1230	67	289.7
Cost of fuel and electric energy	50 and over	2711	1112	82	293.3

TABLE 6 WAGES—BLAST-FURNACE OPERATION

Factor per kmh	Wages for highest operating rate, per kmh	Wages for lowest operating rate, per kmh	Ranges of wages, per kmh
Cost of fuel and electric energy	$658	$321	$750 to $321
Value added by manufacture	587	451	632 to 451
Wage earners	688	339	1012 to 260
Quantity of products	765	369	765 to 369
Value of products	882	302	882 to 302
Cost of materials	898	326	898 to 326
Salaries	635	484	635 to 484
Prime cost	717	299	717 to 299

TABLE 7 WAGES—MACHINE-TOOL BUILDING

Factor per kmh	Wages for highest operating rate, per kmh	Wages for lowest operating rate, per kmh	Ranges of wages, per kmh
Value of products	$774	$568	$774 to $568
Salaries	756	600	756 to 600
Value added by manufacture	713	609	726 to 609
Wage earners	736	667	836 to 577
Prime cost	759	506	759 to 506
Cost of fuel and electric energy	526	623	676 to 526
Cost of materials	708	589	708 to 548

TABLE 8 WAGES—PETROLEUM REFINING

Factor per kmh	Wages for highest operating rate, per kmh	Wages for lowest operating rate, per kmh	Ranges of wages, per kmh
Salaries	$618	$526	$709 to $526
Wage earners	683	505	696 to 495
Value of products	784	572	784 to 517
Prime cost	782	558	782 to 557
Quantity of products	782	528	901 to 528
Cost of fuel and electric energy	643	486	678 to 486
Cost of materials	863	567	863 to 556
Value added by manufacture	773	480	776 to 480

TABLE 9 WAGES—LUMBER MANUFACTURE

Factor per kmh	Wages for highest operating rate, per kmh	Wages for lowest operating rate, per kmh	Ranges of wages, per kmh
Value added by manufacture	$823	$268	$823 to $268
Value of products	475	241	652 to 241
Wage earners	749	414	749 to 414
Cost of materials	572	557	581 to 471
Prime cost	589	252	589 to 252
Quantity of product	674	358	674 to 358
Salaries	544	502	575 to 414
Cost of fuel and electric energy	568	432	568 to 416

TABLE 10 COST OF SUPERVISION—BLAST-FURNACE OPERATION

Factor per kmh	Cost of supervision for highest operating rate, per kmh	Cost of supervision for lowest production rate, per kmh	Range of cost of supervision, per kmh
Cost of fuel and electric energy	$151	$56	$151 to $56
Value added by manufacture	124	61	124 to 61
Wage earners	129	96	161 to 71
Quantity of products	126	56	126 to 56
Value of products	171	52	171 to 52
Cost of materials	167	54	167 to 52
Prime cost	107	50	112 to 50
Wages	148	32	148 to 32

TABLE 11 COST OF SUPERVISION—MACHINE-TOOL BUILDING

Factor per kmh	Cost of supervision for highest operating rate, per kmh	Cost of supervision for lowest production rate, per kmh	Range of cost of supervision, per kmh
Value of products	$912	$ 37	$912 to $37
Value added by manufacture	639	115	639 to 115
Wage earners	540	168	540 to 129
Prime cost	860	95	860 to 95
Wages	559	205	559 to 193
Cost of fuel and electric energy	302	162	342 to 162
Cost of materials	631	152	631 to 152

TABLE 12 COST OF SUPERVISION—PETROLEUM REFINING

Factor per kmh	Cost of supervision for highest operating rate, per kmh	Cost of supervision for lowest production rate, per kmh	Range of cost of supervision, per kmh
Wage earners	$361	$187	$361 to $122
Value of products	337	73	337 to 73
Prime cost	334	218	334 to 55
Quantity of products	334	299	334 to 110
Cost of fuel and electric energy	294	122	294 to 97
Cost of materials	308	212	308 to 80
Wages	171	220	220 to 94
Value added by manufacture	264	99	289 to 96

TABLE 13 COST OF SUPERVISION—LUMBER MANUFACTURE

Factor per kmh	Cost of supervision for highest operating rate, per kmh	Cost of supervision for lowest production rate, per kmh	Range of cost of supervision, per kmh
Value added by manufacture	$130	$51	$130 to $51
Value of products	275	49	275 to 49
Wage earners	95	57	95 to 57
Wages	95	64	95 to 64
Cost of materials	124	75	124 to 58
Prime cost	129	42	129 to 42
Quantity of products	101	66	101 to 61
Cost of fuel and electric energy	107	59	108 to 59

TABLE 14 OPTIMUM LENGTH OF WORK PERIOD—BLAST-FURNACE OPERATION

Factor per kmh	Range of factor	Kmh rate of value of product	Kmh rate of value added by manufacture	Quantity of products, tons	Length of work year, in hours	Length of work week, in hours
Wages	$800 and over	17,649	2753	1026	1810	34.8
Cost of materials	14,000 and over	20,822	3880	1366	2030	39.0
Value of products	20,000 and over	20,811	4329	1357	2040	39.3
Quantity of products	1000 tons and over	21,922	3593	1313	2340	45.0
Value added by manufacture	$5000 and over	18,534	5287	984	2440	47.0
Cost of fuel and electric energy	6000 and over	28,228	8211	1514	2480	47.7
Prime cost	15,000 and over	20,672	3532	1137	2480	47.7
Salaries	150 and over	14,421	2617	756	2610	50.2

TABLE 15 OPTIMUM LENGTH OF WORK PERIOD—MACHINE-TOOL BUILDING

Factor per kmh	Range of factor	Kmh rate of value of product	Kmh rate of value added by manufacture	Length of work year, in hours	Length of work week, in hours
Wages	$900 and over	4,569	3161	1930	37.1
Salaries	600 and over	4,373	2665	2190	42.1
Value added by manufacture	4000 and over	6,874	4887	2260	43.5
Cost of fuel and electric energy	70 and over	2,480	1738	2340	45.0
Value of products	6000 and over	10,673	6114	2420	46.5
Cost of materials	2000 and over	5,566	2650	2460	47.3
Prime cost	4000 and over	6,665	3139	2460	47.3

TABLE 16 OPTIMUM LENGTH OF WORK PERIOD—PETROLEUM REFINING

Factor per kmh	Range of factor	Kmh rate of value of product	Kmh rate of value added by manufacture	Kmh rate of productivity, in barrels	Length of work year, in hours	Length of work week, in hours
Wages	$900 and over	64,386	14,591	40,899	2450	47.0
Cost of materials	50,000 and over	117,751	17,594	79,021	2460	47.3
Value of products	100,000 and over	192,908	15,330	129,010	2480	47.7
Prime cost	100,000 and over	208,159	10,866	141,829	2490	47.9
Quantity of products	100,000 bbl and over	208,159	10,866	141,829	2590	49.9
Cost of fuel and electric energy	$750 and over	104,055	6,811	68,336	2640	50.7
Value added by manufacture	20,000 and over	79,990	24,251	40,704	2690	51.7
Salaries	750 and over	67,989	14,632	29,296	2810	54.0

TABLE 17 OPTIMUM LENGTH OF WORK PERIOD—LUMBER MANUFACTURING

Factor per kmh	Range of factor	Kmh rate of value of product	Kmh rate of value added by manufacture	Kmh rate of productivity, in M fbm	Length of work year, in hours	Length of work week, in hours
Wages	$900 and over	2668	2208	37	1600	30.8
Value added by manufacture	3000 and over	4911	3377	71	1830	35.2
Quantity of products	150 M fbm and over	3624	1460	173	2120	40.2
Cost of fuel and electric energy	$50 and over	2711	1112	82	2160	41.5
Prime cost	3000 and over	3981	1374	115	2210	42.6
Cost of materials	2000 and over	3891	1357	110	2240	43.0
Salaries	250 and over	2878	1711	55	2370	45.5
Value of products	5000 and over	5481	2982	23	2640	50.8

Dicussion

WILLIAM BAUM.[7] The authors, in their first paper on the subject in 1928, aroused favorable comment. They gave management a unified basis for evaluating manufacturing operation through the introduction of the "kilo man-hour," which unquestionably has great advantages over percentages and ratios heretofore used in the measurement of business efficiency. The new paper presents a wider scope in the application of the kmh to the measurement of operating performance, productivity, employment, and various cost elements. The data collected furnish a valuable basis for making comparisons with industries not covered by this new paper. The authors' unique work can be advantageously extended to other phases of managerial control, such as cost accounting and budgeting—not only to make comparisons with previous periods within a certain organization, but also to compare kmh results with other concerns in the same industry.

Direct labor costs expressed in money values per unit of product are meaningless when used for comparisons with previous periods. Variations do not reveal the causes and do not reflect true changes in operating efficiency unless allowance is made for fluctuations in wage rates. Direct labor costs in terms of kmh, on the other hand, give an accurate measure of operation and may be compared with similar figures of other concerns in the same industry. Such direct labor costs are independent of fluctuations of wages and may be broken down to costs by operations, jobs, orders, etc.

Another cost element which often lacks absolute definiteness is "indirect labor." This item is usually expressed in per cent of direct labor cost and therefore is subject to the same criticism. If direct labor wages fluctuate, the indirect labor percentage becomes meaningless. Likewise, if direct labor and indirect labor wages do not fluctuate in synchronism, the indirect labor percentage again is of no value. An absolute measure of indirect labor, however, is given by an expression which relates the indirect kmh to production.

There are other cost elements, such as clerical costs, supervision, selling, administration, etc., which lend themselves readily to similar treatment. Each cost element which has to do with labor in plant and office can be given an absolute measurement by the kmh method.

The present "standard cost system" advocated by progressive accountants is still based on standards expressed in money values per unit of production. The kmh system will some day find universal application in such cost systems. When that day arrives, standard costs will have a definite meaning. No corrections will have to be made for wage-rate fluctuations. A true picture of operating conditions and trends will be at the disposal of cost-controlling officers, leading to higher efficiency and reductions of costs.

The kmh system has been introduced in this organization to evaluate the individual effort of our employees. The standard time for a certain operation as found by time study is brought into relationship to the actual time spent on the job, and this relation expresses the effort of the worker. Each week the relative progress of each employee is followed up and controlled. No other method furnishes a more absolute basis for evaluating and controlling individual efforts.

It is not difficult to visualize revolutionary changes in the measurement of operation and costs. The Management Division of the A.S.M.E. is rendering industry and business a real service in giving the kmh system the publicity which it deserves and which reflects high credit upon its authors.

[7] Director of Personnel and Industrial Engineering, Realsilk Hosiery Mills, Inc., Indianapolis, Ind.

HAROLD V. COES.[8] The authors have demonstrated the value of kmh as an accurate evaluation of employment and as a means of comparing productivity. The reduction in kmh required indicates from studies made that productivity is independent of the phases of the business cycle. What is the significance of this with respect to total employment, particularly in basic industries? It would indicate a continuing but probably more gradual reduction in the kmh to produce society's needs for basic commodities in the stabilized industries. Where does this leave us with respect to labor released in these industries if the conclusions from the facts as presented are sound, and what may be done with the balance?

This study also indicates quite clearly the desirability, in fact the necessity, of reducing working hours per week for constant production, assuming a balance between consumption and production.

It seems, then, if purchasing power is to be maintained, particularly in that group whose annual compensation is $5000 and less comprising 80 per cent of our annual purchasing power, that we must quit thinking of wage and salary scales in terms of a horse-and-buggy age and base compensation on the reduced hours per week and consequently at a much higher hourly base rate.

What are the contributing causes to the wide spread in operating performance? The data pertaining to optimum size of plants and to relative operating efficiency as gaged by kmh seem to check with a study made of relative earnings of large and small industries by Prof. H. B. Summers, of Kansas State College, and reported in the May, 1932, issue of the *Quarterly Journal of Economics*. This study showed that in the nine industrial classifications he studied, comprising 1130 companies, the highest average rate of earnings was found in the smallest size group, including companies with investment below $2,000,000 in five of the nine classes.[9]

W. E. PARKER[10] AND JESS T. HOPKINS.[11] The break down made possible through the use of the kmh unit of measurement provides more specific and tangible basic units than do census figures for comparing costs and industries. This unit of measurement, making possible as it does optimum studies, should provide essential data for planning industrial production.

The kmh units of measurement show trends in employment and the effects of technological changes more accurately than do figures from any other published source and make possible a regular and frequent check of employment trends.

The kmh measurement can be applied effectively to determine seasonal factors in industry. Furthermore, productivity in terms of kmh units and labor stability units give to industry a calibration comparable to that provided by industrial management and engineering for the measurement of the effectiveness of the industrial worker or department.

The kmh system of recording essential data seems so basic and such an improvement over the archaic methods of census reports that its adoption as a common denominator for measurements of labor seems clearly indicated.

E. F. CURTIS.[12] For the past seven years the writer has been working on the same principle, extending its application to three

[8] Manager, Industrial Department, Ford, Bacon & Davis, New York, N. Y. Mem. A.S.M.E.

[9] See the writer's paper, "Dissolving of Concentrated Industries," presented at the Annual Meeting of the Society, New York, N. Y., December 5 to 9, 1932.

[10] Director, Technical Division, Public Employment Center, Rochester, N. Y.

[11] Manager, Public Employment Center, Rochester, N. Y.

[12] Harbirshaw Cable and Wire Corporation, New York, N. Y.

plants of different industrial operations, and has proved conclusively that its use in costing, estimating, and scheduling is efficient, accurate, and most valuable in controlling factory cost during periods of normal and abnormal operating capacities.

The only obstacle that, at the present time, hinders the extension of the general application of this theory is the time necessary to educate others in position to assist and the developments essential for the proper correlation of data.

ARTHUR HUNTINGTON.[13] The agitation for the short week and the split shifts of workers, which has been such a large part of our relief programs, makes this paper particularly valuable. In the face of the stand taken by union labor and by relief workers who are faced with apportioning the few jobs which are available, it would seem advisable that the paper be put in popular form and given wide circulation. Neither the labor leaders nor the relief workers are altogether satisfied with the position they have taken regarding the short week.

It has been stated that shorter hours and split payrolls savor very strongly of the assumption that our maximum living standard has been reached and that we must spread the work necessary to maintain that living standard as widely as possible. Last June the writer presented a paper before the American Society of Agricultural Engineers at Columbus, Ohio, in which it was pointed out that America must decide between:

(*a*) The efficient use of our machinery of production, and
(*b*) The better apportioning of the benefits of production.

There are many people who are advocating the throttling of production until it balances our antiquated machinery of distribution. The engineer has created our machinery of production; he must assume some responsibility toward the apportioning of the benefits of that which he has created. In other words, the engineer has arrived at the place where he must invade the realm which heretofore has been assigned to the business man. He must enter the field of economics.

Thinking labor men are wondering whether it is wise to assume that there is a minimum American standard of living; too many people regard such a minimum as the maximum. Some leaders, even in the labor group, are of the opinion that increased production should be crowded as long as we can find ways and means of apportioning the benefits—in other words, increase the standard of living. Basically this is a sound policy.

[13] Iowa Electric Light and Power Company, Cedar Rapids, Iowa.

TABLE 18

Industries	No. of companies	No. of employees	Wages per kmh	No. of man-hours	Value of product per kmh
Automobiles	12	151,447	$683	382,211,176	$6410
Manufactured ice	6	1,061	796	2,648,845	4240
Explosives	16	3,456	437	8,747,393	3950
Paper and pulp	50	28,832	465	76,603,789	2825
Portland cement	120	35,000	483	97,414,794	2700
Carpets and rugs	4	2,334	605	.....	2640
Lumber	34	14,103	702	35,684,551	2190
Firebrick	3	962	600	.....	1730
Rayon	...	3,500	440	.....	1680
Cast-iron pipe	...	1,783	372	.....	1350
Cotton fabrics	13	11,850	273	31,137,347	1320
Leather shoes	6	6,534	272	16,898,800	1120

TABLE 19

Industries	Total actual wages	Economic wages (by formula)	Actual wages in terms of sales dollar	No. kmh per employee
Automobiles	$261,050,233	$266,109,038	$0.107	2.52
Manufactured ice	2,108,480	1,864,294	0.188	2.49
Explosives	3,822,610	6,072,572	0.111	2.52
Paper and pulp	35,620,761	50,660,996	0.165	2.70
Portland cement	47,051,345	61,498,850	0.179	2.78
Carpets and rugs	.....	.....	0.229	..
Lumber	25,050,554	24,780,522	0.320	2.52
Firebrick	.....	.....	0.337	..
Rayon	.....	.....	0.262	..
Cast-iron pipe	.....	.....	0.276	..
Cotton fabrics	8,500,495	20,821,754	0.206	2.62
Leather shoes	4,596,473	11,480,957	0.243	2.58

The first paper was a long step forward; this paper proves the value of the 1928 paper. It is hoped that this paper will reach well out into the commercial world.

GEORGE G. BERGER.[14] This paper is along the line of statistical treatment which is becoming so popular today despite the emphasis of modern science on more exact inductive methods. Statistical treatment is founded on the hopes that the plus quantities balance the minus quantities and that therefore the conclusions drawn are good bases for prediction purposes. The statistical method does not result in good controls except in cases where large tolerances are economically permissible.

There can be no doubt that costs and values of production on the basis of 1000 man-hours supply valuable comparisons. The writer likes this method of bringing together facts, but does not believe that we can disregard certain assumptions made. It is valuable to know that a plant does not produce as much per kmh as some other plant or that labor cost per kmh is higher or lower, but these comparisons only raise the question as to why this is so.

Some of the dangerous assumptions on which the comparisons made are based are as follows:

The effectiveness of the management is disregarded; in other words, by the statistical treatment it is assumed that the managements of the industries noted are representative and uniformly efficient. Also the paper does not bring out the effect of financial backing on raw material cost per kmh and on equipment policy which affect productivity per kmh. It is important to note, however, that equipment does not take the place of common sense in management, and occasionally we do find small concerns producing more economically.

The most dangerous assumption, however, is that actual wages paid are adequate economic wages. This assumption disregards the condition of one enterprise paying low wages and benefiting by the trade conditions which are due to the competitor and to other industries paying higher wages. In other words, the paper stresses a low labor cost per kmh for non-union labor and for low paying industries. This irresponsibility for buying power is perhaps the primary cause of our present state of affairs; it prevents any real comparisons of productivity or labor costs on the straight kmh basis.

Needless to say, industry is not paying adequate wages. In proof of this statement the writer wishes to submit a study of the responsibility for buying power in 16 groups of manufactures for the year 1929. This study shows three groups more than meeting their responsibility; two are nearly meeting their responsibility; the remaining groups are engaged in overcoming the advantages gained. The net result is a removal of $2,036,000,000 buying power by manufacturing industries in the year 1929.

From studies made on buying power, it was immediately apparent that wages bear a distinct relationship to the sales dollar or price, which could be expressed by formula. Applying this formula to the data in Tables 1 and 4 of the older paper, "A Basis for Evaluating Manufacturing Operation," the writer obtained economic wages per kmh which should be paid if industry is to share its responsibility for trading conditions.

Something is radically wrong when an industry pays $272 per kmh for labor charges and the worker is compelled to work 2.58 kilo man-hours in order to earn a living. One kilo man-hour is the same as 1000 men working one hour, or one man working 1000 hours. An ordinary working

[14] Consulting Industrial Engineer, New York, N. Y. Jun. A.S.M.E.

year of 268 days of 8 hours each would amount to 2.14 kilo man-hours. In other words, the worker is paid $272 for approximately half a year's work and is compelled to work considerably more than 8 hours a day in order to earn a living.

It seems to me that before we compare labor costs per kmh, or productivity per kmh which competitive labor costs are bound to influence, we must first determine what percentage of the sales dollar should go into wages and salaries in the specific case—i.e., distribute the responsibility of paying adequate wages.

TABLE 20

Industries	Wages per kmh	Economic wages per kmh
Automobiles	$683	$697
Manufactured ice	796	703
Explosives	437	695
Paper and pulp	465	660
Portland cement	483	631
Carpets and rugs	605	...
Lumber	702	693
Firebrick	600	...
Rayon	440	...
Cast-iron pipe	372	...
Cotton fabrics	273	669
Leather shoes	272	680

M. W. La Fever.[15] Basically the kilo man-hour provides correct measurement of the trend of employment. The writer is interested in the worker as an individual as well as in the amount of real employment. Therefore, he feels that labor problems should be so developed for the public at large as to express the problem when possible in terms of men. For this reason, to discuss generally employment or unemployment in terms of man-hours leaves out the element of the individual, although it is the most impressive of all employment expressions. Further, when the term "worker" is used, it may mean one of two things: First, a person who is actually on the job; second, a person who is duly listed on a payroll and who under ordinary circumstances may or may not be working. Thus, the two possible meanings of the term "worker" seem to be used respectively in expressing quantity *A* and quantity *D*.

It seems to me that quantity *A*, "average number of workers employed," would be more distinctive and not so easily confused if the term were changed to read "average number of jobs furnished." It is obvious that the "average number of workers employed" is controlled by the number of jobs furnished from day to day, and consequently the suggested change would immediately cause one's mind to visualize the situation, while "the average number of workers employed" without further explanation immediately creates a desire to know exactly what is meant and just how such an average would be obtained. It appears that little change would be necessary to revise the entire discussion on this basis. Changes would only be necessary in units 3, 4, 5, and 6.

Unit 3 could be either "working force" or "job requirement unit." The trend of this unit would indicate an increase or a decrease in the number of jobs per 1000 hours of work in a year.

Unit 4 could be designated either as the "labor" or "job utilization" unit. The trend of this unit would indicate the influence of labor-saving facilities because it would show the available number of jobs used to complete a unit of product.

Unit 5 would be the "job performance unit." It would show the quantity of product output per job. It is obvious that in the course of a week, a month, or a year several workers could have been employed on a single job; but the trend of the unit would still indicate the effectiveness in the performance of the job.

Unit 6 would be either the "labor" or "job-time unit." The trend of this unit would indicate fluctuation in the length of the work period during which the average job was furnished.

[15] United States Department of Labor, Bureau of Labor Statistics, Washington, D. C.

The writer has avoided using the term "available jobs," because he can imagine a condition where jobs might be available, but because of a shortage in the labor market the jobs would not be filled. If such condition prevailed regularly, the term "average number of jobs furnished," as suggested, would equally apply. However, it seems difficult, at least at present, to imagine a shortage of labor.

The motive, as stated, is one to create in the mind of the reader a consciousness of the individual worker, and this is commonly associated when one speaks of a job. However, it will be noted that the use of an expression such as "jobs furnished" leaves the mind open to the element of labor turnover and tends to obviate the necessity of determining just what is contemplated by the word "worker."

There has been considerable discussion, pro and con, regarding the use of figures in the manner in which Table 1 has been compiled. Similar figures have been used recently by the Bureau of Labor Statistics to express the changes which have occurred in certain industries as a result of technological development. So far as is known, little criticism has come from the outside regarding such figures. However, they have been discussed among economists in the bureau. They point to the statement that, in 1830, more than a century ago, had there been no improvement in the method of carrying water in the city of London, half of the London population at that time would have been water carriers. The argument seems to be that to assume a constant production and to apply to that quantity a change of productivity does not take into consideration that portion of increase in productivity which has been made possible by the increase in production resulting from greater demand following decrease in the cost of producing and consequent lowering of consumer price.

There are probably instances where production has remained constant or so nearly so that this factor would be minimized, but, by and large, the factor of elasticity of demand is always working whether quantity production increases or remains constant. Hence, to assume a constant anywhere would be unreal.

No way of getting around this situation has been found, and it is felt that the least questionable method may be to confine the more general presentations to the changes in productivity.

In the same manner that the assumption of a constant production is open to question and criticism, the assumption of a constant working force or work week would also be open to question and criticism.

Statisticians, because they deal with facts more than with the theory of probability, are prone to avoid such assumptions, and consequently are constantly exerting such influence as they can to prevent the compilation of figures based on such assumptions. However, no better method has been found of depicting the probable trend in the immediate future than by the presentation of figures based on such assumptions.

The kmh method from the point of view of factory operation and management will furnish a more intensive analysis of factory operations and of factory efficiency than any method which has come under observation. Its application as a method of measuring statistical trends in employment and the elements of technological changes appears also to surpass any method yet proposed. However, labor and the public in general are interested in men as individuals and in the number for whom work is furnished by employers. Thus, in adapting it for lay thought we are at all times confronted by a demand for statistics expressed in terms of individual men, because it is the unit of which the public ordinarily thinks.

It seems also that there should be some definite discussion concerning the method or process of determining quantity *C*, "prod-

uct output." To elaborate on the construction of the trends of the various elements without a suggestion as to the method or process of determining "production" leaves a loophole for criticism among those acquainted with certain types of manufacturing but unacquainted with the possibilities of the kmh method.

The writer feels that it is possible to develop a measurement of production for any plant or any product even among the service industries, but he is often confronted, when discussing the method with others, by a question, for example, How would you measure production in a foundry, one floor of which is devoted to castings weighing tons and another floor to bench work, with perhaps a large amount of die casting on the side? Hence, if a hypothetical case suggesting a method of determining an adequate measurement of production were included, the whole presentation would be more convincing to the uninitiated. Diversified production seems to be the bugaboo which causes the average person to hesitate in indorsing the general application of the kmh method.

On the theory that things equal to the same thing are equal to each other, the writer has worked up a hypothetical case of determining the composite trend of productivity for several plants manufacturing unlike products. (See Table 21). At a given

TABLE 21

Plant No.	Average employment	Product	1922	1923	1924	1925	1926	1927	1928
			Production per kmh						
1	5000	Motor vehicle, cars	2.14	2.12	2.60	2.55	2.71	2.90	3.1
2	2000	Motor vehicle, cars	1.16	1.40	1.60	1.70	1.90	2.30	2.20
3	200	Woolen goods, yd	400	410	412	389	410	420	425
4	1000	Wheelbarrows	200	210	220	240	260	280	300
5	5000	Cement, bbl	1000	800	750	600	650	500	450
6	2000	Shoes, pr				1000	900	800	700
			Productivity Index						
1	5000	Motor vehicle	100.0	99.1	121.5	119.2	126.6	135.5	144.9
2	2000	Motor vehicle	100.0	120.7	137.9	146.6	163.8	198.3	267.2
3	200	Woolen goods	100.0	102.5	103.0	97.3	102.5	105.0	106.3
4	1000	Wheelbarrows	100.0	105.0	110.0	120.0	130.0	140.0	150.0
5	5000	Cement	100.0	80.0	75.0	60.0	65.0	50.0	45.0
6	2000	Shoes				100.7	90.6	80.6	70.5
			Weighted for Combining						
1	5000	Motor vehicles	500,000	495,500	607,500	596,000	633,000	677,500	724,500
2	2000	Motor vehicles	200,000	241,400	275,800	293,200	327,600	396,600	534,400
3	200	Woolen goods	20,000	20,500	20,600	19,460	20,500	21,000	21,260
4	1000	Wheelbarrow	100,000	105,000	110,000	120,000	130,000	140,000	150,000
5	5000	Cement	500,000	400,000	375,000	300,000	325,000	250,000	225,000
6	2000	Shoes				201,400	181,200	161,200	141,000
Total	15200		1,320,000	1,262,400	1,388,900	1,530,060	1,617,300	1,646,300	1,796,160
			Composite Productivity						
All plants			100.0	95.6	105.2	100.7	106.4	108.3	118.2
Plant 1 and 2			100.0	105.3	126.2	127.0	137.2	153.4	179.8
Plant 1, 2, and 3			100.0	105.2	125.5	126.2	136.3	152.1	177.8
Plant 1, 2, 3, and 4			100.0	105.2	123.6	125.4	135.5	150.6	174.4
Plant 1, 2, 3, 4, and 5			100.0	95.6	105.2	100.7	108.8	112.5	125.4

moment it is assumed that in the broad sense progress has generally reached similar development in our industrial structure (although it probably is never so); hence, the average production per kmh in one industry should be comparable at the same time to the average kmh production in another. To date no one has suggested sufficient reasons to invalidate these assumptions. "Average employment" may be taken to mean either man-hours or full-time jobs; for purposes of theory it is immaterial, as its main function is to demonstrate the handling of plants of unlike size. It is assumed that production per kmh has been developed to start with in each case. The shoe plant was not added until the fourth year, and hence was assumed to be 100.7 per cent productive compared to the year of origin, because that appeared to be the stage of progress which the other five plants had reached. Notice the recessional productivity of the cement plant. Does this not involve the same principles used in compiling data for Table 1?

WALTER N. POLAKOV.[16] The value of the paper is not so much in the method as in the signification of the examples marked out. As far back as 1922 the writer said that "The time rate of man's work is the only measure of production which is of the same dimensionality . . . the scales may be as varied as are the activities under consideration themselves." There is nothing new in that idea, but its application brings to light new and illuminating data and relations. Studying the industrial data in their relation to the man-hour, the authors succeeded in presenting a bold picture, incomplete, to be sure, due to lack of suitable statistics, but believed to be indicative of the new trend.

The cry of today is for leadership out of the welter of fundamental contradictions in which our economic life finds itself. Yet the leadership is not forthcoming, and, it is feared, is due to the lack of full understanding that the old machine age is dead and a new power era is upon us.

Consider for a minute these staggering ratios of the lowest rate to the highest rate of productivity within an industry: Blast furnace, 1 to 9; lumber and timber, 1 to 13; petroleum, 1 to 224, or more especially consider the value of product per 1000 man-hours varying from $9918 to $192,908, and the conclusion that the old standard cannot fit the new relations is inescapable.

In the past we believed that there is a relation between the number of men employed and the volume of output. Today this notion is junked. In the past we believed that there is a relation between the investment and the value of the product. Today we know that for a few dollars' worth of competent counsel and perhaps a slight rearrangement in management, productivity may be doubled and trebled.

Mr. A. W. Mellon recently remarked that we have a "left-over" leadership of men who still think that today's problems are a continuation of the past and do not see that it is new. Perhaps Mr. Mellon can see that it is a new power era clearer than others, because in his own aluminum industry the power use increased in the decade 1919–1929 by 140.5 per cent, and correspondingly wages rose 123 per cent and the number of employees increased 86.1 per cent. Neither is the aluminum industry an exception in this direction. We have studied 40 industries. The trend throughout is unmistakable. Employment and wages have gone up as the use of power and advanced technology progressed during the past decade.

Now we have come to the point of curtailing production as the financial charges and overhead overtook the purchasing ca-

[16] Consulting Management Engineer, New York, N. Y.

pacity of the majority of our population. If we have increased productivity of labor per man-hour, does it mean that we must reduce the "man" component or the "hour" component? The answer seems clear, but the action was in the opposite direction, and we got the worst and the most ominous unemployment.

But reducing number of hours to 1400 per year per man will not establish the desired "balance" between the increased productive capacity and buying power unless the economic scissors—the divergence of prices and wages—is reduced and the scissors are closed.

G. S. RADFORD.[17] The authors' earlier paper on the kmh method impressed the writer with a profound respect not only for the kmh of intellectual effort that went into its creation, but also with its implied promise of future richly helpful and timely revelation. These impressions are bound to be accentuated by the authors' subsequent work as evidenced by this paper.

In fact, as events have transpired, there appears such a degree of fortuitous and fateful timeliness in their work as to make its rapid extension intensely practical, opportune, and necessary. They should be encouraged to extend the analysis as rapidly as possible and to give their interpretations of its implications, at least in general terms, for the more important or key industries.

JOS. A. PIACITELLI.[18] The authors have established eight units of measurement. One of the most significant of these, and perhaps the most difficult to apply correctly, is the "productivity unit." In following the productivity values through, the authors point out that the ratio of the lowest to the highest rate of productivity is 1 to 9 in blast-furnace operation, 1 to 13 in lumber manufacturing, and 1 to 224 in petroleum refineries. Now, before we can determine the degree of credibility in these cases, we should try to get behind the scenes and find out why these ratios exist.

The authors are not very clear in their expression "quantity of output." Can these ratios be due to the fact that the physical volume of the product is expressed in actual units when it would be more desirable to express it in "equivalent" units if a concern manufactures more than one product of similar substance? Are they due to variations in time of processing or the size of the production unit? The quantity of output is not always expressed in corrected equivalent or compound units which would place the various classes of products on the same basis of measurement.

About four years ago, shortly after the authors presented their first paper, the writer interested an executive in making an analysis of the productivity of one of his plants, month by month. In establishing the compound unit for this analysis approximately 150 products were first grouped into classes on the basis of the process involved, and then relative weights were assigned to each class on the basis of kilo man-hours and in terms of the most common class of product. For example, if class A, the most common class of product, is assigned the value of 1 and if it requires 0.6 as many man-hours to produce a ton of class B products as it does of class A, then class B will have a weight of 0.6, and so on. With the compound unit thus established, the amount produced each month is plotted on ratio paper along with the kilo man-hours required to produce them. In this way a month-to-month comparison is being made of the productivity of this plant.

It occurs to the writer that it would be wise to enlist the aid of the various A.S.M.E. professional divisions in establishing the compound unit of volume with respect to kilo man-hours for the various groups of products in each industry. With these kmh relationships for the various classes of products as a basis, it should not be a difficult matter to group them into a compound unit and make reasonable comparison of concerns in one industry. We must certainly be able to do this before we attempt to compare one industry with another.

The authors point out the effect of productivity on working periods, and if the results of their studies of the industries selected are representative of America's industrial structure, then the shorter work week is here by necessity and here to stay. The six-hour day and five-day week, wherever practicable, are not far off. We will either have that or continued unemployment difficulties.

L. W. WALLACE.[19] The writer is greatly interested in this paper, for it represents fundamental research and indicates an important method of making a more effective analysis of industrial operations. The authors' thinking is receiving wide study and application. It is interesting to know that Congress has instructed the Department of Labor to make a continuing study of technological employment. Something over a year ago the Secretary of Labor organized a committee to study the problem and advise the department as to the line of studies that would be the most beneficial. Its work disclosed that there was real need for a new approach to the analysis of industrial operations; hence the committee approved the Alford-Hannum method and included it in its report to the secretary. The report of the Committee on Technological Employment has been printed by the Government, and is recommended to the consideration of any one concerned with the broad problem it deals with, and as an evidence that the Alford-Hannum method is receiving consideration by governmental officials.

Mr. Eaton and Mr. Alford challenged the writer's statement relating to technological employment. The writer was thinking of one phase of the problem and they were thinking of another. If they will carefully read the report, they will get the full import of what the writer was saying and would then probably not disagree with the statement. This is another evidence of the need for the standardization of definitions. In this case there is need for a clear-cut definition of what is meant by technological employment. It is rather fully discussed in the report referred to.

AUTHORS' CLOSURE

The comments in the discussions are, in the main, so favorable to the paper that elicited them that little needs to be said by way of closure. However, a few additional comments are justified. They will be given in the order in which the occasion for them appears in the discussions.

Baum emphasizes the value of applying the kmh method to cost accounting, budgeting, and the analysis of direct and indirect labor costs. He says, "Each cost element which has to do with labor in plant and office can be given an absolute measurement by the kmh method;" and, referring to standard cost systems, "The kmh system will some day find universal application in such cost systems."

Curtis' discussion is along this same line when he says that he has "... proved conclusively that its use [kmh system] in costing, estimating, and scheduling is efficient, accurate, and most valuable in controlling factory cost during periods of normal and abnormal operating capacities."

The authors are conscious of these possibilities, have done some work in these fields, and in their next presentation intend to make a prominent feature of the application of kmh to cost accounting.

[17] Consulting Engineer, New Canaan, Conn. Mem. A.S.M.E.
[18] Consulting Engineer, New York, N. Y. Assoc-Mem. A.S.M.E.
[19] Executive Secretary, American Engineering Council, Washington, D. C. Mem. A.S.M.E.

Coes turns to the implications of the findings in the paper, pointing out certain indications. The " . . . gradual reduction in the kmh to produce society's needs for basic commodities in the stabilized industries;" " . . . the necessity of reducing working hours per week for constant production, assuming a balance between consumption and production;" and " . . . base compensation on the required hours per week and consequently at a much higher hourly base rate." Radford refers to the possibility of interpretation of the findings and hopes the authors will do this ". . . at least in general terms, for the more important or key industries." Polakov also directs his discussion to the significance of the examples.

The authors are in full sympathy with these comments, and have felt strongly the urge to offer explanation and interpretation of the findings brought forth by the kmh analysis. Hitherto, they have rigidly excluded all such comment, for the reason that they particularly wished to present their method of analysis and to show its possibilities. The data used and findings disclosed were incidental to this main purpose. They believe, however, that from now on more attention should be given to the findings, their implications and meanings, for the kmh method now seems to be established as one of the working tools of the engineer and industrialist.

Parker and Hopkins, La Fever, and Wallace give strong approval to the application of the kmh method to the analysis of employment and unemployment. The first two state, "The kmh units of measurement show trends in employment and the effects of technological changes more accurately than do figures from any other published source, and make possible a regular and frequent check on employment trends;" and " . . . a common denominator for measurements of labor seems clearly indicated." La Fever opens his discussion by saying, "Basically, the kilo man-hour provides correct measurement of the trends of employment." Wallace refers to the consideration being given to the kmh method in governmental circles, and points out that there is " . . . real need for a new approach to the analysis of industrial operations."

This generous recognition of the value of the kmh method in the study of employment and unemployment is especially gratifying to the authors. In spite of the tremendous amount of work done in studying all aspects of industrial labor, the need for a satisfactory analytic method had not been met. If the kmh method realizes its seeming possibilities in this field, we shall be richly repaid for the work involved in its development.

Huntington hopes " . . . that this paper will reach well out into the commercial world." It is too soon after its presentation to hazard a prediction as to whether this hope will be realized. However, the kmh paper of 1928 attracted far more attention among business men than among engineers.

Berger lists and objects to several assumptions which he says are in the kmh analysis: "The effectiveness of management is disregarded;" the " . . . effect of financial backing on raw material cost per kmh and on equipment policy which affect productivity per kmh" is not brought out; and "The most dangerous . . . is that actual wages paid are adequate economic wages."

These assumptions are Berger's and not the authors'. In the entire kmh disclosure there has been but one assumption. That is in the paper of 1928 and is: " . . . that on the average an hour of work in one industry, or in one plant, is the substantial equivalent in human effort (mental, manual, or combined) of an hour in another industry or plant."

Berger also includes a table of economic wages, taking his data from the 1928 paper. He does not give his formula whereby "economic wages" are calculated, and for that reason little can be said in reply. However, the creditability of the data in the 1928 paper is not high, as was plainly stated at that time. For this reason, alone, considerable doubt must be expressed as to the value of Berger's computed figures.

La Fever and Piacitelli point out the difficulty of expressing the "productivity unit" in physical quantities. This difficulty is real in some situations, particularly where products must be combined. However, the authors believe that skilful experimentation in any situation will yield a satisfactory solution of the problem.

La Fever suggests certain changes in the names and designation of the units involving the "worker." He prefers the term "job." The authors are grateful for this suggestion, which seems to be sound. When the terms in the paper were "coined," the authors were trying to give expression to new concepts in connection with employment, and were never entirely satisfied with their final selection.

La Fever objects to the setting up of a constant production factor in the analysis of rates of change. It is true that this constant is theoretical and not factual. However, the period of time to which it is applied is so short that surrounding production factors will have varied from simple rather than from complex causes, and its use gives a striking presentation of the rates of change of the associated factors. The kmh system is based upon a study of rates. If these rates are to be projected to indicate possible happenings, some bases of prediction must be selected.

La Fever presents in Table 21 a study of composite productivity. These composites he builds up from index numbers. The method he uses seems to have standing among statisticians. The authors have been unwilling in their work to apply mathematical processes to index numbers, considering the latter as expressions of relationships only, not as quantities. La Fever's study, therefore, while interesting, seems questionable because of the method used. However, his objective is highly commendable.

Progress in Materials Handling

REFLECTING general business conditions, activity in the material-handling field has been considerably restricted during the past year. This restriction has occurred in spite of the low prices of equipment and the evident savings possible with suitable material-handling facilities.

The sale of all kinds of machinery, of course, has been abnormally low, but the sales resistance to processing equipment is somewhat less than that to material-handling equipment. It is easier to see the need of a machine actually to make a product than of a piece of equipment to cut costs between machine operations. But often the lack of material-handling equipment more than sacrifices the savings made by an efficient processing machine. A double saving is possible, in many cases, when the installation of a new machine is accompanied by new material-handling facilities.

Savings Possible by Improved Handling

It is not necessary, however, to wait until new manufacturing machines are purchased to make improvements in material-handling methods. Many of the instances of substantial savings have occurred recently in plants that installed handling equipment without any change in processing machines aside, perhaps, from a new layout. In fact, there is scarcely a plant now operating at a loss, or at an extremely small margin of profit, that could not better its situation by improving its handling methods. Better handling may turn operating losses into profits. Even highly efficient plants make large savings by changes in material-handling methods. One large automotive plant, noted for its excellent handling facilities, at one time saved $275,000 in four months ($825,000 annually) by installing new handling equipment and changing the layout and methods at the beginning of its chassis-assembly line.

A conservative estimate of the savings possible by improvements in material-handling equipment ranges from 2 to 20 per cent of the total shop-labor payroll, direct and indirect. It is not impossible for the actual handling operations within a plant to total 60 per cent of the total work and the total labor cost, notably in plants that are concerned largely with simple processing of raw or semi-finished materials where packaging or repackaging constitutes a large part of the work. A plant of this kind, at present lacking in suitable material-handling equipment, can save as much as one-third of its labor cost by installing such equipment. One-third of 60 per cent is 20 per cent, the saving in total labor cost that probably can be brought about under such conditions. Higher savings are possible in cases where no attention has been paid previously to material-handling methods.

Progress Hampered by Two Factors

Material-handling progress is hampered from two sources—one within and one without industry. In both cases the obstruction is based on lack of knowledge and on unfortunate misconceptions.

Within industry the trouble arises in those managements controlled by officers or directors who originate from non-industrial fields. Banking, legal, commercial, and even political connections and obligations sometimes place enterprises under the influence of such groups. Plant administration is thus directed more along financial lines than along sound industrial and economic channels. The stock-market crash, the difficulties facing many mergers that got into trouble during the past two or three years, and the widespread readjustments of capital structure that occurred (partly through accounting measures and partly through actual reduction of stock values, but both acknowledging actual capital losses), all attest to this misdirection of emphasis. If the efforts and funds thus overconcentrated on financial matters had been expended, in part, on improving material-handling facilities in factories and shops, the profits made could have been permanent and not evaporative. Material-handling proposals meet too much sales resistance in the executive offices and board rooms for the general good of industry, and the resistance often is not based on good business judgment nor on a knowledge of the needs of the plant.

From without industry, the obstacles in the way of extensive use of material-handling equipment arise among those who are conscientiously laboring to solve the unemployment problem but in a field that unfortunately they know little about. It is not wise to pay much attention to agitation recommending a slowing up of technical and manufacturing progress in the effort to increase employment. Such agitation emanates from persons little informed on the fundamentals of industry and its possibilities of development.

Economics and Material Handling

In the first place, wealth in the economic sense comes only from the agricultural, extractive, and production activities of a nation or a group of people. Certain facilitating services, such as transportation, communication, finance, etc., add place, time, or "fluidity" utility, but are not creators of wealth. The correct distribution of wealth among those who produce it, or facilitate its production, gives purchasing power on which the ability to satisfy demand is based. There must be a balance between purchasing power and price so that each group of persons will be able to acquire goods and services sufficient to satisfy their reasonable needs and maintain or improve their standards of living. Anything that reduces the cost of making a product, and consequently its economic selling price, brings it within the purchasing power of wider groups of people. Material-handling equipment reduces such costs, and therefore fulfils an important economic function.

Even in those industries that ultimately may reach current saturation points—where the existing population of the nation or the world cannot absorb more without waste, because all needs are satisfied at prices practically within the reach of all—a further saving by cost reduction will place commodities on the market at lower selling prices, but with the same, or greater, total profits and greater percentage profits to the makers. Surely this is worthwhile, for why should we spend more time and labor on work than is necessary?

Material-handling equipment certainly tends to reduce the amount of human energy expended in the making of products. In this sense, of course, it cuts down employment. We could go in the opposite direction, however, as did the Canadian road builders who purposely reverted to pick-and-shovel work on a big job in 1932, and celebrated the increased employment of hand labor by burying a miniature steam shovel, the symbol to them of a technical progress that had brought about widespread unemployment. But one speaker, in commenting on this move,

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Presented at the session of the Materials Handling Division, Tuesday morning, December 6, 1932, during the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Personnel of the Executive Committee of the Materials Handling Division for 1932: J. B. Webb, Chairman; M. W. Potts, Secretary; C. D. Bray, F. L. Eidmann, E. D. Smith, G. L. Morehead; Associates: R. H. McLain, G. E. Hagemann, F. D. Campbell.

pointed out the huge wastes that would follow a general adoption of such a plan, and the huge costs it would entail on the taxpayers. Quite properly he added, "Why didn't they use toothpicks on this job! Toothpicks would move even less earth than the shovels!"

It is not the job itself, as such, that is the pivotal factor, but the job *economically* done. The creation of wealth in the form of goods or services that can be distributed to everybody is the important factor in employment, not the mere employment itself. A foundry could engage a small army of men to carry pig iron by hand, unit by unit, from freight car to cupola; sand, bucket by bucket, to the mixing and molding machines; molten iron, ladle by ladle, from cupola to flask; but who would pay the exorbitant price for castings made under such methods? If we would not go back to plans like these because we have found ways cheaper and better than such antiquated methods, why not continue to go forward?

Herein lies the trouble: The need exists for some sort of control over technical developments so that plans can be made in advance to take care of the labor that will be displaced. Industry has social obligations to give useful employment to all who are qualified and desirous of employment in industry. A forecasting of the employment shifts that must follow imminent technical developments must be made. Perhaps strong trade associations can be formed to which advance notice of changes can be made, in confidence. Then, perhaps through a central federal employment bureau, with state and municipal branches, arrangements can be made to absorb the workers who probably will be displaced. Unemployment insurance may have to be provided to carry over those who cannot be readily readjusted. Extensive training facilities will be imperative not only to train men to fit the changes in occupation that they must undergo, but also to give all workers greater training to enable them to become more adaptable. Lack of such means at present is the cause of the "45-year-old man" problem in industry. The training courses must be conducted in part by industry and in part by universities and state, county, and city boards of education.

New developments will absorb some of those displaced by technical improvements and compelled to seek other employment. Others who are displaced will be absorbed by industries making the equipment that caused the displacement of labor. The increasing demands for service facilities, which multiply rapidly with the growing complexity of society, will provide employment perhaps for the remainder displaced. Service facilities include public works which may have to be increased or decreased to aid conditions. Furthermore, the workday has been gradually shortened in industry and further shortening of the workweek can be of assistance. Individually, the workers displaced from the factories may find employment in the service or distribution channels of these industries. The man losing his job in an oil refinery may open a gasoline station; the man laid off by an automobile factory may become a garage mechanic. The difficulty is not that technological progress causes unemployment, it's the *delay* in making the adjustment that really causes the unemployment and the resultant suffering.

Foreign Competition

There is another angle to the situation that is not often remembered. To a certain extent, nationalism does influence social developments. We necessarily are limited in the comprehensiveness of our undertakings because of the differences in climate, language, habits, customs, political background, and racial characteristics of the various countries. Each nation develops largely within itself because it cannot go outside and reconcile its political, social, and racial characteristics with those of other countries. There is usually no common basis of interchange of ideas, such as exists internally, that completely overcomes national barriers. However, technical progress can jump all barriers even though the products made through such progress cannot so readily surmount such obstacles. Hence, active, progressive industries within a nation push technical and industrial progress to the limit of their ability, to secure commercial and trade advantages and not be outstripped by other nations. Should we hamstring technical progress at home and therefore lose pace with what is going on abroad? Even friendly national competition would hardly countenance such a course. Should we deprive our people of the benefits of progress while those of other nations enjoy and profit by advances in science, invention, and the industrial arts? None would agree to this plan. Those who would put on the brakes in the interest of employment should realize that this step will have the directly opposite effect, and will throw our workers out of jobs while those of foreign nations will be the ones most benefited.

Material-Handling Equipment Does Not Have to Cause Unemployment

The foregoing analysis is presented to remove the misconception that material-handling equipment must cause unemployment. It naturally causes some *shifts* and readjustments in employment. It is "labor-saving equipment" in that it removes much of the drudgery from industrial work, and performs many tasks that would be impossible—certainly economically impossible—without it. But it is preeminently "cost-reducing equipment." Material-handling equipment manufacturers and users should clearly recognize the distinction and cooperate in efforts to stamp out the false, even though well-meaning, propaganda against the development and installation of such kinds of equipment. To clear up any existing misunderstanding on this point, the material-handling formula itself has been renamed the "cost-reduction equipment formula" or "economic equipment formula" and is no longer called the "labor-saving equipment formula."

The Material-Handling Formula

This formula, useful for productive-machine equipment as well as for material-handling devices, is not applied anywhere near to the extent that it should be by users and prospective users of material-handling equipment, who could discover worth-while savings from proposed new equipment through this medium. It is not employed as much as it should be by manufacturers of material-handling equipment, who could stimulate sales if they would train salesmen in its simple arithmetical calculations, and in the related cost-finding work applicable to clients' material-handling problems. The simple formulas, expanded in 1925 and refined in one or two subsequent studies, will yield all the results needed in most cases. In fact, when applied to material-handling equipment, they often point out savings of far greater extent than when applied to many productive-machine installations upon which some user was easily "sold," and to his benefit. Further developments in the formulas, suggested by Prof. Fairfield E. Raymond and H. L. Vorlander in their paper on "Economic Life of Equipment," presented before the Society in December, 1931, still await the cooperation of manufacturers and users of material-handling equipment who are asked to present concrete cases out of their own experience to Prof. Fairfield E. Raymond, Massachusetts Institute of Technology, Cambridge, Mass., for investigation and solution. Eventually, a series of simple formulas for various groups of cases will evolve from Professor Raymond's studies of such examples.

Sales Problems in the Industry

There are two troublesome problems facing installation of

better material-handling equipment and methods. The first lies partly within and partly without the industry and causes the second. A material-handling installation usually involves more than the selling and placement of a unit of equipment to do a certain piece of work. While many productive machine tools are standard and can be sold on the basis of performance as units, without a study of more than the particular operation that they are to do, a large percentage of material-handling installations, on the contrary, require an engineering survey to develop a system that will serve a number of operations—perhaps the entire manufacturing cycle—satisfactorily and economically. A personal, detailed investigation by a sales engineer, and sometimes considerable designing and calculation, are required before a proposal can be submitted and a price quoted. Prospective users have not hesitated to avail themselves gladly of such free services from a number of competing material-handling manufacturers, only one of whom could get the contract, with a profit sufficient to give him only a normal return on his venture. The losses to the other manufacturers, in engineering services and drawings, have had to be met at the expense of their own business development, or else have been spread over succeeding jobs secured, or equipment sold with little engineering service. A general regulation of services to the submitting of general proposals and prices, subject to engineering revision upon securing the contract, or to a charge for any actual engineering survey made for the prospective user prior to submitting bids, is required for the stabilization of the material-handling industry. Under present circumstances, the user cannot be blamed for adopting for his installation a number of ideas submitted in competing surveys and drawings, or for agreeing to have such surveys made, and then deciding not to make any installation, or to do the work himself.

The second problem is caused by certain "marginal producers" who unwisely introduce into the industry the same problem that is faced by other industries, namely, unwarranted price cutting, which is to be deplored because it is uneconomical in the end for both manufacturer and user.

New Equipment Developed

The reports received from manufacturers indicate that not as much new equipment was developed during the past year (September, 1931, to October, 1932) as was the case in previous years. This condition was to be expected on account of business conditions. Manufacturers of consumer products often could sustain business activity by introducing new products, but manufacturers of equipment for industrial use were not so fortunately situated.

Conveyors[1]

The Lamson Company lists three new devices developed since September, 1931. The first is a "Traylift," a conveyor designed primarily for handling trays of dishes vertically, either ascending or descending. It consists of a so-called Uni-strand vertical conveyor, and is motor-driven at the upper end, the final driving medium being a lower chain and sprockets. Devices are provided on this drive to protect the machine and operators in case of accident. The main carrying chain has arms at suitable intervals to provide a capacity of six to eight trays up and down per minute. The loads do not pass over either the upper or lower carrying sprockets, but are carried up to a predetermined level and, if they are not removed, the machine is automatically stopped. A similar arrangement is provided on the down-bound travel.

The second device is a "Trayvator," an elevator or lowering unit for the handling of trays. It consists of a reciprocating unit so arranged that the placing of the load, for instance at the bottom on the platform, will automatically start the machine and it will travel to the upper level and automatically stop and remain in this position until the load is removed. A similar arrangement is provided for down-bound travel. This unit may be made fully automatic, or entirely manually operated, or semi-automatic. The operating medium is an electric motor with suitable drive connections, while the raising and lowering element consists of an endless chain, thus insuring continuous motion in both the up and down directions. This unit may be built as an entirely self-contained device which may be installed quickly and effectively in locations where the service demands are not such as to require the use of the more elaborate continuous machine.

The third device is a "Selectoveyor," a conveyor that automatically selects commodities to insure their delivery to a predetermined station. This type of machine presupposes the use of a uniform-sized tray or box on which an indicating mechanism is mounted that may be sent by the operator to the predetermined delivery point. Upon its dispatch to the vertical conveyor, this indicating mechanism automatically sets another mechanism on the car, which, upon its arrival at the predetermined destination, automatically throws into operation a discharge station. This discharge station operates through a very small angle and, having no close clearances nor intermeshing arms, the box or tray is automatically discharged and the car upon its continued travel again resets the unloading station to the inoperative position. This machine provides for the utmost flexibility and may be applied to any number of floors in a building up to a present maximum of twenty. In connection with this machine, automatic loading of trays from a loading station outside the shaft containing the conveyor may be provided. Similarly, completely automatic safety devices are provided to insure the safety of the machine and operators in case of non-delivery of a tray or box, or of improper dispatch by an operator.

The Mathews Conveyor Company recently made two interesting installations of material-handling equipment. The first is that furnished for the Hastings Potato Growers Association's packing shed at Hastings, Florida. The system includes a belt conveyer, potato grader, vegetable washer, drier, etc. This is the first set-up used primarily for the washing of potatoes on a wholesale scale. The farmers' wagons and trucks back up to the platform and laborers dump the potatoes on to a conveyor, which carries them to the dirt and undersize eliminator. From there they are discharged to the receiver end of the washer and, after traveling through an impeller wash, they are discharged to the drier, which consists of a battery of blowers mounted over a rolling roll conveyor that automatically turns the potatoes over as they pass under the current of air. This action removes loose water from the potatoes, and from the point of location of the blowers to the discharge end of this unit the potatoes have ample time to dry before entering the grader. Fig. 1 shows the blowers in position, mounted over the top of the rolling roll conveyor. Picking out of imperfect potatoes is done alongside this rolling roll conveyor, pickers being stationed on each side of the unit. This installation has a capacity of 1100 bushels per hour. The present models have capacities ranging from 6 to 20 cars per day.

A most unusual application of heavy-duty roller conveyors, is that of a 90-deg twisted-spout conveyor furnished the Sharon Steel Hoop Company, Sharon, Pa. Fig. 2 shows the 90-deg twisted spout and vertical Y-sections of the heavy-duty roller conveyor.

[1] The data on new equipment were obtained through the kind assistance of representative men in the various fields, and by means of about 200 questionnaires mailed to individual manufacturers. Information on conveyors was collected under the direction of the following: Gravity conveyors, Frank E. Moore, president, and W. J. Ramsey, of the Mathews Conveyor Company; continuous conveyors, Jervis B. Webb, president, Jervis B. Webb Company.

Fig. 3 is a quarter-sectional view of the Mathews Type No. 100-B bearing and roller construction. This design has a continuous load rating of 8000 lb per roller. It is the largest of the line. The smallest has a continuous load rating of 150 lb per roller, and there are various sizes in between these two, having load ratings of 600 lb and 2000 lb per roller.

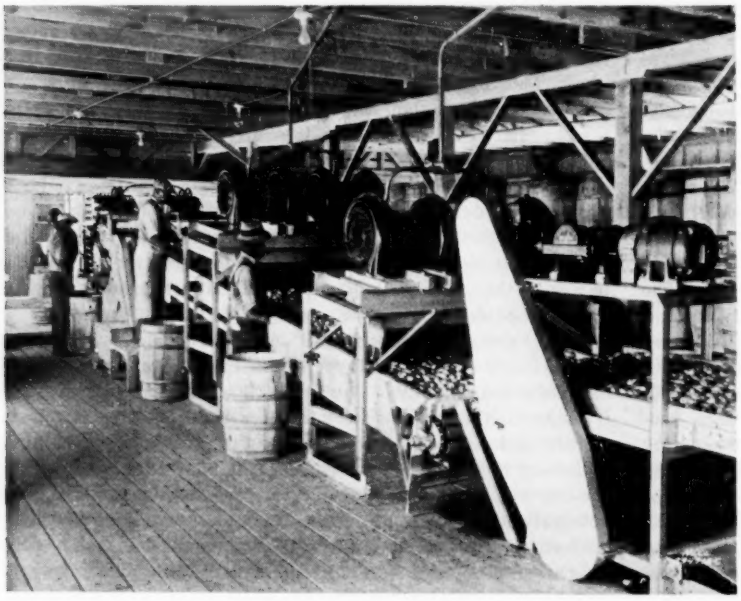


FIG. 1 BLOWER AND ROLLER-CONVEYOR SECTION OF A POTATO-WASHING INSTALLATION

The Standard Conveyor Company has developed rollers and bearings for conveyors, of which Fig. 4 shows one of the latest. In this design there is a heavily constructed ball bearing, protected both front and back, and using steel shields as retainers for the seal. The center of the inner shield is bent inward, forming a funnel shape to keep the grease from working out through and into the roller. The seal is a special but inexpensive grease which is forced through the end of the shaft. The shaft is crossdrilled to allow the grease to come in between the shaft and the inner ball race, then through the inner ball race to the chamber between the bearing and the outer shield. The rotation of the roller and outer part of the bearing carries a sufficient quantity of grease to the inside chamber, thus completely surrounding all working parts with an effective seal designed to eliminate wear. The time for replacing or flushing the bearing varies from 12 to 24 months. When the grease is replaced, any dirt or grit that may have accumulated around the outer edge of the steel shield is automatically flushed out. In addition to keeping out foreign matter, this type of seal also protects the bearing from rust. The hexagon-shaped shaft furnishes a positive lock for the inner ball race. Although this lock is particularly important when the bearing is not protected, it is also an important item where rollers are spinning idle after the commodities have passed over and the inner unit tends to slide around and would wear down the shaft.

A one-piece roller is also furnished in the lighter gages for use in handling light commodities. It is important to have free-running rollers and ones that are concentric so that the lightest commodity will start the rollers with the least possible resistance, because on long lines not enough pitch may be available to overcome high friction. These rollers are of one-piece construction and there are no inserts to get loose. Their rounded smooth ends adapt them for loading and unloading from the sides. The rollers are also used as idlers for belt conveyors.

The Sandvik Conveyor Manufacturing Company recently installed a steel-belt noodle cooker, of unique design, in the plant of the LaChoy Food Products Company, of Detroit, to produce high-quality chow mein noodles. A fabric-belt conveyor delivers the uncooked noodles to the steel belt (see Fig. 5). By the use of snub rollers, the steel belt dips down into a 20-ft long tank of hot vegetable oil, thence operating horizontally through the cooking oil and submerging the entire stratum of noodles, which have been uniformly deposited on the belt. The vegetable oil has a temperature range of 250 to 325 F. The steel belt is 24 in. wide and the approximate center distance between drive and take-up pulleys is 35 ft. Belt speed is variably controlled, ranging from 2 ft to 8 ft per min.

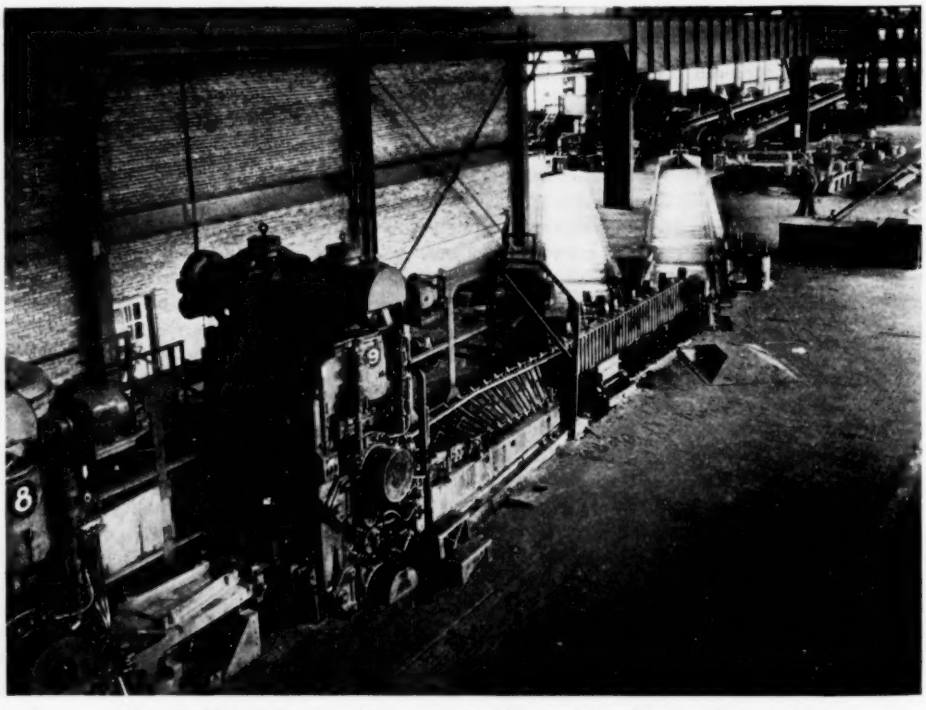


FIG. 2 NINETY-DEGREE TWISTED-SPOUT HEAVY-DUTY ROLLER CONVEYOR

At the unloading or discharge end, the steel belt operates on an incline, thus carrying the cooked noodles out of the tank, and thoroughly draining them of oil, which returns by gravity to the tank. The noodles are then automatically carried by belt conveyors to the point where they are packed in containers. The overhead traveling chain, to which rakes are attached at intervals, operates directly over and at the same speed as the steel belt. This chain, with the rakes, keeps the noodles in position on the steel belt while traveling through the hot oil and also up the incline at the discharge end of the unit.

Steel skirts are provided at each edge of the steel belt for the length of the tank, thereby forming a perfectly tight trough. This arrangement prevents any spillage of noodles into the tank and consequently the purity of the oil is not lowered and its period of usefulness is greatly increased. This feature is an attractive element of the equipment, because any product spilled into the tank would settle to the bottom and burn, thereby necessitating frequent cleaning of the tank and renewal of the cooking oil. The steel belt is kept clean at all times by automatic means provided for this purpose, thus removing any possibility of rancidity or taint to the product.

Previously noodles were cooked by intermittent overhead basket-type conveyors, and were hand-dipped into a kettle for cooking. The output per hour by the old method was 60 lb; it is now 250 lb. The saving in labor effected by the installation is 40 per cent, in oil used, 30 per cent, and in fuel consumption, 50 per cent.

A chip conveyor on the new "American" 100-hp, 22-in. lathe developed for testing metal-cutting carbides, handles 15 cu ft of chips per min, or more than 2 tons per hr. The chips are fed to the conveyor by a sloping bed and chip pan.

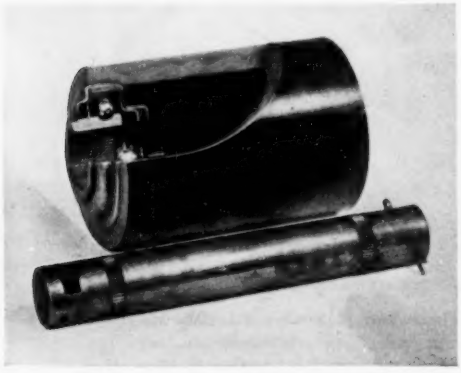


FIG. 3 SECTIONAL VIEW OF A NEW CONVEYOR ROLLER WITH SEALED BEARING

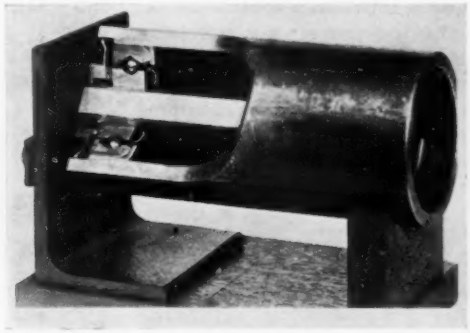


FIG. 4 ANOTHER TYPE OF ROLLER WITH SEALED BEARING

SPEED REDUCERS AND DRIVES[2]

The Palmer-Bee Company reports that, with many satisfactory speed reducers now available, the progress of development of the past year has dealt more particularly with refinements and improvements of design and construction rather than with the design of entirely new machines. Present conditions demand economy, both in first cost and operating cost. However, there exists a tendency on the part of both buyer and manufacturer to hold first cost so much in view that it completely overshadows operating cost. This trend can be, and probably has been, carried too far in many instances. True low cost can be described only in terms of cost per unit of capacity per year. In this item must be included first cost, maintenance charges, operating charges, and depreciation and obsolescence based on the useful life of the machine.

With these facts in mind, the company has carried on investigations of existing designs, and has developed new and desirable characteristics in its machines. Studies of tooth surface tension, zones of contact in worm and herringbone gears, and materials and their heat treatments, have made possible a new procedure in design which is based on the prevention of tooth pitting and inordinate wear, without recourse to special lubricants. The breaking value of the teeth is considered, but it is based on the fatigue limits of the materials used. From the breaking point of view, all of the company's H-series speed reducers will now safely carry 200 per cent of rated load and still be within safe allowable working stresses. Even these high values, while they protect the machines from failure through tooth rupture, are not sufficient without proper consideration of tooth wear. Stated simply, the contact pressure between the meshing teeth may be so great as to cause pitting of the contacting surfaces. This action rapidly leads to wear, and although the teeth will not break off, they will wear out. This wear may not cause immediate shutdown of the equipment, but it does materially lower the efficiency of the unit and will necessitate eventual shutdown for repairs. The company now rates its gears to give full and proper consideration to the contact pressure, based on the materials used in the gear and pinion, the zone factor existing between the gear and pinion, and the radius of curvature of the teeth.

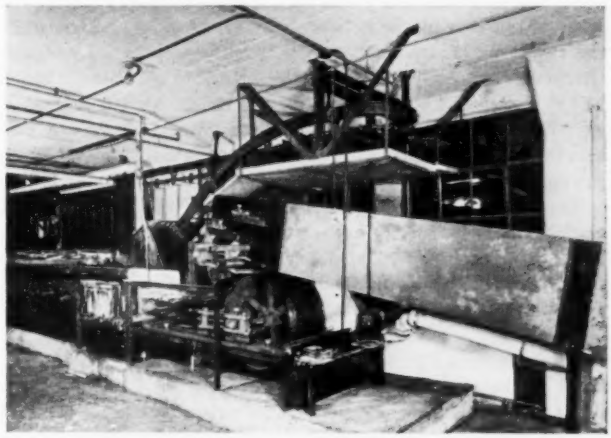


FIG. 5 STEEL-BELT NOODLE-COOKING UNIT

To a great extent, gear efficiency and life depend on the materials used for gears and pinions, and tooth form. They also depend upon the ratio between the two members, gear and pinion, and the number of teeth in the pinion. In its designs, the company no longer attempts to obtain the maximum possible ratio from a given number of gear steps, but first considers efficiency in determining the maximum ratio for one-, two-, or three-stage reduction.

Special attention has been given to the application of anti-

[2] Information collected by William E. Bee, president, Palmer-Bee Co.

friction bearings to speed reducers. In connection with bearing mounting, secondary stresses have been eliminated by getting away from constrained types of mounting. This plan has effected economies and given much better load factors in the bearings. The method of lubrication employed for either standard or special high-speed units has not been changed.

The Stephens-Adamson Manufacturing Company has developed a new type of variable-speed transmission known as the JFS transmission. Hardened steel disks are splined on two shafts. One is driven by the motor or other power source, while the other transmits the power to the machine to be driven. A series of rollers is placed between each of the disks and transmits the power by means of the traction between polished surfaces. Originally, this traction was set by means of set screws, which had to be tightened a trifle from time to time. This arrangement gave a constant pressure for all loads, light and heavy. A cam arrangement is now placed on each side of the machine to increase the pressure as the load is increased.

Some tests have recently been made on the latest type of machine with two disks on one shaft and three on the other. These tests indicate an efficiency ranging from 85 per cent to 95 per cent, with pressures ranging from almost 0 up to 300 lb, depending on the load transmitted. The speed ratio is changed by moving the rollers between the two shafts. The normal variation is from 2.3 to 1, past 1 to 1, and up to 1 to 2.3. The drive can be arranged to give a uniform range of about 5 to 1. When once set, the speed ratio remains constant until the position of the rollers has been changed to give a new output speed.

Mechanical Handling Systems, Inc., has designed an automatic load control for conveyor drive units, and the like, which can be used either as a means for stopping the conveyor in case of a jam, or as a device for synchronizing multiple-drive units when the conveyor is driven at more than one place. For this latter purpose it can be used in any equipment where it is necessary to synchronize the speed of multiple drives.

The device, Fig. 6, is used in combination with a variable-speed transmission, and, in accordance with a predetermined load variation, automatically speeds up or slows down the drive. This action is accomplished through a balancing arm installed

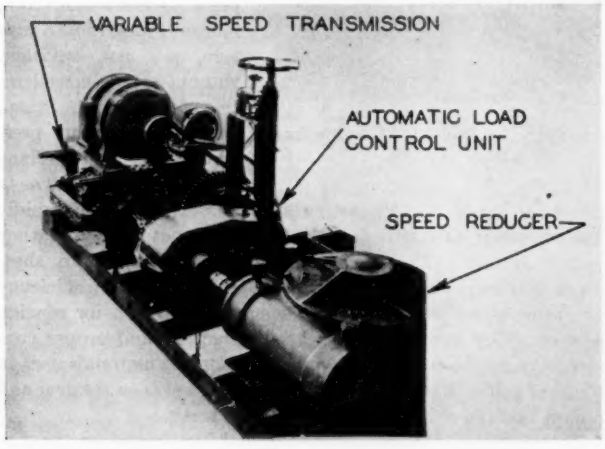


FIG. 6 LOAD CONTROL FOR CONVEYOR DRIVE UNITS

between the variable transmission and the reduction unit. This balancing arm is counterweighted to sustain the normal conveyor load. When this load varies up or down, the balancing arm reacts immediately upon the electrical remote control device that regulates the speed of the variable transmission. This arm is so sensitive that a fractional variation in load is measured at once and compensated for. When the control is used as a safety device, any dangerous increase in load imposed on the machine or conveyor, due to accidents or other reasons, will cause the balancing arm to operate a limit switch cutting off the power.

The Jervis B. Webb Company has introduced a new development in synchronous conveyor drives, described in a paper on "Application and Design of the Floating Drive for Conveyors," by N. H. Preble, engineer of the company, presented in December, 1931, and published by The American Society of Mechanical Engineers.

TRAMRAIL EQUIPMENT[3]

The 1932 developments reported by the Cleveland Tramrail Division of the Cleveland Crane & Engineering Company are as

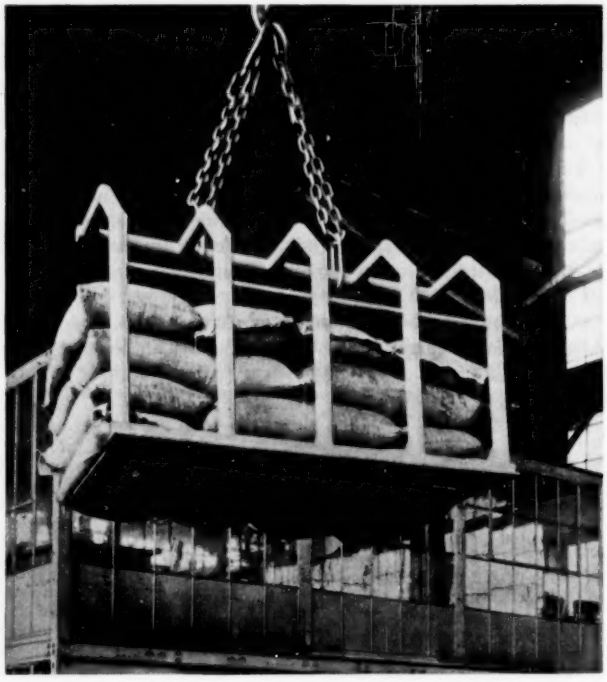


FIG. 7 GRAB FOR HANDLING BAGGED MATERIALS

follows: The arch-beam rail system was further developed by adding the Series 2400 arch beam, also called the 8-in. arch beam, comparable in all respects (except as to loading) with the original 10-in. Series 2500 beam. This 8-in. arch beam is designed for use as a self-supporting rail for handling of light loads where it is not practical or economical to suspend the regular rail from superstructure. The arch beam consists of a composite section. The lower portion is a high-carbon rail and the upper portion is scalloped, or arched, by slitting a Bethlehem I-beam, which is then welded to the lower high-carbon rail of the arch beam. New suspension fittings have made possible the further application of flexible suspension. The arch beam, being a self-supporting member on a longer span, is suspended on alloy-steel hanger rods, the upper ends of which are provided with a rounded head fitting in a cupped ceiling fitting, providing a ball-and-socket suspension. The lower portion of the hanger rod is threaded and fitted with a rounded nut that engages with a hardened-steel, inverted-concave washer, which is part of the coupling assembly, thus incorporating the ball-and-socket method of suspension in the

[3] Information collected by A. F. Anjeskey, sales manager, Cleveland Tramrail Division of the Cleveland Crane & Engineering Company.

lower fitting of the suspension system as well as in the upper fitting. This suspension, and the design of the couplings that join the various lengths of arch beam, requires less headroom than the conventional method of fittings attached to the upper flange of an overhead rail. Electrically welded steel sections have been substituted for castings, etc. in various parts of the tramrail systems, thus increasing their strength and effecting economies in manufacture.

The company has also designed a grab, Fig. 7, for the handling of bagged materials. This grab consists of a framework provided with tines at its lower end, which are inserted between the spaces usually formed when the edges of bags are piled close together. The grab is designed to handle approximately one ton of bagged material. In the case of sugar and similar heavy commodities, sixteen 100-lb bags can be placed on the grab, delivered to the storage point and deposited by withdrawing the grab. When later reclaimed from stock, the same unit load of 16 bags can be handled by reinserting the grab between the bag spaces. Tests have indicated that, even with bags in storage for a considerable length of time, the space between the bags is always large enough to permit reinserting the tines of the grab the full length of the bags. In using the grab, it is necessary to pile the bags in such a manner that they interlock, forming a unit of as many bags as the grab is capable of handling. This grab can be used also for handling any objects that, when piled close together, leave a space between them for the insertion of some modified form of the grab.

The Osborn Manufacturing Company, during the last year, through its Materials Handling Division, has developed, and is now in the process of testing, a new overhead tramrail line, to be known as the Osborn Tramrail Systems. It is complete in every way to fill the needs of overhead handling up to 6000 lb capacity, with either hand-operated or motor-propelled trolleys. The lift on the trolleys may consist either of chain block or electric hoist. These lifts can be furnished in either hook-on or built-in designs.

The new system has minimized headroom loss to an extreme. It has simplified erection by its very complete group of ceiling fittings, by jig drilling all rails at the factory, and by having the correct rail for all span conditions. It is built entirely of steel, with large factors of safety. Switches, turntables, and transfer bridges have safety equipment, and the electrical equipment meets the Underwriter's requirements, as well as the industrial codes of the various states. Standardization of rail widths has been accomplished by selecting a width that permits the system to tie up with other systems. The plant has been tooled up completely for mass production with precision methods, and is further continuing its research.

A New Principle in Bulk Handling[4]

During 1932, an American manufacturer, The Pneumatic Scale Corporation, Ltd., secured the patent and manufacturing rights to a conveyor that had been placed on the market successfully in England a few years before and of which several specially designed installations had been made in Europe and Africa. The principle of this conveyor was discovered by chance. Arnold Redler, the inventor of the equipment—the Redler conveyor—was a flour miller who had devised a seasoning method for flour in bulk, and reclaimed the flour by having the floor of the vertical-sided seasoning bin alive with chains.

When a careless night miller had loaded a bin from the back, leaving a space at the front with the chains bare, Mr. Redler halted the filling operation, got into the bin, stood on the chains, and ordered the unloading to start, so that he could observe the interaction of chain and flour at close range. Something seemed to be wrong, however. The chains were slowly moving under him, as usual, but unloading did not start. Puzzled, he kept pace with the live floor and stooped to see what was going on. Suddenly something bumped him from behind and he found himself between the end wall of the bin and an advancing slope of flour. He made a quick exit to safety, but the peculiar action that he had seen so impressed him that he turned his attention to a study of conveying, with the result that he became the manufacturer of a new kind of material-handling equipment.

The law discovered by Mr. Redler may be paraphrased briefly as follows: In bulk-material handling, if the boundary layers are



Fig. 8 An H-Type Horizontal and Vertical Conveyor for Bulk Materials

urged forward, the entire mass moves as a unit because the shearing strength of the material on itself is greater than the friction of the same material against the relatively smooth walls of the conveyor conduit. Applying this principle, Mr. Redler made a conveyor out of a square conduit, equipped with a longitudinally moving element having relatively light members sweeping all four sides. This turned out to be capable of conveying material *en masse* and quiescent at any angle in any plane. Later, he discovered that this conveying could be done with more finished design without complete four-side sweeping. The common shapes assumed by the transverse elements today are the "H," the "U," and the "L."

This equipment is now used to unload silos of foundry sand, vertically sided bins of hot, ground cocoa, 75-ft-high bins of flour, and for conveying and elevating cement, coal, cocoa, sugar, starch, bauxite, coffee, beans, ceramic clays, abrasive soap powder up to 20 per cent moisture, flaked breakfast foods, grains, bran, flour, semolina, rolled oats, and cookies. Inasmuch as there is almost no relative movement of one particle on another, the most fragile of these products is carried with practically no degradation.

In Fig. 8 is shown an H-type horizontal and vertical Redler

[4] Description prepared by Nixon W. Elmer, chief engineer, Redler Conveyor Division of the Pneumatic Scale Corporation, Ltd.

conveyor. The loading point is shown on the horizontal section and the unloading may be at any desired height and on any of three sides of the vertical leg, or, for that matter, on the horizontal portion. Partial discharge at any or all of these points is obtainable. In the horizontal portion, the H-element is laid on its side with the cable midway of the crosspiece of the H. The design of the transverse element is such that it automatically centers itself in the material, remaining out of contact with the casing. Under laboratory conditions, it has been possible to keep the transverse elements from ever touching the curved section at the bottom of the vertical while making the 90-deg change of direction. Friction at this point is negligible in commercial installations. In this case, one simple machine conveys both horizontally and vertically. The space occupied is small and no pit is required.

The first commercial installation of this conveyor made in the United States handles up to 330 bushels of uncleaned wheat per hour, over a distance of 340 ft, not quite in a straight line, and requires 2 hp for operation.

Where the material to be conveyed is of such a character as to corrode or chemically injure the cable, either a complete, flexible waterproof covering is molded over the cable or the cable itself is constructed of an 18-chrome, 8-nickel steel.

Material	Ground Coffee	Ground Coffee	Ground Coffee	Ground Coffee	Ground Coffee	Ground Coffee	Whole Beans	Ground Coffee	Whole Beans	Wheaties (Flakes)	Cocoa	Whole Beans	Ground Coffee	Whole Beans
Test No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Weight of Material Lbs. Per Cu. Ft.	28.5	28.5	28.5	28.5	28.5	28.5	54.0	28.5	54.0	12	21.5	54	28.5	54
Speed of Conveyor Feet Per Minute	28	38	28	30	28	28	28	28	28	28	28	28	28	28
Angle of Repose	35°	35°	35°	35°	35°	35°	29°	35°	29°	55°	35°	23°	35°	23°
Wt. of Mat. Delivered Per Min. (Lbs)	126	176	132	146	135	136	252	136	252	48	104	264	128	196
Capacity Factor	79%	79%	81%	84%	84%	84%	87.5%	84%	87.5%	70%	85%	86%	79%	64.5%
Actual Vol. Delivered Per Section (Cu. Ins.)	92	92.8	94.5	97.6	97.5	98	96	98	96	82.3	99.3	201	185	150
Theo. Vol. Del. Per Section, No Slippage (Cu. Ins.)	116.4	116.4	116.4	116.4	116.4	116.4	116.4	116.4	116.4	116.4	116.4	239.5	232.8	232.8
Press. on Diaphragm Mat. Moving (Oz. Per Sq. In.)	.634	.634	.646	.646	.646	.646	2.58	.646	2.60	.162	.646	2.58	1.54	4.85
Press. on Diaphragm Mat. Stationary (Oz. Per Sq. In.)	.313	.313	.323	.323	.323	.323	1.29	.323	1.29	.081	.323	.807	4.85	1.29
Total Watts Per Lb. Per Min.	2.980	2.100	3.080	2.600	2.660	2.340	2.140	2.640	1.990	4.160	2.50	1.97		
Ft.-Lbs. of Useful Lift in Terms of Watts	.625	.860	.625	.680	.625	.625	.625	.625	.625	.625	.625	.625		
Watts Per Lb. Per Min. (Frictional)	2.359	1.240	2.455	1.920	2.035	1.715	1.515	2.015	1.365	3.535	1.87	1.35		
Wattage (Loaded)	370	370	370	380	360	320	540	360	500	200	260	520		
Wattage (Empty)	250	250	260	200	200	200	200	200	160	160	160	160		
Weight of Sample (Oz.)											19			
Wt. of Breakage (Oz.) in Sample Before Test											.5			
Wt. of Breakage (Oz.) in Sample After Test											.5	8	Spacing	
Remarks		Same as Test 1		Same as Test 3			Same as Test 6	Diameter of Inlet Same as Outlet	Same as Test 8	Same as Test 8	Diameter of Inlet and Outlet 4½ x 4½	Same as Test 8	Diameter of Inlet Same as Test 13	Same as Test 13
Size of Flights (Ins.)	5x5	5x5	5x5	5x5	5x5	5x5	5x5	5x5	5x5	5x5	4½x4½	5x5	5x5	5x5

Fig. 10 Test Chart of the 5-In. Conveyor Shown in Fig. 8

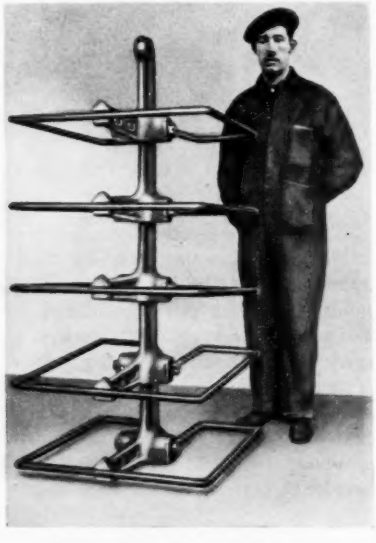


Fig. 9 Moving Element of a Conveyor Handling 400 Tons of Wheat per Hour

With this machine, when loading from a choked feed, an absolutely uniform delivery is obtained by putting a short sheet of steel between the two runs beneath the hopper opening. The upper run then acts as a measuring and feeding device for the lower run. This method assures a loose and uncompacted load on the carrying side, which is of importance with sticky substances.

Another installation of a pair of side-pull Duos serves 31 machines and occupies less than 6 in. of headroom, except at one spot between beams, and at the terminals, also between beams. This arrangement allows the release of the whole floor above for other purposes. The capacity of this small machine, $5^1/_2$ in. square, is 25,000 lb per hr. The material handled weighs 60 lb per cu ft.

Fig. 9 shows the moving element of a vertical Redler that handles 400 tons of wheat per hr at a speed of 40 ft per min. An interesting point here is that the outside dimensions of the flights shown are 30 in. × 30 in., whereas the casing through which these flights rise vertically is 36 in. × 36 in. The column of wheat carried up is 3 ft square, and the tiny grains of wheat do not fall backward through the 3-in. clearance on each side.

Another installation feeds coal from the bunkers of an English ship in the Australasian trade, first into the crusher, and then into the pulverizer. The second ship of this line is now being equipped with Redlers.

Fig. 10 shows the test chart of the 5-in. H-type Redler shown in Fig. 8. It represents a summary of a two-months' series of tests made under the direction of Professors Miller, Norton, and Wilkes of the Massachusetts Institute of Technology.

A test was run on this same machine to determine the breakage of friable rice anthracite in a 10-min closed-circuit run. During the $10^1/_2$ complete circuits of the horizontal and vertical H and the 45-deg chute from the outlet of the H to the inlet of the same machine, there was made only $^{50}/_{100}$ of 1 per cent of "fine" which would pass through a $^3/_{16}$-in. rectangular mesh in ten passages by gravity. Of these fines, 6 per cent would pass through a U. S. standard 60-mesh screen.

Because the internal pressure in the Redler conveyor is a function of the spacing of the cross-members rather than of the height lifted, the accuracy of this conveyor as a measuring device is unusually close. This fact has been demonstrated in cement, coking-coal, and cattle-feed blending, and similar service. The No. 1 measuring Redler conveyor measures accurately by the volumetric method from a plurality of hoppers, the measured amounts being carried to a single delivery point on the lower run of the conveyor. The No. 2 measuring Redler has a single inlet and the measuring is done into a plurality of outlets.

An English cement manufacturer was guaranteed accuracy to ½ of 1 per cent with one of these machines. In placing subsequent orders, this cement manufacturer said that his first machine had exceeded its guarantee of accuracy.

The capacities, with various weights of materials, of the different commercial sizes of Redler conveyors are shown on the chart in Fig. 11.

Horsepowers for driving this kind of a conveyor naturally are lower than those required by types of conveyors that disturb the material in transit, and thereby do work upon it. There is almost no internal friction in the material carried by the Redler method. The English and Swiss use empirical formulas which are reasonably satisfactory, but a general and logical one probably will be worked out shortly, based on the area involved, coefficient of friction, and the pressure due to the weight of material between flights, which latter seems to be a controlling factor. The arching action is controlled by this spacing, and the pressure against the sides of a vertical is independent of the total height. This fact means that the action here is similar to that of grain in a silo where the weight of the grain above the arch is transferred completely to the sides.

With material flowing quietly through a closed trough, there is no dust, and, because these conveyors habitually operate with a choke feed and can be pulled through a choked outlet, it is obvious that dust problems may be solved without dust collectors. This condition provides one reason why the Redler is the only elevator sanctioned unsprinkled for grain by the British Board of Underwriters.

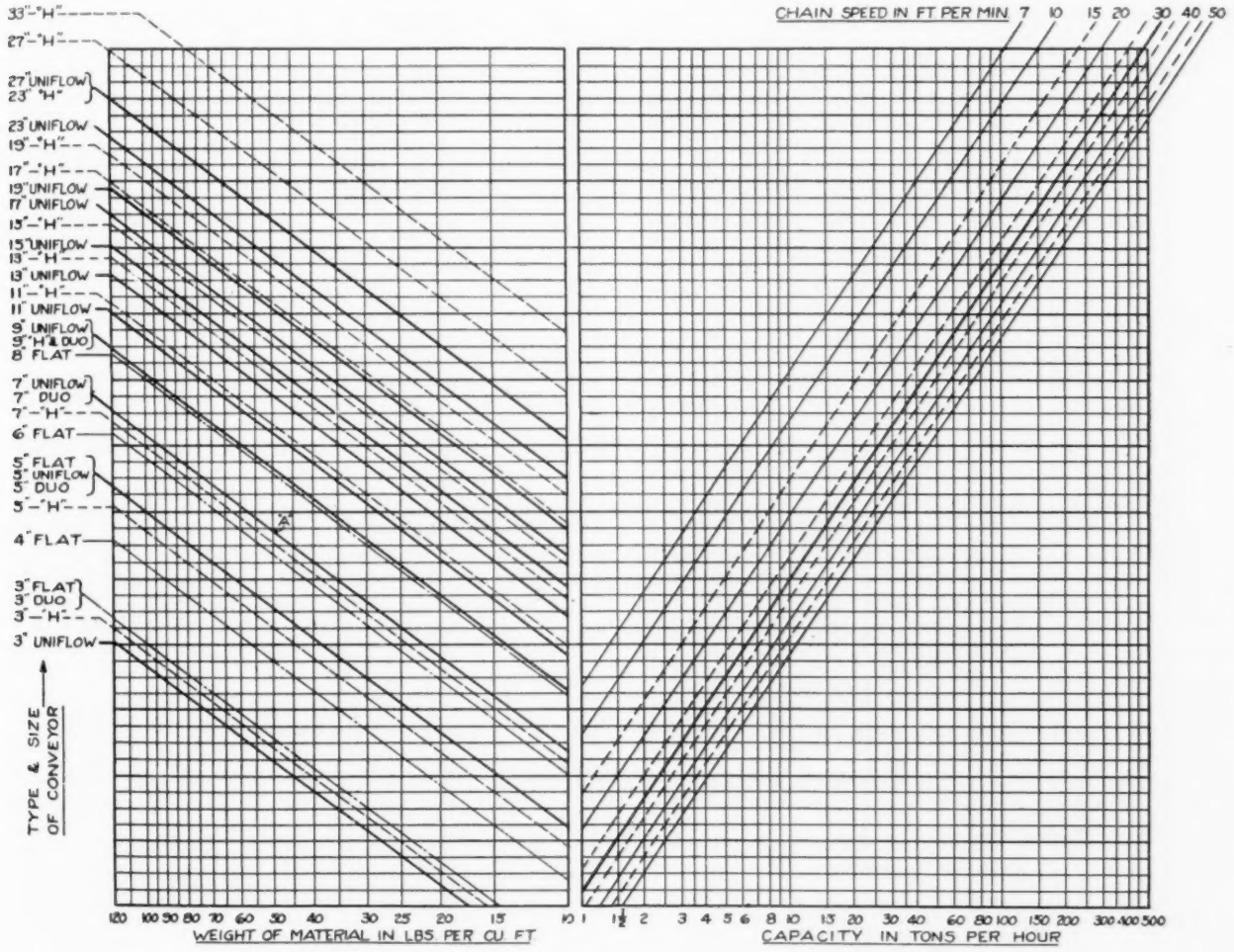


FIG. 11 CHART FOR DETERMINING THE CAPACITIES OF REDLER CONVEYORS

The field of prospective usefulness of this kind of conveyor ranges all the way from a 2-in. pipe conveying coal to, and ashes away from, a small domestic heater, to massive conveyors and elevators handling 600 tons per hr. Although this latter is the largest size built today, there is no indication that the limit of size is being approached. The principle appears to hold good irrespective of size. Other considerations, therefore, can be left to govern the size of the individual units best suited for the particular job.

SAND HANDLING AND POURING UNIT

The Modern Equipment Company has developed a sand-

conditioning unit that is portable, operates on overhead tracks and serves a series of hoppers. The unit includes screens, pug mill, aerator, and magnetic separator, also an automatic adjustable tempering device. It is adaptable to any capacity foundry (capacity per unit being 8 to 12 men) doing production or jobbing work, and operating either on a daily-pour or continuous-pour basis. It allows for the use of different kinds of sand on each floor, with various amounts of moisture if desired, can be used in foundries where many different metal compositions are poured, requires little electric power, and needs no floor pits. Molding increases through the use of this equipment have been more than 50 per cent. For example, on one hard iron ball job, production was increased from 180 to 308 molds; on a heavier job, from 16 to 36 molds, within the same period of time.

The molder makes his molds on his molding machine directly under the sand-hopper outlet, and places them on a gravity conveyor on which they roll down into the pouring zone. There the molds are poured by a pouring device operated on a pouring crane spanning the entire section. Each molder has two conveyors, the second one being used when the molds on the first one are being poured and shaken out.

The farther end of this roller conveyor is raised after pouring, so as to allow the molds to travel back to the shake-out end. The portable conditioning unit is then located directly in back of the hopper and the molds are lifted off the end of the conveyor and dumped directly into the vibrating shakeout. The castings are picked off the grate and put into a bucket operating on a monorail system. In the meantime, the sand drops through the shake-out and on a special shake-out belt which carries it into the bucket elevator. This bucket elevator, in turn, carries the sand to the top of the conditioning machine and discharges it on a belt operating over a magnetic pulley by means of which the metal drippings and small metal-shot particles are removed automatically. The sand then is screened through a fine-mesh vibrating screen and drops into the positive batch weigher.

The batch weigher is part of the adjustable automatic tempering system. It weighs off accurately a definite amount of sand, discharging this into a mixing unit and, at the same time, actuating the water apparatus and causing a discharge of water. The special mixer then thoroughly works and reworks the moistened sand, producing a well-bonded mixture. This sand is then conveyed by a flight elevator up to the aerating device where it is blended. Finally, it is discharged into the hopper by means of a high-speed belt.

Screening Equipment

The Dester Concentrator Company reports an important installation of its equipment in the new plant of the Mason City Brick & Tile Company, Mason City, Iowa. Twelve new Leahy heavy-duty vibrating screens were installed in this plant, in batteries of four. Each battery is in a unit including, in addition, a granulator, grinder, rotary drier, and accessory conveying equipment.

Pneumatic-Conveyor Systems

The Dust Recovering & Conveying Company made three installations during the past year that are of special interest to the material-handling field. The first is a high-vacuum suction-type pneumatic conveyor to unload crude arsenic. This material is poisonous and it is essential to observe special precautions in unloading it. It was decided to remodel several tank cars, load the material in bulk in these tank cars, and unload them by the use of a pneumatic system with a capacity of 4 tons per hr of crude arsenic, weighing from 135 to 150 lb per cu ft. There are four manholes on the top of the tank car, through which the nozzle and flexible hose are introduced, and the operator is not required to enter the tank car at all. Since the complete system is under suction, there is no leakage of material at any point, and the high-efficiency fabric filters remove over 99½ per cent of the dust from the conveying air before this air is exhausted to the atmosphere through the vacuum pump. The intake line is approximately 40 ft long, and a 15-hp motor actuates the filter system.

The second interesting installation is for the conveying of dicalcium phosphate from a loading platform to a mixing tank. This material is the base of a well-known tooth paste, and extreme care must be taken to prevent its contamination by any foreign substances whatever. The material is shipped in bags, and these are emptied into an intake hopper in a special room beside the unloading platform. The pneumatic-conveying pipe lines run from this intake hopper to a receiving station over the material storage bin. Two precautions are taken to insure absolute purity of the product. First, there is an atmospheric filter in the room in which the hopper is located. Secondly, all parts of the intake hopper, pipe line, receiving hopper, and discharge mechanism are of non-ferrous metals. The material, therefore, is conveyed by clean air, and is brought in contact only with rust-proof metals throughout the whole system. A capacity of 5 tons per hr is handled through this system over 75 ft of pipe line, with a 25-hp motor on the vacuum pump.

The third installation is a pneumatic conveyor to unload alum, lime, and soda ash from freight cars to storage bins. The receiving hopper is mounted in a tower over the storage bins so that the material is all conveyed first to the receiving station and is then distributed through gravity chutes to the various bins. The material also can be reclaimed from any storage bin and diverted to process through the same conveying system. There is a maximum length of about 120 ft from the intake nozzle in the car to the receiving station, and over this length of line more than 8 tons of these materials can be handled with a 50-hp motor on the vacuum pump. A 3½ in. pipe line is the only connection between the receiving hopper and the loading platform, and is carried for most of the distance outside the walls of the building.

During the past year the Kennedy Van Saun Manufacturing and Engineering Corporation has developed new equipment for pneumatically transporting material. This equipment can be installed for any capacity, distance, or elevation, is entirely automatic in its operation, has no motors nor feed screws, and is claimed to require no more air per ton at low capacity than at high capacity. It may be installed to deliver from the hopper of a bin or take the feed direct from a pulverizer. It is not sensitive to fineness, and can be used to handle flue dust or coal such as is used in stoker or hand-fired boilers.

Excavating Machinery

For general excavation work, a complete new series of revolving gasoline excavators has been developed by the Bucyrus-Erie Company. These machines, ranging in size from ½ to 2¼ cu yd., are available with gasoline, Diesel, or electric power, and may be equipped to suit various types of work, such as shovel, dragline, crane, clamshell, or drag-shovel operations.

For the contractor who needs a small, light shovel for county and state highway-department work, and for industrial concerns removing sand, gravel, earth, etc., the company has introduced a new ½-yd machine with maximum digging capacity per pound of weight, due to a new distribution of the weight of the equipment. It is available with gasoline, Diesel, or electric drive, and as a shovel, dragline, crane, clamshell, drag shovel, or skimmer scoop.

A line of heavy-duty, full-revolving shovels for ore and rock

digging in mines and quarries has been extended by the addition of three new units designed to bring about lower operating costs. The present line includes six electric shovels from $1^1/_2$-yd to 5-yd capacity each, suited to heavy digging.

For levee, drainage, and irrigation work and for certain other construction work, the company has developed specialized equipment for the speeding up of work and lowering of costs. This equipment includes the following three items: First, an improved type of electrically driven tower excavator has been provided, capable of handling a 10-yd bucket with a span of 800 ft between towers. Second, a new 1- to $2^1/_4$-yd dragline has been introduced, together with two similar machines having larger capacities but with the same low ground-bearing pressure, for work on soft terrain. Third, dragline booms have been introduced that are built of aluminum alloy and weigh only two-thirds as much as steel booms. These booms are applied to draglines up to 175 ft in length and permit the use of longer reaches with standard buckets or larger buckets at standard reaches.

A new series of dragline buckets has been introduced, designed to give maximum strength with minimum weight and intended for heavy, medium, or light work. These buckets are built in sizes from $1^1/_2$ to 8 cu yd. The arch is a one-piece annealed steel casting of great strength but light weight. The manganese-steel lip has renewable teeth inserted in bases cast in the lip and held by wedges. Replaceable runners and wearing plates protect the bottom of the bucket from damage.

A new Diesel-driven (Bucyrus-Monighan) walking dragline handling a 10-cu yd bucket with a 160-ft boom has been introduced. This equipment is especially adapted for such large irrigation and drainage work as is at present being done in the Mississippi River Valley.

For use in the open-pit coal-mining fields the company has introduced a new counterbalanced-hoist stripping shovel. This huge machine carries a dipper of 18-cu yd capacity. By means of the hoist counterbalance, this shovel is enabled to show an increase in operating speed, a decrease in power consumption, and lower overall cost per yard of material moved.

For industrial service, the Loadmaster has been redesigned and improved. This machine is a small, mobile, gasoline crane with a full-revolving boom, which can work in almost any place where an automobile can be driven. It is now offered with three makes of power units. The Model CT machine has 4500 lb lifting capacity, a hoisting speed of 47 ft per min, and a swinging speed of from 2.5 to 4 rpm. Standard booms are 10, 12, and 14 ft long. Special booms can be furnished. Clearances are 9 ft $^1/_2$ in. in height, and 6 ft $^1/_2$ in. in width.

For the railroad market, the company has developed and introduced a railway crane capable of handling a load of 200 tons at a radius of $17^1/_2$ ft. Among the features of this crane are: lower center of gravity, permitting high-speed transportation; strong construction; an unusual degree of accessibility; and wide clearance for lifting bulky objects.

Industrial Power-Driven Trucks and Tractors[5]

The Clark Tructractor Company has announced a new lifting and tiering truck, Fig. 12, that will pick up skidded or cleated loads with as low as 2 in. underclearance. It differs from the ordinary lift truck in that the load is carried on tapered steel fingers, with a standard length of 26 in., that touch the floor in the down position. Fingers tilt back slightly in the first 5 in. of rise, thus insuring balance for the load. Flexibility of operation is obtained by four-wheel steering with rear-wheel drive. The 2-ton model has a turning radius of 89 in., the 3-ton model, 98 in. Maximum lifting and carrying capacity are provided for by the powerful hydraulic lift, mounted over extra large (15 in. × 7 in.) rubber tires.

While the machine is capable of tiering to a height of 50 in., its low overall height, 77 in., makes it specially suitable for loading box cars. Special machines with greater tiering height to meet unusual conditions are built by the manufacturer on this same chassis.

The Elwell-Parker Electric Company has developed a complete line of tiering trucks fitted with true telescoping uprights on which the load carriage travels. Telescoping uprights are those which provide for a set of movable inside uprights sliding *within* a pair of fixed outside uprights. These trucks are intended to conserve floor space by utilizing all the cubical contents of storage areas. They are designed to pass low doorways, piping, and other overhead obstructions, and yet stack stored objects to the roof.

Tiering platform trucks of this general type are made for from 500 lb capacity with an overall width of truck of 30 in.

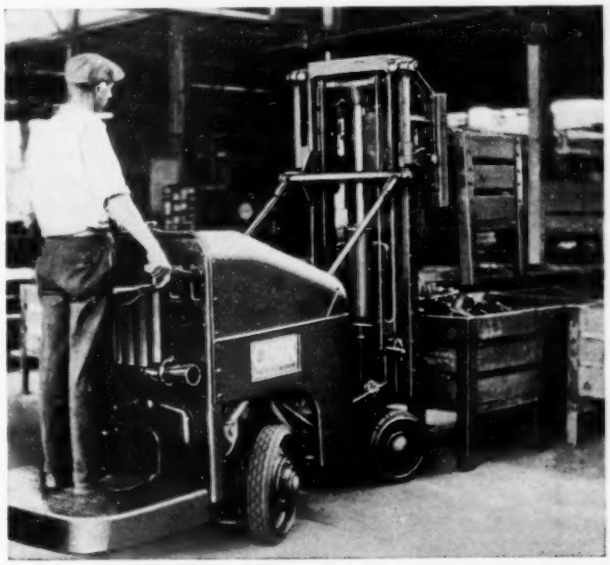


Fig. 12 Tiering and Lifting Truck for Cleated Loads

and a lift of 184 in. to a 10,000-lb capacity and a lift of 106 in. The intermediate sizes built have been the 4000-lb and 6000-lb. sizes (see Fig. 13). In fork-type trucks, those developed have been rated at 1000 lb, 3000 lb, and 6000 lb, with varying outreaches to the load center of gravity.

In addition to these machines, two special roll-on winch-type trucks have been developed to handle heavy cable reels; a special tilting-fork-type truck to handle cooker pallets of canned goods in food factories; and a special type of tilting-fork truck to carry bulky automobile bodies through a 7-ft high doorway, after passing which the body is stored on end.

There is also available a heavy-duty 10,000-lb combination fork-and-ram-type truck with motor-operated side-shifting device for handling mill plate and rod coils, and a combination crane-and-fork truck. (See Fig. 14.)

The development of accessories on electric trucks has advanced apace. Rubber pads have been placed on operator's platforms to reduce the vibration and add comfort to the operator. Wheel steer instead of either the vertical- or horizontal-type steer has been in greater demand on the lesser-capacity trucks. Ball-and-socket-type steering connections, pressure lubricated, have been made standard equipment, replacing the older pin-and-yoke, gravity-oiled steering connections.

[5] Data collected by S. W. Gibbs, assistant sales manager, The Yale & Towne Manufacturing Company, Philadelphia.

The higher types of battery, both alkaline and lead acid, have found greater favor and, with the gas-electric power unit, have required a redesign of most power unit compartments. In the redesign, greater attention has been necessary to provide more ventilation for the batteries, since the cells that are placed in the center of the high type demand better cooling facilities.

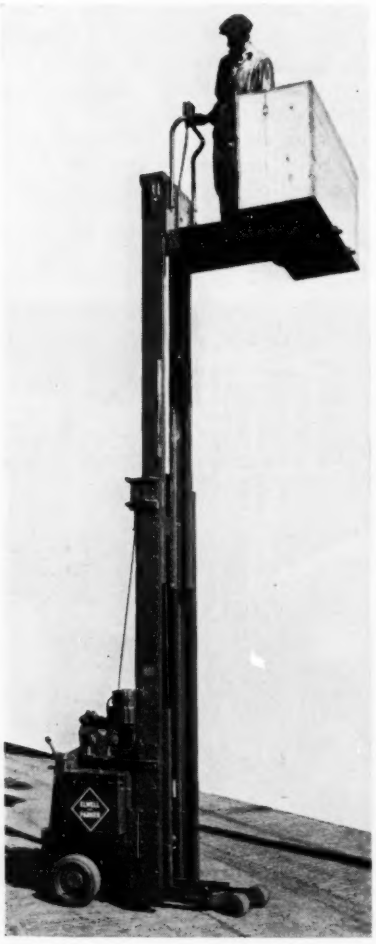


FIG. 13 TIERING PLATFORM TRUCK WITH TELESCOPING UPRIGHTS

Wider uses for the various kinds of equipment have been noticeable during the past year, since there has been insufficient work for the trucks to perform on the jobs for which they were purchased. Tiering trucks have been fitted with removable booms to serve as cranes, while slip-over platforms have been employed to permit landing skid loads on industrial trailers of the older types which do not provide the clearance necessary to permit the trail axle beneath the truck platform to be thrust under the load platform of the trailer.

The Mercury Manufacturing Company has added several designs of new units to its electric-lift-truck line. One of these new units is designated the "Little Giant" and is a small-sized, but powerful, truck of the standard "Type EL" line. The unusually light chassis weights and capacity ratings of the trucks of this classification enable them to operate where floor or elevator capacities are limited, and to negotiate narrow aisles and congested areas.

There are three modifications of the basic design—the low-lift, the high-lift, and the telescopic. The low-lift model has an overall length of $94^1/_8$ in., an overall width of 32 in., and an overall height of $46^5/_8$ in. It has a rated capacity of 3200 lb. The chassis, less battery, weighs 1850 lb.

The overall height of the high-lift model is 83 in., and the truck is capable of lifting the platform to a maximum height of $67^3/_4$ in. Trucks of the high-lift type have the same overall length and width dimensions and the same carrying capacity as those of the low-lift design. The chassis, without battery, weighs 2100 lb.

The telescopic model has an overall length of $96^1/_2$ in. and an overall width of 32 in. The overall height of the truck in its lowest position is 83 in. When raised to its ultimate height, the top of the platform is 112 in. above the ground. It has a lifting capacity of 3000 lb on the standard uprights and a capacity of 2000 lb on the telescopic members. The chassis, less battery, weighs 2400 lb.

Another development is a special adaptation of the standard "Type EL" fork truck to handle paper rolls of varying diameters and lengths and stack them three high without the employment of skids or supports of any kind. (See Fig. 15.) Mechanically, this unit is described briefly as follows: It has an overall length of 166 in., an overall height of 82 in., and an overall width of 46 in. It has a 50-in. wheel base and the turning radius is 138 in. The lifting forks are adjustable to handle rolls of paper varying in diameter from a minimum of 27 in. to a maximum of 40 in., with a maximum length of 6 ft. The truck lifts paper rolls weighing up to 4200 lb and tiers them to a height of 96 in. The truck, without battery, weighs 6000 lb.

Both of these new units have hydraulic-hoist systems, contactor-type controls, internal-expanding brakes in each drive wheel, semi-elliptic spring suspension of the frame, and efficient unit-gear-drive axle assembly, ball-bearing-mounted throughout.

The Terminal Engineering Company reports as probably its most outstanding development during the past year a crane with a boom that will give a lift of 26 ft from the ground to the hook. The chassis is a standard 60-in.-wheelbase crane chassis having two hoist units, one for the hook and one for topping the boom. The crane with normal-length boom has a capacity of 6100 lb two feet from the end of the chassis, and the long boom has a capacity of 1800 lb at the same overhung distance. The boom, however, is movable and also telescopic over a distance of 6 ft, so that different capacities are available with the different combinations of adjustment.

The long-boom crane was developed for working on one-story-building construction jobs, and has been used in particular for the erection of prison cells. One-story construction makes it impractical either to set up scaffolding for the steel-erection work or to put in overhead hoists on trolleys. The material for the high crane is supplied to it by electric lift trucks with separable large-area platform bodies. Last winter and spring



FIG. 14 COMBINATION CRANE-AND-FORK TYPE TRUCK

these four-wheel drive lift trucks operated over unpaved areas on building jobs without the use of chains on the wheels.

The high crane on prison-construction jobs was also used to install the ceiling plates, to raise all of the furnishings, and to hoist the concrete for the cell floors, instead of using a conveyor which would be at a disadvantage on account of the limited space available.

Additional lift trucks were used while the bricklaying was being done. Each of the seven cell blocks, three cells in height, was 320 ft long and 35 ft high, so that scaffolding would have

been expensive to erect, either around the whole area or in sections to be moved. Instead of building up and tearing down the scaffolding every four days, the contractor built a group of forty individual towers on four legs, complete with the necessary ledgers. These towers were counterbalanced with two tons of brick in the bottom and were moved from place to place by the electric lift trucks. Each tower was a complete scaffolding in itself, and a group of them, set at proper intervals with decks, provided a continuous scaffolding the length of the building.

These forty separate tower units on the outside, used with the electric trucks, replaced about 25,000 linear feet of scaffolding 30 ft high and the total saving on the bricklaying operation, which included supplying the masons with bricks by the same lift trucks, approximated $40,000. The scaffolding could be moved from one 320-ft section of wall to another in less than two hours' time.

The Yale & Towne Manufacturing Company has introduced as new models this year a line of rack-and-pinion tilting-fork trucks, in both the single and multi-lift, telescopic type. Another new development is a complete line of midget low-capacity trucks with models corresponding to the existing models in the higher capacity machines. These low-capacity midget trucks are rated up to 4000 lb maximum, and embody the advantages of small overall dimensions, lightness in weight, short turning radius, etc., necessary in this class of equipment.

Under the head of mechanical improvements during the past year, the company has perfected and now embodies in its equipment several features. One is a free-wheeling ratchet used on all high-lift trucks to guard against damage to the equipment when overrunning the travel or hitting an object when lowering the platform. Another is an interlocking four-speed control that prevents the application of power against the brake and necessitates the return of the control manually to neutral before reversing the direction of the truck. Wheel and bearing design has been improved with a view to permitting wheel dismantling or replacing without exposing or disturbing the bearing.

The chassis construction has been changed considerably, and wide use is made of form plates and alloy or heat-treated bar stock for the main frame, in place of the castings and shapes heretofore used. This change has been made to simplify the design, increase the strength, and provide a better distribution of load.

Hand Lift Trucks[6]

The improvements introduced by the Yale & Towne Manufacturing Company during the past year in hand lift-truck equipment have been in the development of added types rather than in any decided mechanical improvements. Included in this list of added types has been a U-frame hand-type single-stroke model with a pan for handling small reels. When lowered, the pan rests on the floor, the reels are rolled on to the pan and then the entire unit is then elevated. The second development has been in a hogshead or barrel truck, U-shaped, with the two sides of the U designed to fit the contour of the barrel. A third development has been in an extremely low platform truck for use in handling tinplate and similar commodities when these are shipped on low skid legs.

In the skid-platform field, which is accessory to lift trucks, a number of mechanical improvements have been made in the use of better structural sections, including channels and special shapes to provide more strength, and the use of both welding and stronger bolting to improve these members.

[6] Data supplied by S. W. Gibbs, Yale & Towne Manufacturing Co.

Floor Trucks[7]

The truck-caster manufacturers are cooperating with the Standardization Division of the Department of Commerce, through R. L. Lockwood, on a standardization program for plate casters. A preliminary meeting has been held and a second one is scheduled for an early date.

An outstanding improvement in the floor-truck field has been the adoption, by several makers, of rubber-tired wheels for casters and floor trucks. Improvements in design and construction of the tires and wheels have been very noticeable and a number of manufacturers have adopted the new easy-rolling rubber in the wheel tread. Wheels of this rubber require much less power to move than those made of any of the kinds of rubber formerly used. Under most floor conditions, except for the smoothest marble and new concrete, the rubber tire will run easier than a metal wheel of the same diameter under the same load.

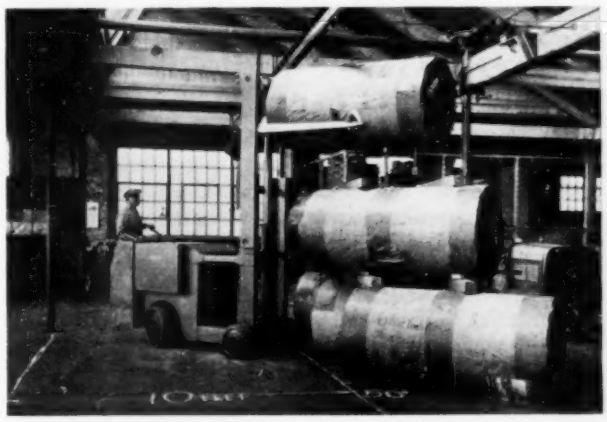


Fig. 15 Fork Truck for Handling and Stacking Paper Rolls

There also has been a more decided trend toward the use of roller bearings in floor-truck and caster wheels, particularly for the lighter and medium-duty equipment which generally has used plain bearings. Roller bearings usually have been standard units in heavier equipment.

Concerted action has been undertaken by the floor-truck and caster group to prepare and provide publicity on new equipment, new and outstanding installations, and improved handling methods. Most members of the industry feel that the importance of its equipment has not been brought to the fore as it should be, and that consequently the false impression is being created that this equipment is being supplanted by mechanical methods. As a matter of fact, floor trucks have been extensively employed in factories during the recent period of low production to carry on handling operations where mechanical equipment was shut down because of the low volume of output.

The Nutting Truck Company has announced a new all-steel platform truck, with frame and platform pressed from a single piece of heavy steel so that the edges form a channel shape. A steel angle welded into the frame forms the caster bar. The truck has the regular U-shaped handle, which is removable or can be fastened in place by set screws. These trucks are furnished in several sizes of standard platforms with a choice of two kinds of wheels.

Tires and Casters for Trucks[7]

The Nutting Truck Company has introduced three new

[7] Information furnished by J. F. Thomas, general sales manager, Nutting Truck Co.

types of tires made of "free-wheeling" rubber. The N type, for light and regular duty, has a 14-strand wire cable reinforcing the base, and is securely held between steel wheel disks which prevent it from jumping off or creeping. The RD type, for regular and heavy duty, is a demountable, wide-tread tire clamped between the halves of a split wheel. It has a hard-rubber base impregnated with cord fabric. The tire is made in sizes to fit any truck or caster. The RV type, for heavy and extra-heavy duty, is vulcanized to the wheel, and has an extra wide tread. Crosswise corrugations on the metal prevent loosening of the tire and consequent creeping. Advantages claimed for the "free-wheeling" rubber are: noiselessness, 60 per cent easier starting, 33 per cent less rolling effort, cushioning of loads to prevent breakage, elimination of creeping or jumping off, durability, low depreciation, and reduction of wear and tear on floors.

Divine Brothers Company during the year put on the market a new TonWate steel forged caster in both stationary and swivel types. The base of the caster attaches to the truck by bolts or screws on standard centers. The majority of the load in the swivel type is carried on a single ball which, acting as a flexible unit, distributes the remainder of the load to the vertically split, ball thrust bearing. A Dardelet self-locking safety set screw provides for adjustment in installation. Heat-treated forgings provide great strength without excess weight. Hyatt axle bearings, pressure lubricators on the oversize axles and swiveling mechanism, heavy-gage-steel hub guards for protection and cleanliness, and wide-faced, machined wheels are other features. Canvas-cushion wheels are provided where floor protection and quiet operation are desired.

Tests made for the Port of New York Authority, on June 20 and 21, and conducted at the New Haven Railway Express terminal of the Railway Express Agency in New York City, indicated the superiority of rubber tires over steel tires for use on trailers, and incidentally on industrial power-driven trucks and tractors and on floor trucks. The tests were conducted to demonstrate tires of the vulcanized-on type made of a new compound and produced by the B. F. Goodrich Company. Ball-bearing, swivel casters were mounted on the front of the truck and Hyatt roller-bearing wheels on the rear. Starting and rolling drawbar pulls on smooth and on rough floors, with casters parallel to and at right angles to the directior of pull, were run, pulling the trailer with different loads over different-sized obstructions. The pull to move the trailer over a length of hay wire was only one-sixth as much with rubber-tired wheels as with steel wheels. Tractors pulling trailers showed slightly less power consumption from batteries when the trailers were equipped with rubber tires. Increased resiliency was demonstrated by the new tires, for which the usual advantages of rubber tires, namely, reduced noise, wear, maintenance and breakage of objects carried, have been claimed.

Adherence to Tire Standards

The Division of Simplified Practice of the Bureau of Standards, in cooperation with makers and users of industrial truck tires, has issued its 1931 revision of the simplified-practice recommendation on widths of tires and nominal wheel diameters for 30 flat-base tires and 6 channel-base tires. Five tire manufacturers, in 1931, produced 48,459 tires, of which 77.3 per cent were of the recommended sizes. Nine truck manufacturers used 12,957 flat-base tires, of which 73 per cent were of the recommended sizes. Eventually, it is hoped to eliminate all the channel-base tires and 9 sizes of flat-base tires. Most non-standard sizes made now probably are required only for replacement purposes, except some oversize tires needed for trucks of higher capacities. Certain of these latter tires may be placed on the standard list.

Industrial Cars

The Koppel Industrial Car and Equipment Company has developed a larger size, automatic, air-operated, side-dump car of 50 cu yd capacity, along the lines of the 30-cu yd car that has been on the market for several years. This new car was specially designed for use in handling various kinds of metal, particularly waste materials in steel plants. Lower operating and maintenance costs, and lower initial investment, on the basis of cubical capacity, are claimed for the 50-cu yd car. The car will carry 140,000 lb, is 8 ft $7^5/_{16}$ in. in overall height, 10 ft $4^1/_2$ in. in overall width, and 46 ft in length overall over striking plates. It is 42 ft $2^1/_2$ in. long inside the body at the top, 9 ft 3 in. wide inside at the top, and 3 ft 7 in. deep. It weighs 80,300 lb and has a dumping angle of 45 deg. Dumping can be controlled from the locomotive or any car, safety devices protect against faulty manipulation, and the cars are universally accepted for movement in interchange service because they meet all A.R.A. and I.C.C. requirements.

Motor Trucks and Trailers

The General Motors Truck Company, to supplement its Model T-18 motor truck, which has a capacity of from $1^1/_2$ to 2 tons, has introduced a semi-trailer, Model TT-218, with a body capacity of from 600 to 800 cu ft, to carry from 3 to 5 tons. The working parts of the trailer are standard with the truck itself, for low-cost maintenance. A 42-in. stake-and-rack body, and a stake express body with a 2-ft reinforced tail gate are available. The supporting wheels at the front are raised or lowered by a steel cable and lever mechanism operated by a crank at the side. A dog locks the mechanism, for safety.

Tank Cars for Bulk Material

The General American Tank Car Corporation has introduced a standard railroad tank car for the handling of bulk materials in granular or powdered form. It is applicable for cement, clay, lime, sand, fertilizers, soda ash, sulphur, flour, and other products that require protection from the weather and from dust and contamination. Weights up to 70 tons of material can be transported.

The car is filled from hoppers or pipe lines through any of six 16-in.-diameter loading manways in the top, closed by dust-proof and water-proof covers. Loading time may be as little as 30 min. The car is unloaded through a hopper opening, $11^5/_8 \times 12$ in. in area, at the middle of the bottom. Unloading is done by two drag-chain conveyor systems within the car, one for each end. Slope sheets at the bottom of the car divert the material into the conveyor line, which runs in this trough along the bottom toward the central discharge hopper, then travels up and back toward the end of the car, above the material. The chain flights are 14 in. wide and travel at 10 ft per min, driven by a 1200-rpm motor through two high-ratio speed reducers, one for each conveyor. Clutch connections enable the operator to start or stop either conveyor independently of the other. A discharge rate of either 400 or 800 cu ft per hr is thus available, and the car is emptied completely. The total capacity is 1600 cu ft, and the car can be unloaded in from two to four hours, discharging to storage hoppers or onto a conveyor system passing beneath the track. Unloading is dustless. Power is obtained by cable from any convenient 220- or 440-volt line, or may be applied externally by means of a shaft provided for this purpose. Two bulkheads under the central dome house the driving mechanism and protect it from dust.

Savings indicated by tests are as follows: elimination of barreling, loading, unloading, and spillage in handling arsenic, $200 per 65-ton shipment; purchasing of quicklime in place of hydrated lime, where the former was too dangerous to handle

under previous methods, $1 per ton; elimination of packaging and the incident handling, $2 per ton; substituting some materials in bulk for less efficient ones in packages, as high as $1000 per month per car.

Handling on the Railroads

New methods of handling on railroads deserve special mention. A method for hauling freight containers in and out of the ends of covered cars has been developed by the Milwaukee Electric Railway and Light Company with the assistance of the Harnischfeger Corporation. Aluminum-alloy containers are used, mounted on skids moving on steel slides on the car floor. A Milwaukee electric hoist is mounted beneath the car, but the cable can be unreeved from the drum without power, because the hoist has a slip clutch for this purpose. The cable passes up through the car floor at the end, between two horizontal rollers, runs through a snatch block anchored there, and is pulled to the other end of the car where it is fastened to the container, which has arrived on a motor truck. Operation of the hoist moves the container into the car, which holds two containers. Power is obtained by hanging a pole over the overhead trolley wire, and the hoist is operated by any one of four push buttons at various points in the car. Unloading is done in a similar manner, because the motor truck has a snatch block back of the cab.

The National Malleable Steel Castings Company this year has adopted special containers, leased from the L. C. L. Corporation by the railroad, for receiving limestone, coke, dolomite, and similar material at its plant. The containers are loaded by grab bucket and crane at the shippers' yards, or by chutes from storage bins. They are carried on beams in gondola cars that hold 12 containers, or 60 tons. The car floors consist of screens, for drainage purposes. Open containers are used for stone, and closed containers for materials that must be protected from the weather. Eventually, it is hoped to carry clay and brick by this method. At the National Malleable plant, a crawler-tread crane is used to transport the containers from car to storage bin or other point of delivery.

An economy of about $12.50 per car, as well as a saving in time has been brought about by the new method. Under previous methods, it took six men from five to six hours to unload a car that can be emptied in 50 min under the new method. Demurrage on cars is obviated and, in addition, the cost of materials thus bought in bulk averages $1 per ton less than when bought in bags or barrels.

The Pennsylvania Railroad is experimenting with containers designed to transport face brick, hollow tile, firebrick, and similar products of the ceramics industry. Two types of face brick container have been tried out, a bottomless variety and a sectional variety. With the former, the bricks are stacked on skids which, when loaded, are stacked vertically (at present four high) so that a container can be lowered over the stack and fastened to the bottom skid by means of two beams swiveled from the sides of the container. The container is then hoisted by a crane into an open-top car.

The sectional container consists of a series of boxes or rectangular shells, each 16-in. high, to fit over steel skids or platforms mounted on legs with a clearance of 7 in. to permit the use of lift trucks. The box section in each case latches on the platform and the sections, when loaded, are stacked, usually four high, although any number up to five high is permissible. Cranes may be used to load the containers on cars, with lift trucks to do the intermediate handling, although hand loading is possible, due to the light weight of each section.

The data on the two kinds of containers experimented with are as follows:

Item	Bottomless container Small	Large	Sectional container
Length of base	4 ft 3 in.	7 ft 0 in.	6 ft 4 in.
Width of base	3 ft 6 in.	3 ft 6 in.	4 ft 3 in.
Height of container	7 ft 6 in.	7 ft 6 in.	6 ft 2 in.[1]
Skids per container	4	4	4
Bricks per skid	300	500[2]	600
Bricks per container	1,200	2,000	2,400
Containers per car	24	14	14
Bricks per car	28,800	28,000	33,600
Weight of container and contents, lb	2,000	3,350	.....

[1] This figure is approximate. Sections are 16 in. high, and four are used, which, with allowances for skid platform thicknesses and clearance under the legs of bottom skid, give a total height of about the amount stated.

[2] Figures in this column, from here down, have been estimated by calculation.

Coordinated Railway and Truck Service

The Chicago, North Shore & Milwaukee Railroad, as a new branch of its "ferry truck" service, is offering another facility to shippers in Chicago, Racine, and Milwaukee. Entire motor trucks and trailers, filled with merchandise, are now moved on specially constructed flat cars. The charge is 10 cents per 100 lb of merchandise, with a minimum load of 15,000 lb, for trucks and trailers up to a total of 23 ft in length, and the same rate for longer trucks but with a minimum load of 18,000 lb. Ramps at receiving and delivery points allow the trucks to be run on the freight cars and fastened. Loading and unloading are done by shippers or consignees, or their agents. The railroad, which has a main line 90 miles long, thus is counteracting competition from motor trucks on short hauls.

Seatrain Freight Service

Seatrain freight service was extended during 1932. The original service, with one vessel, has been in effect between New Orleans and Havana for three and one-half years, connecting the 2600 miles of Cuban railways with those of the United States. A car-ferry vessel, 427 ft 6 in. long, with a 62-ft beam, has provided the transportation. The vessel has three decks, all served by one large hatch amidships. Each deck has four rows of standard-gage track, and the superstructure deck also has tracks, giving a total trackage of about one mile, sufficient for 100 cars. A 110-ton crane picks up a cradle on which standard railroad freight cars are run, hoists the car and lowers it to the desired deck, where it is switched into position by a steam-driven car-hauling gear. The crane then picks up a car from the ship and hoists it to the dock on the return travel. From 15 to 20 cars per hr thus are loaded and unloaded. The cars, during the voyage, are secured by chocks and jacks. Two new vessels have been completed to start service between Hoboken, N. J., and Havana, late in 1932. On its trial run, one boat made a speed of 17.1 knots at 8800 hp, or $^3/_8$ knot faster than the designed speed, and thus will be probably the fastest freighter afloat. The New Orleans-Havana service to Cuba handles gondola cars of coke, tank cars of lard, cottonseed oil, gasoline, kerosene, lubricating oils, acids, etc., refrigerator cars of fruit and vegetables, box cars of unbagged grain, flat cars of lumber, etc. On return trips from Cuba, the gondola cars bring manganese ore, rock asphalt, scrap iron, and similar bulk materials for which economical transportation mediums formerly were lacking. Refrigerator cars bring back perishable commodities.

The advantages of the service are that materials are loaded in cars at the shippers' plants, moved over the railroad to the water front, hoisted on the Seatrain boat, hoisted off at the port of call, and moved over another railroad to the plant or warehouse of the consignee. In one year, the New Orleans service increased by $1,000,000 the sales of five Southern products in

Cuba. The one vessel, in three years, furnished over 40,000,000 ton-miles of transportation without incurring any damage claims.

Material-Handling Papers and Publications

A reduced amount of literature was produced during the year, due to the restricted number of technical meetings held and the concentration of attention throughout industry on problems of securing business rather than widespread improvements in plant equipment and operation. The quality of published items, however, was up to a good standard.

The Materials Handling Division, at its Annual Meeting in December, 1931, presented three papers in addition to its Progress Report: "Rustless Steel Conveyors—Their Availability, Application, and Use," by Nixon W. Elmer; "Improved Material-Handling Methods in a Container Plant," by H. N. Macdonald, and "Applications and Design of the Floating Drive for Conveyors," by N. H. Preble.

At the National Meeting in Detroit, held on May 4 and 5, 1932, in conjunction with the American Foundrymen's Association and the Materials Handling Institute, six papers were given: Three papers under the auspices of the Materials Handling Division were: "Maintenance of Materials Handling Foundry Equipment," by T. A. Bissell, associate editor, *Maintenance Engineering;* "Relative Wear of Metals by Abrasion," by C. R. Weiss, chief engineer, Link-Belt Co., Indianapolis, Ind.; "Management Policies and Practices in a Continuous Production Gray Iron Foundry," by W. G. Reichert, metallurgical engineer, Singer Manufacturing Co., Elizabethport, N. J. Three papers under the auspices of the American Foundrymen's Association were: "Tram-Rail for Materials Handling in the Foundry;" "Materials Handling Problems in the Foundry Department of the Western Electric Company;" "Materials Handling as Related to Miscellaneous Jobbing Foundries."

The Engineering Index lists and summarizes in *Mechanical Engineering* 85 articles on materials handling, from August, 1931, to June, 1932, inclusive, distributed as follows: cableways 2; chains 3; coal and ash handling 4; containers for shipment 1; conveyors 22; cranes 11; elevators 7; hoists 1; hooks 1; industrial trucks 2; materials handling 18; motor trucks 8; reducing gears 1; steam-power-plant handling methods 3; tractors 1.

One book of note, intended both for technical reference and college text use, appeared in the field: "Modern Materials Handling," by Simian J. Koshkin, assistant professor of machine design, Cornell University (John Wiley & Sons, Inc., 1932, $6). This is the first book of its kind to be published. Professor Koshkin has undertaken the preparation of another text on the subject of "Characteristic Modern Materials Handling Equipment" (types, applications, and design features), and has the active support of members of the Materials Handling Division in the securing of data for this volume.

The British Standards Institution, in April, 1932, brought out a revised edition of its 32-page pamphlet on "British Standard Specifications for Derrick Cranes."

Manufacturers' Committee of the Materials Handling Division

The Manufacturers Committee of the Materials Handling Division has been carrying on a number of activities, although efforts have been limited by the let-down in business during the current year. At the National Meeting of the Division in Detroit, in May, 1932, the committee held a session at which a number of reports were presented. The report on "Performance Standards of Locomotive and Crawler Type Cranes and Shovels" was submitted in final form by a committee consisting of F. A. Smythe, President, Thew Shovel Company, A. J. Fries, District Superintendent of Motive Power, New York Central Lines, Buffalo, New York, William Ogden, formerly of the Association of General Contractors, Washington, D. C., and J. C. Wheat, Industrial Brownhoist Corporation, Cleveland, Ohio. After approval by this committee, the Shovel and Crane Manufacturers' Association, and equipment manufacturers and interested users, it will be coordinated with pending safety codes of the Society. Copies are available on request. The report was accepted by the Manufacturers' meeting.

A committee on "Formulas for Figuring the Capacities of Containers" presented a summary of its survey of the field, which was accepted at the meeting and is being prepared for circulation to obtain comments and suggestions from manufacturers and users. The report was compiled by Chairman W. E. Farrell, president, Easton Car & Construction Co., assisted by H. E. Brendlin, of the Hayward Bucket Co., and M. J. Andrada, chief engineer, The Robins Conveying Belt Co.

The committee on "Nomenclature and Classification of Materials Handling Equipment," headed by Chairman George L. Morehead, vice-president, Link-Belt Co., and consisting of the following other members: L. J. Kline, general manager, Mercury Manufacturing Co., and S. Buckley, Shepard Niles Crane and Hoist Co., reported that its activities would have to be postponed until such time as funds could be raised to compile data for a manuscript to be published in book form on this subject. The work is too extensive to carry further than the planning stage at the present time.

A committee on publicity and the securing of material-handling data for publication in magazines, of which C. B. Crockett, of Crockett & Smith, Eastern Representatives of the Automatic Transportation Co., is chairman, presented a report prepared by M. W. Potts, Secretary of the Materials Handling Division, and covering the results of a survey of engineering colleges to see whether the mechanical-engineering departments or the local Students Branches would be willing to hold material-handling meetings, and desired speakers to be suggested for such meetings. Favorable replies were received in response to a majority of the 72 letters sent out. The Publicity Committee of the newly formed Materials Handling Institute is collecting lantern slides and motion picture films for such purposes and is cooperating in plans for such meetings.

A paper by J. C. Enders, Engineer, General Electric Co., on "The Need for Standardization in Material Handling Equipment," pointed out the expense and inconvenience to which users were put by the lack of standards for items such as conveyor chain, conveyor trolleys, gravity rollers, height of angle-iron sides on conveyors, and similar parts. M. W. Potts, Secretary of the Division, and engineer, Alvey-Ferguson Co., was appointed to set up a committee to study the situation and work out suggested standards for recommendation to manufacturers and users of equipment.

In December, 1931, the Material Handling Institute was formed as a trade and development association of manufacturers of material-handling equipment, to provide for activities that lay without the proper field of operation of other existing agencies serving the industry. A plan of cooperation has been established between this newly formed organization and the Materials Handling Division of the A.S.M.E., the first active step in the program being the joining of the two groups in meetings at Detroit in May where the convention of the Materials Handling Institute and the National Meeting of the Materials Handling Division were staged simultaneously.

MH-55-2

Rustless-Steel Conveyors—Their Availability, Application, and Use

By NIXON W. ELMER,[1] OAK PARK, ILL.

The paper covers corrosion-resistant metals for material-handling use, giving tables of relative corrosion resistance and costs under reasonably average conditions, particularly where food products are concerned. The various available materials are briefly described, and their good and bad features and qualities are developed. The types of conveyors for which these materials have been used are given, and the important and characteristic installation features of each are discussed.



FOR the purposes of this paper the word "rustless" will be interpreted in its broadest and non-technical sense as corrosion-resistant. From the point of view of the operator using the metal in his daily work it makes little difference whether the staining, corroding, or pitting is due to the formation of a sulphide of iron or an oxide of iron; what he is interested in is the deterioration of the equipment itself.

From an engineering standpoint it is almost as bad an economic error to make equipment better than is needed as to make it less good than is required. Therefore the first problem in design is to decide what degree of corrosion resistance is desirable or essential. Knowing the corroding agents and the working conditions, it is the duty of the engineer to find the cheapest suitable material to give the desired length of life.

Corrosion is undesirable, and in designing and making conveyors it is avoided, for two reasons, either of which may control: (1) Physical deterioration of the equipment itself. (2) Discoloration of and injury to the product conveyed. An example of the latter would be yellow rust stains on food products, making them unsalable.

All metal corrodes; some slowly under few reagents, others more rapidly with many reagents. The equipment to be discussed comes under the first class, the problem being to select the most suitable material (where the correct economic life is the object) to resist the corroding agent encountered in the given circumstances. If, for instance, the conveyor is to be used only two years, it would be poor engineering to make it good enough to last 20 years or to build it so cheaply that it wore out in less than the two years.

To start, let the materials generally available for conveyor use be arranged in the approximate order of their corrodibility, as follows: cast iron, malleable iron, hot-dipped steel, hot-dipped firebox steel, galvanized Armco or Toncan metal, Monel metal, 18-chrome–8-nickel steel, pure nickel, or block tin.

In the order of their cost, which is vital to the economic solution of the problem, these materials are as follows:

Material	Cost per 100 lb	Ratio
Malleable iron	$5.25	1.00
Cast iron	4.25	0.81
Hot-dipped galvanized steel	5.25	1.00
Hot-dipped galvanized firebox steel	5.75	1.10
Electro-galvanized Toncan and Armco	5.50	1.05
Hot-dipped tinned steel	9.25	1.76
Block tin	40.00	7.63
Monel metal, polished	53.00	10.10
18-chrome–8-nickel, polished	55.00	10.45
Nickel, polished	63.00	12.00

The engineer must so select and combine the materials that the best result will be obtained for the least money over the proper period of time.

In the comparative-cost table, which naturally changes with a changing metal market, malleable iron has been taken as a basis of comparison.

Cast iron of course lacks tensile strength, will not withstand shock, and rusts sufficiently to stain.

Malleable iron rusts enough to stain, but is stronger and not brittle.

Hot-dipped galvanized steel does not rust until the galvanizing has worn off or corroded off; thereafter it acts like any other plain steel.

Electro-galvanized Toncan or Armco has the slow rusting base of a pure wrought iron or copper-bearing steel, with a thin zinc coating which is not amalgamated with the iron at the surface by heat. The spread of rust from the spots exposed by wear is very much slower here than in the preceding case, although the exposure is quicker to occur. Inasmuch as this material is generally obtainable from stock only in sheets, no mention is made of it in hot-dipped form. Hot-dipped galvanized steel after fabrication is not practical for sheet-metal work.

Block tin is attractive in almost every way except physically. Perhaps its weakness and softness could be modified by the addition of minute quantities of metallic impurities.

Monel metal is expensive, but is easily worked, and where kept in use is itself stainless.

The 18-chrome–8-nickel, when properly made, under either the German or the English patents, and polished, is truly stainless for general uses, both as to itself and the products in contact with it. Nothing lower on the scale of chrome and nickel content will uniformly withstand fatty acids and fruit acids, and there is no need for going higher in the chrome-nickel content unless high-temperature oxidation or physical strength at elevated temperatures is involved.

Common salt is so widely distributed and aluminum is so readily dissolved in its presence that aluminum has not been mentioned in the table. However, the Aluminum Company's silicon alloy and the Major Metal Company's product show hopeful signs of progress in overcoming this defect.

Cadmium in the form of plate cannot be used either safely or legally in contact with food products, it having been declared poisonous by the Bureau of Animal Industry. This plate has, however, attractive properties, both as a rust preventive and as a lubricant, where there is no contact with food products.

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[1] The author was engineer for the Link-Belt and Lamson companies from 1905 to the World War and consulting material handling engineer since.

Contributed by the Materials Handling Division and presented at the Annual Meeting, New York, N. Y., Nov. 30 to Dec. 4, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

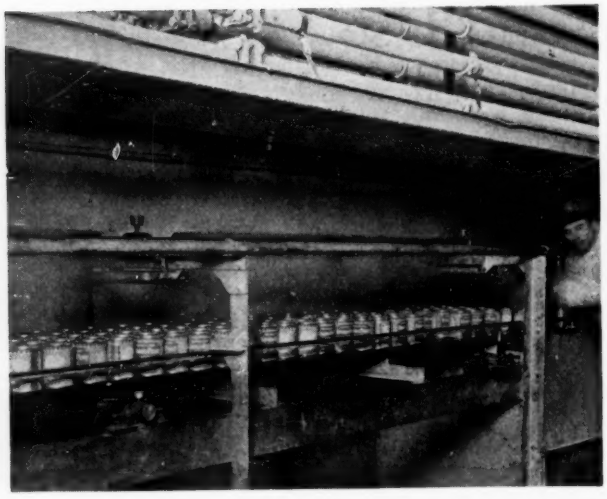


FIG. 1



FIG. 2

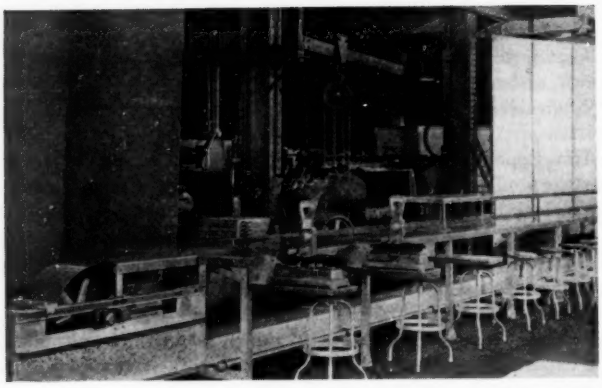


FIG. 3

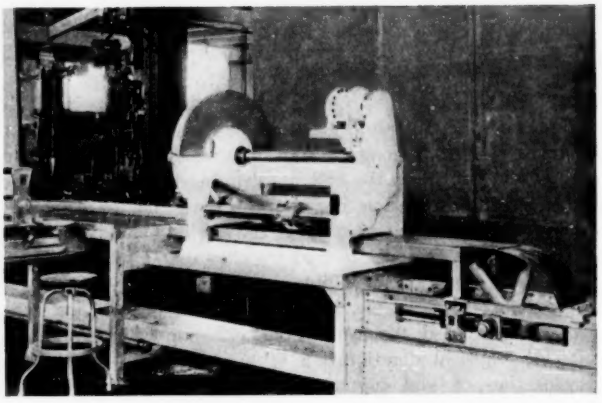


FIG. 4

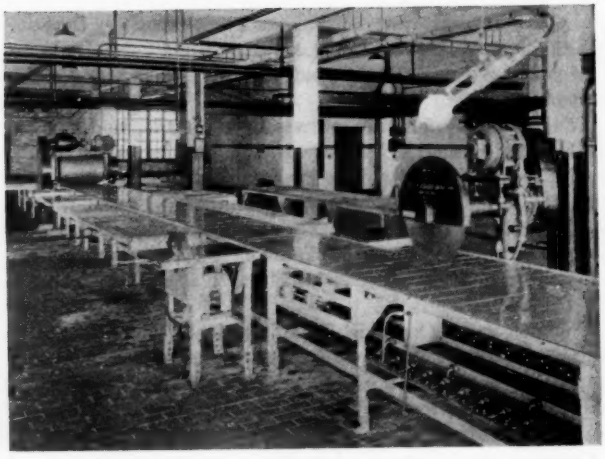


FIG. 5

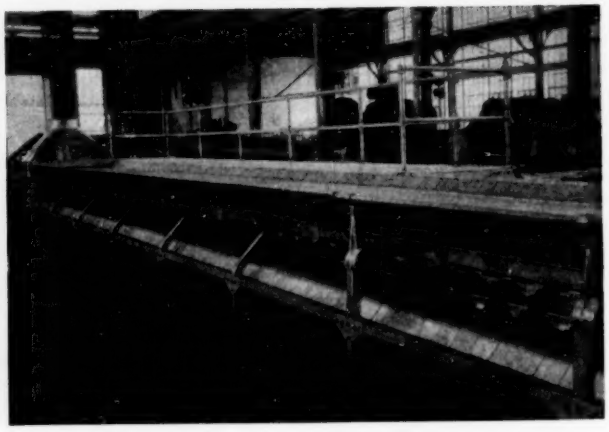


FIG. 6

With this outline of materials and their properties, consideration will be given to the types of conveyors for which these and similar materials may be used and are commercially available. These conveyors are as follows:

1 Steel belt conveyors:

(1) Woven wire: Woven of almost any material that is available in wire form.

(2) Doormat type: The choice is not as wide here, as the material must be available in both strip and wire form. The wearing qualities of this type are beyond expectation from a casual inspection of the design. The stretch is small, and all that is needed is good alignment and a good-sized, many-sided polygon for its pulleys.

(3) Swedish iron: This is an imported solid-steel conveyor belt that is not soluble in fruit or fatty acids, but which, when clean, will rust in clear water and air after the fashion of cast iron. (The author has used this conveyor with good success for trimming out cocoanut meats, for trimming fat and lean pork, and for conveying bacon from the slicing machine in front of the packers, as well as for the filling and cooling of canned lard.) The fundamental requirement here is, however, accuracy of alignment. This was particularly noticeable in the case of the cooling conveyor, Fig. 1, where the range in temperature was about 150 deg. and the speed was less than 1 ft. per min. (As the pulleys are practically uncrowned, it is necessary to have efficient scrapers on them.)

2 Apron and pan conveyors:

(1) A typical installation of a pork-cutting table is shown in Fig. 5. The frame is hot-dipped galvanized after fabrication and before assembly, and the flights are made of Monel, with rivets of the same material. As a matter of cleanliness, return rolls must be used to support the lower run. To minimize rust, malleable-iron chains are used, their size being determined by the amount of material and the area exposed to rust and wear, instead of by listed strength. The large knife in the foreground revolves in the same direction as the table and slightly faster, cutting in the groove between the two conveyors, which are kept in accurate alignment by V-shaped blocks on every flight and continuous V-shaped guides. The short right-hand conveyor is driven from the take-up of the main conveyor, and the shoulder is cut off by the knife and delivered over the end of the short conveyor. After the hams, shoulders, loins, and ribs are removed, the sides are flattened out under the heavy double roller shown at the left. The same motor drives the rolls and the conveyor, insuring synchronism.

(2) Fig. 6 shows the latest type of beef-viscera table. This is 5 ft. wide, so that the operators can stand on it, and is synchronized with the overhead chain carrying the carcasses. The moving part of the table passes through a continuous sterilizer on the return run at the take-up end. The flights are of $^1/_2$-in. firebox steel, with the edges planed, jigged, hot-dipped, and wiped. Neither zinc nor ordinary steel seems to be able to withstand the action of blood. It is the expectation of the designer of this table that, at the end of ten years, there will not be more than $^1/_4$-in. deterioration through rust of the $^1/_2$-in. hot-dipped galvanized firebox steel. The extremely heavy malleable-iron combination chain has the same life expectancy. The top-vent sterilizer is shown at the left. The overhead synchronized cattle chain brings the carcass in contact with the near side of the flight at the sterilizer, with the head projecting under the flights, but not touching the framework or stationary part of the conveyor, which is not subject to continuous sterilization. The operators ride the conveyor with the carcass and walk back on the platform shown on the far side. The overhead conveyor, with the carcass, turns away from the table near the middle of the picture, where the frame straightens up and the guard rail begins. The government inspectors operate from here on. Both runs are supported throughout on self-lubricating rollers and never touch any stationary part of the conveyor. The underlying patent on the sterilizing and synchronizing features has been dedicated to the public.

(3) A hog-viscera table is usually made of either 18-8 steel or Monel, with pans. There is a patent on trays and another on pans. The trays or pans are made 3 ft. wide and 3 in. deep on two strands of malleable-iron chain. Half of the pans are 30 in. long and half are 6 in. long, alternating. There are six double sets of K2 attachments under the long pans, with two pairs of plain links between. Only the center pairs of these attachments are riveted; the others merely support the loaded pan in the horizontal position.

3 Screw conveyors:

(1) These can be and have been made in both 18-8 steel and Monel. From the nature of the screw action the 18-8 is to be preferred, provided the purchaser will agree to accept the odd diameters that the manufacturers' standard equipment for rolling helical flights will turn out in this material.

4 Bucket elevators:

(1) Bucket elevators are made in these materials, but as the buckets are comparatively few in number and usually bolted in place, it is generally satisfactory to regalvanize or retin these when required, except in a few special cases, such as fancy table-salt and certain special cosmetic items.

(2) Any of these materials that are available in wire form may be used in the elevator type of self-cleaning screens which are standard for water intakes and sewage-disposal plants.

5 Various other forms of conveyors can be obtained in rust-resisting materials, such as scraper conveyors, gravity roller conveyors, etc., but it is so difficult to keep these types mechanically clean that there is less occasion to justify the added expense.

In conclusion, the author believes that future progress along these lines will be rapid as soon as the patent and licensing situation is straightened out on a tonnage production basis and the metallurgists learn to produce 18-8 steel from the chrome ore direct. Until something of this sort is accomplished, it will be economically impractical to make general use of 18-8 for rolled shapes and shafting, attractive as the idea is to the food-products manufacturing plant.

Discussion

S. Craig Alexander.[3] The author's statement on the order of cost, taken with the table on the first page, is likely to prove

[3] San Francisco, Calif.

misleading. Comparing, for example, Monel metal and the 18-chromium–8-nickel steel, it is not brought out that Monel metal weighs about 0.320 lb. per cu. in. against about 0.283 for the 18-8 steel, thus giving a percentage difference in weight in favor of the 18-8 steel of a little over 13 per cent. Taking two pieces of the same size and thickness, the Monel metal would weigh over 13 per cent more than the 18-8, thus affecting the cost to this extent were the two priced exactly the same per pound.

There would be no occasion for using a greater thickness of the 18-8 steel than of the Monel metal, because the elastic limit and modulus of elasticity are higher in the 18-8 steel than in the Monel metal, while the resistance to wear or the abrasion resistance of the chromium-nickel steel alloy similarly is superior to that of the nickel-copper alloy. If anything, the thickness of the design could be reduced with the 18-8 steel over the same design in Monel metal without sacrifice of safety and service.

Author's Closure

The two points brought out by Mr. Alexander and a third fact (that 18-8 is more expensive to work than Monel) were carefully considered, and the manufacturing costs of a concern which advertises its willingness to furnish either Monel or 18-8 at the same price were checked over a year's period before this table was passed.

This table is just as correct in the final analysis as in its more obvious interpretation

MH-55-3

Application and Design of the Floating Drive for Conveyors

By N. H. PREBLE,[1] DETROIT, MICH.

Increasing demand for longer conveyors and conveyors more heavily loaded per foot of length has brought about the necessity of using multiple drives on a single strand of conveyor chain in order to keep the weight and cost of chain and carrier units down to a reasonable point. To meet this requirement there has been developed a floating type of driving mechanism so arranged that two or more drives located at considerable distances from each other will automatically synchronize as to speed so as to maintain proper driving action at each drive point. The paper describes in detail the development of this design and its application to various types of conveyors, pointing out the advantages obtained through its use not only as a synchronizing means, but also in properly distributing the total load, in effecting smoother operation, in replacement of the standard form of shear pin, etc.

AS IS TRUE in most forms of engineering construction, the demand for industrial conveyors has tended during the last decade toward larger and larger units—units larger not only as to weight of material carried, but also as to length of line. This has been, perhaps, more especially true in the overhead trolley conveyor, a design particularly suitable as a transportation or processing medium. (See Fig. 1.) Due to the flexibility of layout which is possible with this conveyor, great length is entirely feasible, so far as layout and general conditions are concerned. Many other forms of conveyors, such as the slat conveyor, the belt conveyor, or any double-strand construction, can operate only in a straight line at right angles to the center line of chain pins. They may be curved up or down, but cannot be turned in a horizontal plane. Their length is then limited by the maximum length of the building within which they are installed. With the overhead conveyor, however, or with certain other types of single-chain construction, it is possible to curve the conveyor not only up or down, but also to one side or another, so that it may be run from one floor to the next, through bridges from one building to another, curving at will in either plane. (See Fig. 2.) It is therefore apparent that in the large plant or in a plant having a number of buildings located fairly close together there is practically no limit to the length of conveyor which may be laid out to transport material from one place to another within the building or grounds.

In view of the fact that the floating drive was originally designed particularly with a view to use on the overhead conveyor and its application is peculiarly suited to such a conveyor, a few brief comments in regard to the design of this equipment will perhaps be in order.

Essentially the overhead continuous power-driven trolley conveyor consists of some form of monorail track, usually an I-beam, although occasionally made up of two angles or a pair of rails of some other description. On this monorail track are operated 2-wheel or 4-wheel trolleys and ordinarily having anti-friction bearing in the wheels. These trolleys are attached to a conveyor chain at suitable intervals, and from the trolley or with lighter loads, from the chain at intermediate points between the trolleys, are suspended hooks, trays, or racks for carrying the material

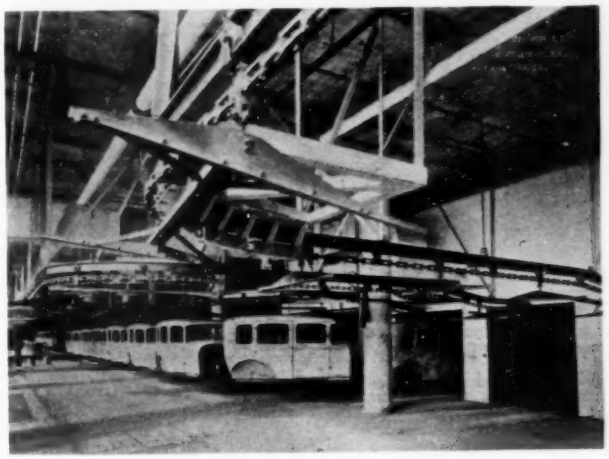


FIG. 1 TYPE OF OVERHEAD TROLLEY CONVEYOR

to be transported. The conveyor chain itself is of a flexible construction which can turn not only in the usual manner, but also can be bent in the plane of the pin center line. If the track is curved upward or down, the trolleys, being guided by the track, will rise or descend, guiding the chain in a line parallel to the track itself. Turns can be made in a horizontal plane, the conveyor chain in this case serving as the guiding medium and running against either a sprocket or a series of small rollers. Power is applied directly to the chain through a conventional type of driving sprocket or through a so-called caterpillar drive, which consists of a chain driven from a small-pitch sprocket and carrying dogs which engage with the conveyor chain and serve to propel it.

In laying out a conveyor of this type a typical calculation would be as shown in Table 1, where a conveyor is assumed having a total length of 4600 ft., with trolleys spaced at 2-ft. intervals, each trolley carrying a hook from which the loads of 80 lb. each is to be suspended. It is assumed that the line will be 80 per cent loaded and that there are no vertical lifts to be considered in figuring the power required; in other words, that any loads that are carried up an incline also run down the same distance, so that they will balance. The minimum radius of the horizontal turn, which is given as 24 in., will be determined by the clearance

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required between the loads as they pass around these horizontal turns. The minimum allowable vertical radius will depend upon layout conditions and should be kept as large as possible so as to reduce to a minimum the bending stresses imposed on the chain and the reactions of the chain on the trolley at the vertical bends. Usually a conveyor of this sort will be most efficient if

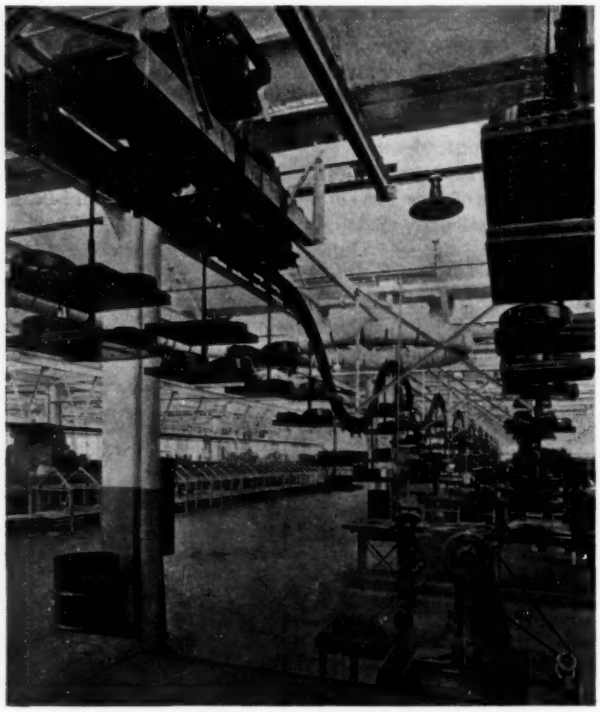


FIG. 2 OVERHEAD CONVEYOR OF SINGLE-CHAIN CONSTRUCTION THAT CAN CURVE UP OR DOWN

arranged with a variable-speed drive unit, so that the speed can be changed at will to meet varying production conditions. In this example it is assumed that the speed is to vary from a minimum of 10 ft. per min. to a maximum of 50 ft. per min. In order to determine the size of units involved, the first calculation must be as to the tension required in the chain at the drive to operate the conveyor, which will be a function of the total load moving plus the friction produced by the vertical and horizontal bends. The total load will consist of the weight of chain which, to start off with, will be assumed at 10 lb. per foot, plus the weight of trolleys, assumed at 12 lb. each, plus the weight of hooks calculated, knowing their design, together with the total live load carried. The total figures out to 232,300 lb. Assuming that the trolleys will have some form of anti-friction wheel, experience indicates that for the average layout the total friction will be approximately 5 per cent of the total weight of the moving load. The chain pull at the drive will then be approximately 11,600 lb. To determine this exactly, it would be necessary to examine the layout more carefully, calculating the known friction of the trolleys plus the friction at each turn, which is a direct function of the chain tension at the turn. In other words, the friction at each turn or bend will increase as the driving mechanism is approached and as the chain pull approaches its maximum. A 5 per cent figure, however, would be indicated by experience of average conditions. The horsepower required would then figure out to approximately 14 hp. to operate the line at its maximum speed of 50 ft. per min., which would involve the use of a somewhat larger motor to take care of frictional losses through the driving mechanism itself. If it is assumed that there will be used a caterpillar type of drive unit with a 30-in. power-drive sprocket, the torque on the head shaft would be 174,000 in-lb., requiring a shaft approximately $6^1/_2$ in. in diameter.

Examining these calculations it is immediately evident that with a conveyor of this length, carrying a heavy load, all of its component parts must be of very strong construction. The chain-pull requirement is over 11,000 lb., which means that the chain, in order to take care of the conditions which arise on the vertical bends, should have an ultimate strength of around 120,000 lb. The trolleys will have to be designed not only to carry the imposed load of 80 lb. plus 5 lb. for the hook, but also must be capable of withstanding the chain-pull reaction on the vertical bends. With a 2-ft. trolley spacing and vertical bends on 10-ft. radius, this reaction at points adjacent to the drive unit will run up to a figure somewhat over 2300 lb., so that instead of having a trolley which will merely carry the load, there must be one that will withstand momentarily this considerably heavier reaction. The track also will have to be designed at the curves to take such a reaction, and the horizontal turns will have to be braced progressively stronger as the drive unit is approached. The drive itself will have to be extremely massive in construction, for while the power requirements are relatively low, the torques involved are large, due to the slow speed at which the chain is traveling. Under actual conditions it probably would not be feasible, from an economic standpoint, to install a conveyor to meet these requirements. The cost would be excessively high, and there would be a dead weight equivalent to very nearly the total of the live weight or useful load.

It is at once apparent that if this conveyor can be split into four separate units, there can be effected some very considerable economies. If one were handling bulk material on a belt conveyor, it would be a simple matter, because such material can be

TABLE 1 TYPICAL CALCULATION FOR SINGLE-CHAIN CONVEYOR

Total length of conveyor, ft	4600
Trolley spacing, ft	2
Weight of load per trolley, lb	80
Weight of hooks, lb	5
Per cent of line fully loaded	80
Minimum radius of horizontal turns, in	24
Minimum radius of vertical bends, ft	10
Speed in feet per minute, variable	10 to 50
Moving load	**Single drive**
Chain, 4600 ft. at 10 lb	46,000
Trolleys, 2300 ft. at 12 lb	27,600
Hooks, 2300 ft. at 5 lb	11,500
Live load, 2300 × 0.80 at 80 lb	147,200
Total, lb	232,300
Coefficient of friction, per cent	5
Chain pull at each drive, lb	11,615
Power at drive sprocket (11,615 × 50/33,000), hp	17.5
Diameter of drive sprocket, in	30
Torque on head shaft, in.-lb	174,125
Head-shaft diameter, in	$6^1/_2$

TABLE 2 SINGLE-CHAIN CONVEYOR WITH DRIVE APPLIED AT MORE THAN ONE POINT

Total length of conveyor, ft			4600
Trolley spacing, ft			2
Weight of load per trolley, lb			80
Weight of hooks, lb			5
Per cent of line fully loaded			80
Minimum radius of horizontal turns, in			24
Minimum radius of vertical bends, ft			10
Speed in feet per minute, variable			10 to 50
Moving load	**Single drive**	**Four drives**	
Chain, 4600 ft. at 10 lb	46,000	at $3^1/_2$ lb	16,100
Trolleys, 2300 ft. at 12 lb	27,600	at 7 lb	16,100
Hooks, 2300 ft. at 5 lb	11,500	at 5 lb	11,500
Live load, 2300 × 0.80 at 80 lb	147,200	at 80 lb	147,200
Total, lb	232,300		190,900
Coefficient of friction, per cent	5		4
Chain pull at each drive, lb	11,615		1,909
Power at drive sprocket (11,615 × 50/33,000) hp	17.5	(1909 × 50/33,000) hp	2.9
Diameter of drive sprocket, in	30		20
Torque on head shaft, in.-lb	174,125		19,090
Head-shaft diameter, in	$6^1/_2$		3

easily spilled from one section to another, but if one is handling a package, using the word "package" in the broad sense, it is quite essential that there be one continuous chain, to avoid the necessity of making a manual transfer or having extremely complicated automatic transfer devices. It is therefore desirable to have some form of driving medium which can be applied to the chain at more than one point.

Assume for the moment that such multiple units are available and analyze again this same conveyor. (See Table 2.) It found first of all that one can cut the size of the chain and use one of considerably lighter weight. The trolleys will not have to stand the extremely high reactions involved in the first example, so can also be reduced in weight. The hook load and live load will remain the same, but the total is immediately reduced to 190,000 lb. in place of the previous total of 230,000 lb. As the maximum chain pull will be considerably less than previously, it is safe to presume a friction load of 4 per cent in place of the 5 per cent due to decreased turn friction. A maximum total chain pull of 7600 lb., or a total horsepower under 14, is then indicated. If it is assumed that there will be used four drive units, the maximum chain pull at each unit, if they are evenly balanced, will be 1900 lb., and one can safely employ four 5-hp. motors. The chain being lighter, can be economically furnished in a shorter pitch, so that one can reduce the diameter of the caterpillar-drive sprocket. The torque required at the drive shaft, assuming a 20-in. pitch diameter sprocket, would be 19,000 in-lb., requiring a shaft of $2^{5}/_{16}$ in. in diameter. Specifications such as this would be quite usual practice in overhead trolley construction and would be met by the use of 4-in. I-beam track with a two-wheel ball-bearing trolley attached to a 4-in. pitch No. 458 forged chain, having an ultimate strength of 30,000 lb., giving a factor of safety of 15 to 1, based on the calculated maximum chain pull of 1900 lb. It also will be noted that the total live load is close to 150,000 lb., whereas the weight of the conveyor proper which must be moved is only 45,000 lb., or 1 lb. of conveyor to about $3^{1}/_{2}$ lb. of pay load.

It might be well to point out another factor which frequently enters into this problem of drive design and strength of units. Frequently one will have an existing conveyor which it is desired to extend. If multiple drives are used, this can be done without any change to the existing construction, whereas if one had to drive at only one point the added loads might make necessary new trolleys and chain.

In order to operate satisfactorily, the first essential of any multiple drive is that all driving points must be exactly synchronized with each other. If one drive should be running only a very small fraction of a revolution faster per minute, it will very quickly tend to pull the conveyor away from the other drives and assume the total load itself; a condition of synchronization of all separate driving points must be maintained exactly at all of the varying speeds, a problem which is somewhat more difficult to solve than would be the case if the conveyor was run only at a constant and fixed speed.

Under certain conditions a conveyor of this sort may loop back toward itself, and there will be several points where sections of the conveyor will be running parallel to each other and fairly close together. If such points occur, it is possible to drive by means of a single motor and variable-speed unit to a countershaft, and from this countershaft to the reduction gearing and final drive sprocket of the drive unit proper, thus securing exact synchronization of head-shaft speed. Such an arrangement has been used frequently and is fairly satisfactory so long as the pitch of the chain is the same throughout its length and a spot can be found where the necessary number of lines run adjacent to each other at reasonably close intervals and so located as to divide the conveyor as to length or load into approximately equal intervals. (See Fig. 3.) Such a condition is almost impossible except where the conveyor is confined to a single building or even to a single floor of one building. It also should be noted that such an arrangement does not allow for any compensation of load between the individual driving units to effect an equalization should the conveyor be of different lengths between the units or differently loaded.

If only a constant speed is required, satisfactory results can be obtained by the use of the ordinary form of synchronous alternating-current motor drive through a reduction gearing to a driving sprocket, fitted with a ratchet or one-way clutch. If one

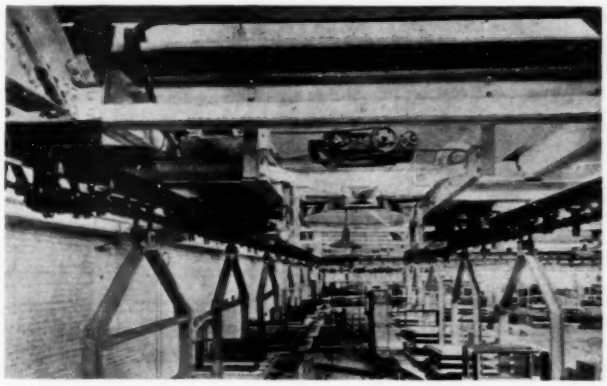


FIG. 3 SECTIONS THAT ARE PARALLEL MAY BE DRIVEN FROM ONE MOTOR

drive is getting more than the proper load, the motor will be slowed down automatically, whereas under an opposite condition, the chain will pull away from the drive unit through the ratchet mechanism and the motor will speed up. A number of installations of this type have been made and operated, but experience would indicate that a large number of separate drive units will be required for satisfactory results.

Various forms of electrical synchronization have been attempted with varying success. Most of them have, however, been rather complicated and expensive, and involve a considerable amount of excess wiring if the units are located at any distance from each other.

In order to overcome the disadvantages of other methods and to provide a simple and fully automatic synchronizing medium, the floating type of drive was developed and has been applied with entire success to overhead conveyors and also to other forms of conveyor construction. Fig. 4 shows a typical caterpillar drive mechanism consisting of a motor driving through a silent chain or V-belt to a variable-speed transmission. This variable-speed transmission may be of the belt type, as manufactured by the Reeves Pulley Company or the Llewellyn Manufacturing Company, or may be of the positive chain type, such as the P.I.V. unit manufactured by Link-Belt Company. Any of these three variable-speed transmissions operate by means of varying the position of the V-cones on which the belts or chain operate so as to change the ratio between the two shafts of the transmission, the change being effected by a screw mechanism operating on a pair of levers which push the V-cones closer together or pull them further apart as required. The range from the minimum ratio to the maximum between the two shafts is in an infinite number of steps. In other words, one can get as exact adjustment as required and not depend upon a limited number of steps as in the usual form of variable-speed motor. The variable-speed shaft of this transmission is connected through a chain or direct through a flexible coupling to some form of reduction gearing, and from the reduction gearing either direct or through separate gears to

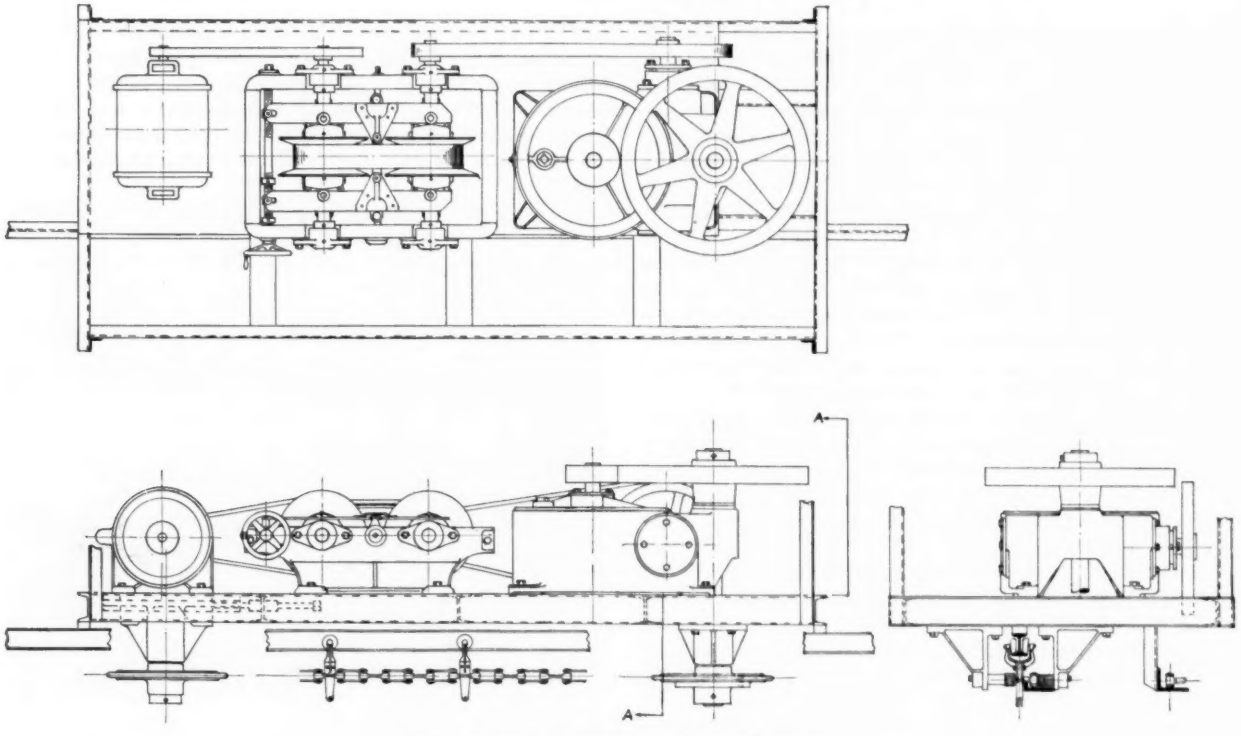


FIG. 4 TYPICAL CATERPILLAR-DRIVE MECHANISM

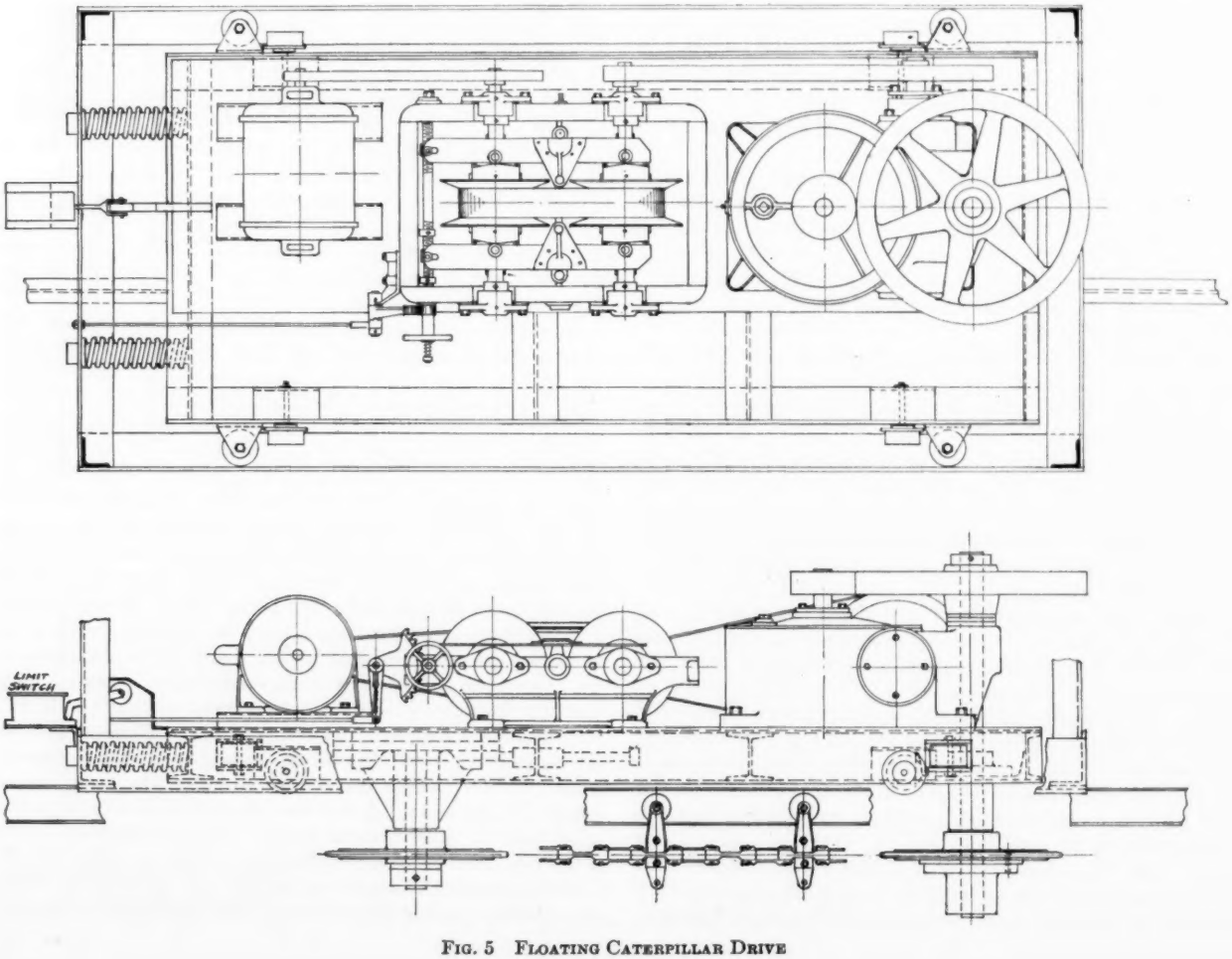


FIG. 5 FLOATING CATERPILLAR DRIVE

the driving head shaft. The entire mechanism is rigidly mounted on the same form of frame, the driving sprocket operating a caterpillar chain and the same frame carrying a take-up sprocket around which the caterpillar chain passes. The floating drive mechanism utilizes this unit frame and machinery units, exactly as indicated in the diagram of Fig. 4, mounted in a separate and fixed frame. (See Fig. 5.) The frame of the caterpillar drive proper is carried on rollers and guided within the secondary frame by means of vertical rollers. As the caterpillar chain operates in a direction parallel to the driven conveyor chain, the reaction of the drive effort on the conveyor chain will move the frame parallel to the conveyor, this movement being counteracted by a pair of coil springs attached to the fixed frame and bearing against the floating frame. The strength of these springs is adjusted properly to counterbalance the calculated chain tension at the individual drive.

It is at once apparent that if the chain tension or driving effort goes above the calculated figure, the coil springs will be compressed beyond their normal position, and conversely if the reaction is less than normal or calculated figure, the floating portion of the drive will be moved in the opposite direction by the springs. This tendency to movement is the operating medium of the speed control. The adjusting shaft of the variable-speed transmission is connected to the fixed frame through a series of gears and levers, so that if the drive moves back under excessive load, it will be automatically slowed down, or if it moves forward under less than normal load, it will be automatically speeded up. The result is that each drive will at all times adjust its speed to a certain chain pull, which will be determined by the strength and adjustment of the coil reaction springs. The detail of design of the connection between the fixed frame and the adjustment shaft of the speed transmission is so arranged as to allow for making the control more or less sensitive by variation in length of lever arms.

Exactly the same principle can be applied to a sprocket type of drive, the machinery framework in this case rotating around the head shaft under the influence of the torque of the shaft. This rotating tendency is again counteracted by means of a coil spring, the movement being used to vary the adjustment of the transmission.

This design will give in effect an exact synchronization regardless of motor speed or variation in motor speed and regardless of the location of the drives with respect to each other. It will further produce a synchronization or equalization of loads between the various units. If one section is carrying more chain than another or is more heavily loaded than another section, the various drives will adjust themselves to such variations and each will help out the other. Actual service tests on an installation of approximately 4000 ft. in length having three drives showed that the maximum in-put to any one of the three never exceeded 35 per cent of the total and the minimum was never less than 31 per cent of the total regardless of loading. The floating portion of the drive is further arranged in such a way that if for any reason the drive compresses the springs past a certain fixed point it will contact with a limit switch and shut off the current from the motor, thus acting as an automatic overload release to protect the conveyor and driving mechanism against any jam or snagging of the line, and taking the place of the usual form of shear or breaking pin. It shuts off the current without damage to any parts, and the unit can be restarted as soon as the line is clear, without the necessity of replacing broken pins, which is frequently the source of delay and annoyance.

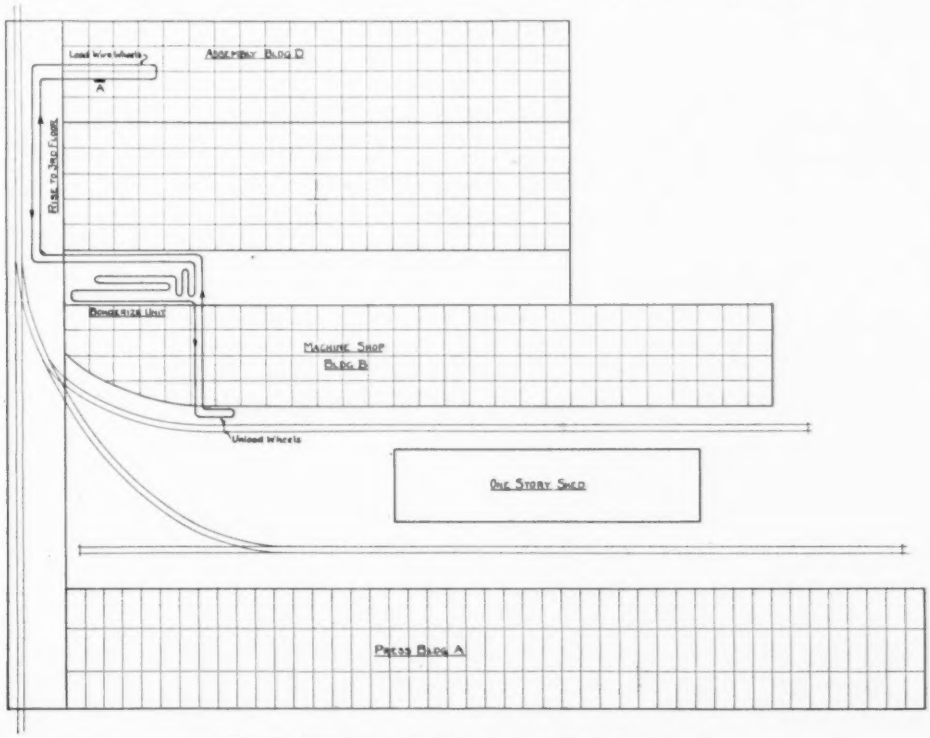


FIG. 6 OVERHEAD CONVEYOR AS AT FIRST INSTALLED

The various limit switches can easily be connected to a signal system so that at a central control point the drive that is shut off will immediately be indicated, and the place to look for trouble will be apparent.

It has been found by experience that a floating drive reduces the tendency of a conveyor to pulsate, the spring back-up of the driving frame smoothing out the power impulses, acting more or less in the nature of a shock absorber to take up sudden loads. A floating frame also will automatically care for variations in the

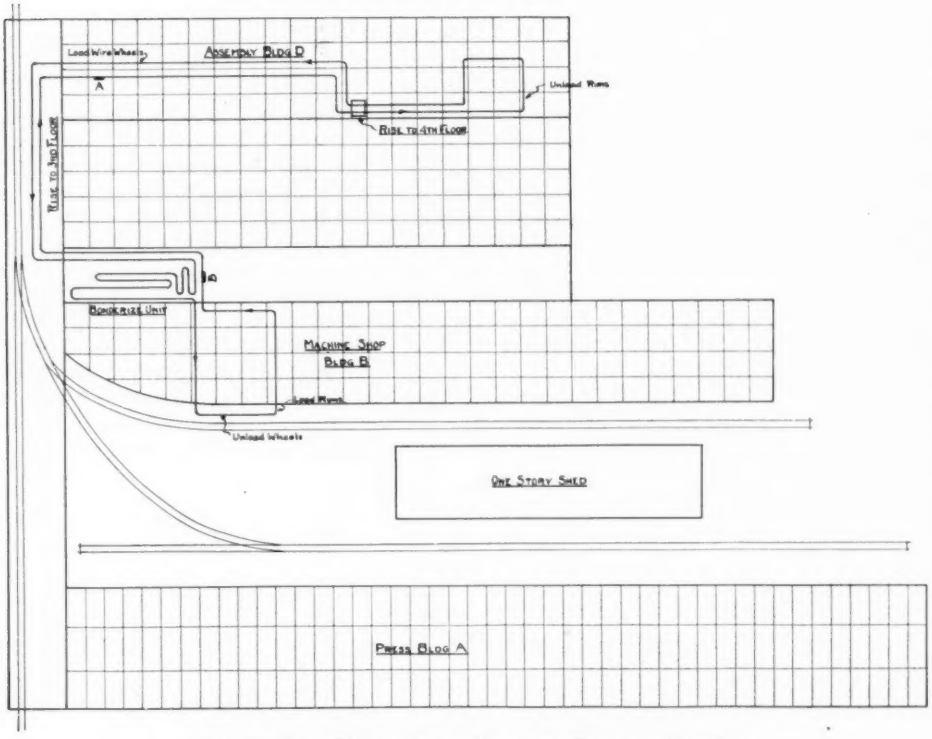


FIG. 7 FIRST EXTENSION OF CONVEYOR SHOWN IN FIG. 6

pitch of the chain in different sections, so that it is feasible to use a new section of chain with one which has been in use and has stretched or worn somewhat out of pitch.

It is not absolutely essential that the drives be located so as to each carry the same theoretical load, thus making it possible to put the driving mechanism at a point which may be most convenient.

A typical installation of an overhead conveyor with this type of drive is shown in Figs. 6, 7, and 8. Fig. 6 illustrates the original installation which was made to carry wire wheels through a bonderizing department, the wheels being loaded on the conveyor on the fourth floor, carried down through the bonderizing process, and then out to the shipping dock, where they were unloaded directly to cars for shipment. At a later date this same conveyor was extended to the layout as shown in Fig. 7, rims being loaded on the conveyor on the dock and carried up to the fourth floor, where the line was extended and the rims were taken to the assembly department. This increase in length, together with the added weight of the rims being carried from the shipping dock up to the fourth floor, imposed a load that was too severe for the original drive unit, chain, and trolleys. A second drive therefore was added, and both were mounted on floating frames. The conveyor was operated with entire satisfaction for a period of about six months under these conditions. At that time it became necessary for the user to bonderize another type of wheel, which was produced in another building, located some 250 ft. from the bonderizing unit. It was expensive to truck these wheels to the bonderizing unit and then bring them back to their starting point for enameling as would be necessary. In fact, it was almost physically impossible to do this by ordinary trucking without seriously interfering with production. The conveyor was therefore extended to the second building, running through a bridge between the two, and at the same time was further extended so as to pick up other parts produced in this building and carry

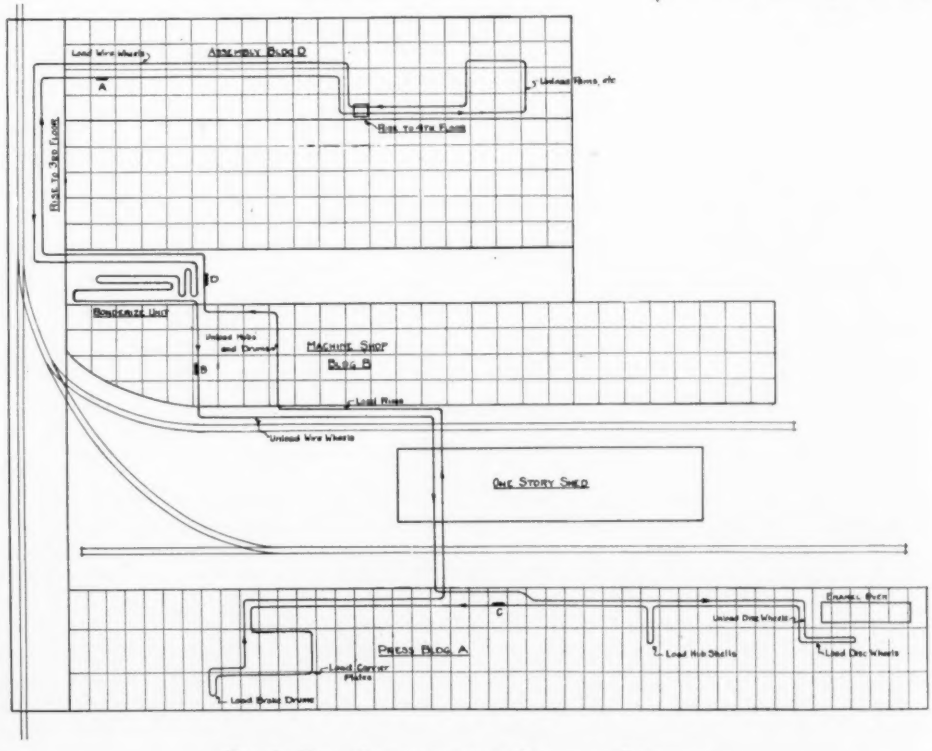


FIG. 8 FINAL INSTALLATION OF OVERHEAD CONVEYOR

them to other sections for machine operations or assembly. The final arrangement of the conveyor, as shown in Fig. 8, is approximately 5100 ft. long and employs four separate drive units, all of the floating type—a conveyor which would be practically impossible with a single driving mechanism, certainly

economically unwise even though physically possible from the engineering standpoint.

The discussion has been confined almost entirely to the application of this driving mechanism to the overhead type of conveyor. It is possible, however, to apply the same principle to any other conveyor construction. If one could conceive of a slat conveyor 10 miles or more long, there is no reason why a series of floating drives could not be so arranged, driving through caterpillar chains on both the load and return sides, as to make such an arrangement entirely practical from an engineering standpoint. In one particular case, applied to a single-strand motor assembly conveyor, there existed three separate conveyors of relatively short length, and due to changing manufacturing conditions, it was decided that it would be desirable to combine the existing conveyors to some additional material to make one long conveyor, the total load on which would be far beyond the capacity of the existing chain or driving mechanism. One of the existing drives was installed on the loading end of the new conveyor, this being fixed and operating at a certain speed as determined by the adjustment of the variable-speed transmission, and drives only the return strand plus a short section of the loads run. A second drive of the caterpillar type, the caterpillar mechanism only being mounted in a floating frame, was added to the load strand side of the conveyor just ahead of the horizontal turn. This floating caterpillar operating against two coil springs was connected by gearing to the adjusting shaft of the variable-speed transmission of a power unit, located in a pit alongside the booster, and thus synchronized with the fixed unit. The installation of this type of drive therefore made possible the utilization of considerable existing equipment of value, and as it was located immediately ahead of the horizontal turn, it materially reduced the wear of the chain operating around this turn.

The mechanism is extremely simple, requires no accurate adjustments, and involves no parts which can readily get out of repair. The maintenance problem is not complicated, and the results achieved in actual use have been satisfactory.

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St. Paul.....James Jerome Hill Reference Library

Mississippi

Columbus.....Mississippi Agri. & Mech. College

Missouri

Columbia.....University of Missouri
Kansas City.....Public Library
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Washington University
Mercantile Library

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Nebraska

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Durham	University of New Hampshire

New Jersey

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Camden	Free Public Library
Elizabeth	Free Public Library
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Jersey City	Free Public Library
Newark	Newark College of Engineering
	Free Public Library
New Brunswick	Rutgers University
Paterson	Free Public Library
Princeton	Princeton University
Trenton	Free Public Library

New York

Albany	New York State Library
Brooklyn	Polytechnic Institute
	Pratt Institute
	Brooklyn Public Library
Buffalo	The Grosvenor Library
	Engineering Society of Buffalo
	Buffalo Public Library
Ithaca	Cornell University
Jamaica, L. I.	Queens Borough Public Library
New York	Engineering Societies Library
	Public Library
	College of the City of New York
	Cooper Union
	Columbia University
	N. Y. University Library
Potsdam	Clarkson College of Technology
Rochester	Rochester Engineering Society
Schenectady	Union College
Syracuse	Syracuse University
	Public Library
Troy	Rensselaer Polytechnic Institute
Utica	Public Library
Yonkers	Public Library

North Carolina

Chapel Hill	University of North Carolina (Engineering Library)
Raleigh	North Carolina State College

North Dakota

Fargo	North Dakota State College
Grand Forks	University of North Dakota

Ohio

Ada	Ohio Northern University
Akron	Public Library
	University of Akron
Canton	Public Library
Cincinnati	University of Cincinnati
	Public Library
	Engineers Club of Cincinnati
Cleveland	Public Library
	Case School of Applied Science
	Cleveland Engineering Society
Columbus	State of Ohio Library
	Public Library
	Ohio State University
Dayton	Engineers Club of Dayton
Toledo	Public Library
	University of Toledo
Youngstown	Public Library

Oklahoma

Norman	Oklahoma University
Oklahoma City	Public Library
Stillwater	Oklahoma Agricultural and Mechanical College
Tulsa	Public Library

Oregon

Corvallis	Oregon State Agricultural College
Portland	Portland Library Association

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Allentown	Free Library
Bethlehem	Lehigh University
Easton	Public Library
	Lafayette College
Erie	Public Library
Lewisburg	Bucknell University
Philadelphia	Engineers Club
	Drexel Institute
	University of Pennsylvania
	Franklin Institute
Pittsburgh	University of Pittsburgh
	Engineers' Society of Western Pennsylvania
	Carnegie Institute of Technology
	Carnegie Library (Schenley Park)
	Carnegie Free Library of Allegheny
Reading	Public Library
Scranton	Public Library
State College	Pennsylvania State College
Swarthmore	Swarthmore College
Villanova	Villanova College
Wilkes-Barre	Public Library

Rhode Island

Kingston	Rhode Island State College
Providence	Brown University
	Providence Engineering Society
	Public Library

South Carolina

Clemson College	Library, Clemson College

Tennessee

Kingsport	Public Library
Knoxville	University of Tennessee
Memphis	Goodwin Institute
Nashville	Vanderbilt University

Texas

Austin	University of Texas
College Station	Agricultural & Mechanical College of Texas
Dallas	Public Library
El Paso	Public Library
Fort Worth	Carnegie Public Library
Houston	Rice Institute
	Public Library
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San Antonio	Carnegie Library

Utah

Salt Lake City	University of Utah
	Public Library

Vermont

Burlington	University of Vermont

Virginia

Blacksburg	Virginia Polytechnic Institute
Charlottesville	University of Virginia
Norfolk	Carnegie Public Library
Richmond	Virginia State Library

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Pullman	State College of Washington
Seattle	Public Library
	Engineers Club
	University of Washington
Spokane	Public Library
Tacoma	Public Library

West Virginia

Morgantown	West Virginia University

Wisconsin

Madison	Library, University of Wisconsin
Milwaukee	Public Library
	Board of Industrial Education, Vocational School Library
	Marquette University

Wyoming

Laramie	Wyoming University

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Bibliotheca Nacional
Sao Paulo.....Bibliotheca da Escola Polytechnica

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Engineering Institute of Canada
Toronto.....University of Toronto, Library

Chile

Santiago.....Universidad de Chile, Facultad de Ciencias Fisicas y Matematicas (Engrg. School)

Czechoslovakia

Prague.....Masarykova Akademie Prace
Association of Czechoslovak Engineers

Danzig Free City.....Bibliothek der Technischen Hochschule

Denmark

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England

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Bristol.....University of Bristol
Cambridge.....University of Cambridge
Leeds.....University of Leeds
Liverpool.....Public Library of Liverpool
London.....City & Guild Engineering College
Institution of Mechanical Engineers
Institution of Civil Engineers
Institution of Electrical Engineers
The Royal Aeronautical Society
Manchester.....Manchester Public Libraries (Reference Library)
Oxford.....University of Oxford
Newcastle-upon-Tyne.....The North East Coast Institution of Engineers and Shipbuilders

Wales

Cardiff.....Cardiff Public Library

France

Lyons.....University of Lyons
Paris.....Écoles Nationales des Arts et Metiers
École Supérieure de Mécanique et d'Aeronautique
École Centrale des Arts et Manufactures de Paris
Société des Ingénieurs Civils de France

Germany

Berlin.....Bücherei des Vereines deutscher Ingenieure
Bibliothek der Technischen Hochschule
Breslau.....Bibliothek der Technischen Hochschule
Cologne.....Universitäts- und Stadtbibliothek
Dresden.....Bibliothek der Technischen Hochschule
Düsseldorf.....Bücherei des Vereins deutscher Eisenhüttenleute
Frankfort.....Technische Zentralbibliothek
Hamburg.....Bibliothek der Technischen Staatslehranstalten
Hanover.....Bibliothek der Technischen Hochschule

Germany (Continued)

Karlsruhe.....Bibliothek der Technischen Hochschule
Leipsic.....Stadtbibliothek
Munich.....Bibliothek der Technischen Hochschule
Bibliothek des Deutschen Museums
Stuttgart.....Bibliothek der Technischen Hochschule

Hawaii

Honolulu.....University of Hawaii Library

Holland

Amsterdam.....Koninklyke Akademie von Wetenschappen
The Hague.....Koninklijk Instituut van Ingenieurs
Rotterdam.....National Technisch-Schlep-Caartkundig Museum

India

Bangalore.....Mysore Engineers Association
Calcutta.....Bengal Engineering College
Poona.....Poona College of Engineering
Rangoon.....University of Rangoon

Ireland

Belfast.....Queen's University of Belfast

Italy

Milan.....Biblioteca della R. Scuola d'Ingegneria
Comitato Autonomo per l'Esame della Invenzioni
Naples.....Biblioteca della R. Scuola d'Ingegneria
Rome.....Biblioteca della R. Scuola d'Ingegneria
Consiglio Nazionale delle Ricerche presso il Ministero della Educazione Nazionale
Turin.....Biblioteca della R. Scuola d'Ingegneria

Japan

Kobe.....Kobe Technical College
Tokyo.....Imperial University Library
The Society of Mechanical Engineers
Yokohama.....Library of Yokohama

Mexico

Mexico City.....Asociacion de Ingenieros y Arquitectos de Mexico
Library of the Escuela de Ingenieros Mecanicos y Electricistas

Norway

Oslo.....Universitetsbibliothek den Polytekniska Forening

Poland

Warsaw.....Bibljoteka Publicazna

Porto Rico

Mayaguez.....University of Porto Rico

Portugal

Lisbon.....Institute Superior Technico

Roumania

Bucharest.....Scoala Polytechnica din Bucharest

Scotland

Glasgow.....Royal Technical College
Mitchell Library

South Africa

Johannesburg.....South African Institute of Engineers

Sweden

Stockholm.....Kungl. Tekniska Hogskolan
Svenska Teknologföreninger
Gothenburg.....Chalmers Tekniska Institut

Turkey

Constantinople...Robert College

MSP-55-1

Progress in Machine-Shop Practice

ECONOMIC pressure, like unpalatable medicine, must be credited with some virtues, and not the least of these is the urge which it has put upon engineers responsible for design and fabrication, to bestir themselves to heroic efforts.

Despite budgets cut to unprecedented limits in so far as engineering activity is concerned, this motivating urge has produced some phenomenally productive machine tools and improvements in manufacturing technique which will tend, more than ever, to render obsolete the tools and methods which earned adequate profits as near back as 1929 and 1930.

The thought of those who earn their daily bread by the manufacturing arts, rising from the ashes of the high hopes of 1928 and 1929, is that an improved technique can still be produced which will earn a reasonable profit from a materially reduced volume of business. There is no complacency in this attitude; it is simply a case of the survival of the fittest; and the competition between methods and industries under this same economic pressure is, and will continue to be, more intense than ever.

Characteristically in line with former depressions, this one is prolific in hard thinking, and this in turn involves consideration of improved tools and methods which in other times would receive but scant attention.

As never before, the proponents of scientifically welded structures, die castings, precision forging machinery, improved machine tools, cold-heading equipment, and deep-drawing metal are being listened to with undivided attention.

Welded Structures

Despite the difficulties of the times under which we are working, a number of large concerns manufacturing welding equipment have carried a veritable torch of progress to the trade, and early in the year Mr. Everett Chapman, in his paper on "Welded Industrial Equipment With Reference to Machine Tools," quoted the case in a very able way.

While it is evident that welded structures cannot in every case replace castings with economic or engineering advantage, it is equally true that the use of these structures in the past has been hampered by lack of knowledge and some prejudice, and that with the availability of this knowledge, now resulting in a vastly increased usage, it may be safely said that during the next few years we will see a steadily increasing number of products produced in this way, with advantages to all concerned in lightness, cheapness, strength, and efficiency.

Concurrent with this development has also occurred a considerable improvement in the technique of cutting metals by the oxy-acetylene torch process.

One fairly large machine-tool manufacturing concern in particular has systematically combed its designs and substituted welded structures for castings in a great number of instances. To the uninitiated the results have been surprising to a degree in strength, saving in weight, efficiency, and grace. This firm has had many of its welded products in use for a long period of time, and their stability and permanence of accuracy are as outstanding as the other merits referred to.

The evidence is not lacking that, despite the conservative attitude of some designing engineers, the day is not far distant when welded structures for major units will be standard practice.

Die Casting

In many cases during the last few years the possible use of the die-casting method of fabrication has been hampered by the lack of definite information, or rather a formula which would give a quick and accurate method of computing costs. However, this information, under the economic pressure induced by the need for business, has now been made more generally available, and many concerns are looking to the die-casting method of fabrication to give them that margin between manufacturng cost and sales price which represents profit, and incidentally the inducement to continue to manufacture.

Accurate Forging Machinery

Concurrent with the development of more efficient machine tools and cutting elements, there has taken place a rapid increase in the quality of work turned out by up-to-date forging machinery, and in many cases work is now passed direct from the forging machine to the grinding machine without intermediate metal-removing operations. While the investment in such forging machinery is necessarily heavy, the results are economically attractive where work must be turned out in large quantities along mass production lines.

Improved Machine Tools

A questionnaire was addressed to the industry, the first three paragraphs of which read as follows:

1 What are the technical advances which have been made during the past 12 months in your specialties which are now ready for commercial presentation?

2 What advantages will be obtained from their use:
   (*a*) Decreased cost
   (*b*) Increased accuracy
   (*c*) Elimination of operations hitherto considered necessary
   (*d*) Specific commercial appeal.

3 On the resumption of normal production activity, do you look for an increased use of your product due to any of the reasons specified under headings *a*, *b*, *c*, and *d*?

And on the many replies received the following data are based.

Material-Handling Facilities

Without doubt, the most outstanding achievement in the attack on the idle or non-cutting time in the machine-tool cycle is the provision of automatic loading devices for individual machines or groups. While to date the majority of applications have been in connection with lathes and automatic grinding machines generally, there is not the slightest doubt that this type of equipment, which has been described in detail in the technical press during the year, will find many uses as an auxiliary to all types of machine tools that are used on high-production work.

When it is realized that the average machine tool is actually cutting metal only about 30 per cent of the total working hours, it will be appreciated how important this contribution is to the progress of the art.

Presented at the second session of the Machine Shop Practice Division on Tuesday afternoon, December 6, 1932, during the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Personnel of the Executive Committee of the Machine Shop Practice Division for 1932: F. C. Spencer, Chairman; R. E. W. Harrison, Secretary; W. J. Peets, Coleman Sellers 3d, and B. P. Graves.

Lathes

The outstanding technical advances made in lathes and automatic turning machines of all kinds during the present year include primarily an increased power input, higher operating speeds, additional anti-friction bearings, more universal automatic oiling, greater structural strength, the addition of automatic controls, and the use of more safety features—all in preparation for and in anticipation of the more universal use throughout the metal-working industry of the new cemented carbide cutting tool.

The specific advantages offered by these advances are:

(*a*) Decreased Cost. All of the improvements enumerated tend to increase the metal-cutting capacity of the machine tool, consequently to speed up production and decrease production costs.

(*b*) Increased Quality. The increased structural rigidity of the 1932 designs insures more constant alignments and diminution of chatter and vibration, resulting in higher quality of work produced.

(*c*) Reduction of Manual Operation. The addition of automatic mechanisms for controlling the machine's functions and of built-in measuring equipment and work gages eliminates many of the manual operations required by less modern designs, thus increasing the combined efficiency of the machine tool and operator.

(*d*) Specific Commercial Appeal. The specific commercial appeal of the modern improvements in machine tools is mainly that of a greater work return per dollar invested in equipment.

Precision Grinding Machines

The last year has been characterized more by a consolidation of the progress already gained than by the introduction of any radically new developments.

Automatic sizing, or in other words the provision of automatic compensation for the wear of the grinding wheel, continues to gain favor, and a number of interesting installations have been made in plants handling a mass-production program.

In the centerless grinding field the application of an increasing amount of horsepower to the grinding wheel continues, with resultant increase in rate of metal removal, and it may now be said that the precision grinding machine ranks high as a metal-removal tool for finishing operations.

Automatic gaging devices, which provide the operator with a quick and accurate means of sizing the work, are being used in greater numbers, and in some cases these gages are fitted with devices which automatically cause the grinding wheel to retract when the work is within the prescribed tolerance.

Screw Machines

In the screw-machine line one progressive manufacturer has announced the availability of a silent stock support which will prove of great assistance to screw-machine-parts manufacturers due to the fact that the excessive noise that is now caused by the stock rolling in the iron tubes will be practically eliminated by this new patented silent stock support.

Another improvement is a new nut-tapping attachment to be applied to screw machines which makes use of the bent tap, and this attachment lends itself to exceedingly high production, especially for brass nuts. Other types of nuts can be taken care of by making special adaptations of the attachment.

Another item of equal importance is the 30-speed countershaft for screw machines, which makes the screw machine a practical unit for either steel or brass work, and avoids the necessity of having the screw machine arranged for the standard speeds or high speeds, as the 30-speed countershaft machine will be suitable for either steel or brass work.

Drilling Machines

The last year has seen an extension of the progress in design which brought forth the continuous type of production drilling units of both hydraulic and mechanical construction.

One make of multi-spindle machine has a number of spindles operated by a hydraulic feed and a circular rotating hydraulic indexing table. Horizontal drilling heads at one or more of the stations are used according to the character of the job and the operations required.

Milling Machines

In the milling-machine field there has been released a new type of horizontal boring, drilling, and milling machine, which may be arranged with compound table. In general review, it may be said that there have been many contributions to the attack on idle or non-cutting time in the form of duplication of controls at the front and rear of machines, as well as the provision of more automatic features.

The planer-type milling machine continues to gain favor at the expense of the planing machine, and some remarkable economies have been reported by those companies that have had the courage to install this type of equipment.

There has been a general increase in the range of spindle speeds. One concern in particular now makes a practice of providing 32 spindle speeds ranging from 25 to 1800 rpm, and feeds from $^1/_2$ to 62 in. per min. This machine will aid in more efficient milling, as it adapts the machine to the latest cutting material, either tungsten or tantalum carbide, as well as being adapted to high-speed or carbon tools.

Die-Cutting and Engraving Machines

This type of machine, which is steadily gaining in favor, can now be said to represent the most up-to-date practice. High spindle speeds up to 4000 rpm are common, and permit the use of tungsten-carbide cutters.

Gear-Generating Machinery

The manufacturers of this type of equipment appear to have concentrated their activities on the provision of greater accuracy and facilities for testing the finished work.

The facilities for generating spiral bevel gears have been improved, and one manufacturer is in a position to cut hypoid pinions so that a right-hand one and a left-hand one will both operate equally well with one straight-tooth bevel wheel, an arrangement which, in many cases, is very desirable for reversible feed mechanisms.

This manufacturer has also completed a new method of cutting elliptic gears. This new method is a generating process, applicable not only to elliptic gears, but also to other kinds of non-circular gears.

Hobbed Integral Keys

One Middle West concern has succeeded in developing an integral key taper shaft end. These shafts are produced by the hobbing method, using a special hob in a special hobbing machine which has been developed for this purpose. The advantages of this construction over the present S.A.E. standard taper shaft, using a loosely fitting key, are evident.

In this construction is provided a conical section, or land, between the key, which conical section is practically impossible to produce accurately by any other method than hobbing. This gives the necessary seating surface, and in addition to this we

have the integral splines, which provide approximately three times the bearing we have at present by a single key. These splines cannot wabble or work loose as the present loose key does, and it is believed the whole construction provides a very superior drive to the present single loose-key construction.

The mating part to this piece, or hub, is very simply produced by reaming a taper hole as a first operation, and pulling through this hole a standard four- or six-spline broach. These two operations finish the boring of the hub, so that it is then ready to fit very accurately on the hob member.

This construction is particularly applicable to such places as the rear axles and drive shafts of automobiles, although it can be used wherever a severe reversible torsional action is likely to be experienced.

New Cemented Carbide for Cutting Steel

The new composition of cemented carbide for cutting steel permits increases in cutting speed as great as those that can be obtained with cemented tungsten carbide on cast iron and non-ferrous materials. Due to its new characteristics, among which is the elimination of practically all cratering action, or cupping out of the top surface of the tool, caused by the pressure and abrasion of the chip coming off the work, it is now possible to do many jobs formerly impossible with hard carbide tools, such as light finishing cuts on soft steel, where the cratering action was so destructive. The chips seem to slide off the surface of the tool easier, there is less tearing action, and a much improved finish is left on the work.

This is an entirely new development in hard-metal composition, being a mixture of several carbides held together with a binder, and embodying a rare metal carbide not previously used in this class of tools. It is manufactured by coating the cutting particles with a softer material which acts as a binder, mixing thoroughly to obtain a homogeneous and uniform mass, pressing together under enormous hydraulic pressure in a mold, and finally sintering at high temperature in an inert atmosphere in an electric furnace. This cementing process is similar to that employed in the manufacture of cemented tungsten carbide, as developed by Krupp, and produces a remarkably uniform and homogeneous product, free from porosity.

Developments in Drop-Forging Hammers

The steam drop hammer has for years borne the onus of being the least efficient tool in industry. It would be difficult to locate the source of its black reputation because there is nothing with which the efficiency of a hammer may be compared except the efficiency of another hammer, as industry has no operation comparable to drop forging.

It is admitted that steam drop-hammer operation has always been expensive in steam consumption and in maintenance, and that these two items of expense have boosted the machine-hour rate on steam drop hammers to an aggregate very much in excess of any machine requiring a like initial investment.

The efficiency of a steam drop hammer must be rated not merely on the amount of forging accomplished by a given amount of steam, but also on the number of actual operating hours in a given number of possible operating hours. The present-day demand for forgings with tolerances very much smaller than ever before has increased the importance of design which will give continued alignment and accuracy without down time for maintenance. Due to the expensive accessories of steam drop-hammer operation, such as the boiler plant and the billet-heating furnaces, down time is a large factor in multiplying forging costs.

After several years of scientific research and development, one company has put on the market a hammer which will produce more forgings with less steam, and will produce more forgings to any specified tolerance without down time for adjustment, than any steam hammer which preceded it.

This has been accomplished by developing a balanced design, based on laboratory results. The correct relation between ram weight, ram velocity, and anvil weight has been determined, so that a maximum of applied energy is absorbed by the forging, thus diminishing the number of blows necessary.

With the anvil weight determined, the other structural details are designed to give the utmost in rigidity and wear resistance. The frame-to-anvil connections are a combination of tongue and groove, step joint, and overlipping, and are rigid in all directions, with provision for maintaining the essential perfection of fit. The upper ends of the frames are restrained from relative motion by heavy interlocked tie plates.

The cylinder bore, ports, and valves are designed on the developed principles of hammer operation, replacing former standards based on steam-engine characteristics. Steam ports and passages are so designed that fluid flow is free and unimpeded, and no loss in pressure is caused by wire drawing.

Valve leakage has been practically eliminated, the throttle valve being self-seated by steam pressure, and the operating valve being a novel type of slide valve, with provision for adjustment for wear within limits of 0.001 in.

Energy consumption during idling has been materially reduced, and in case of compressed-air operation, the idling stroke can be entirely eliminated. Tests under laboratory conditions show a steam saving of no less than 20 per cent of what was until now considered normal, and there is evidence that the saving may reach 40 per cent.

Lubrication gaged to proper proportions is automatic, an oil reservoir and pump being built into the cylinder. The flow is through cored or drilled passages, with no external or screwed connections. Oil lies within the steam passages when the hammer is shut down, so that it is ready immediately to resume operation. Oil consumption is reduced at least 80 per cent.

Research has extended even to gaskets and packing, which as tested and determined to be most satisfactory, are furnished with the hammer.

OGP-55-1

Report on Oil-Engine Power Cost for 1931[1]

THIS report covers information on 119 oil-engine generating plants, with 330 engines, totaling 190,768 rated bhp. The previous report (1930) covered data on 94 plants, with 283 engines, totaling 161,583 rated bhp. Engines listed in the report are full-Diesel, vertical type, direct-connected to generators, unless otherwise noted in Tables III, IV, and V.

In previous years data were solicited from foreign plants, and the report for 1930 contains data for two plants located in Mexico. This year it was decided to limit the survey to the United States, and all data listed are for plants so located.

Plant Numbers. The system used in former reports of designating plants by numbers has been retained. Numbers identifying plants previously reported correspond to the same plants in this report.

Period Covered. All but four of the plants of this report submitted data for periods of exactly 12 months, which in most cases coincided with the calendar year of 1931. Of the four exceptions, one submitted data for 8 months, one for $7^1/_2$, one for slightly over 12, and one for an unstated period that was less than one year.

Output Represented. The total *net* output for the 119 plants in this report amounts to 333,066,644 kwhr.

Bases for Costs and Performances. Units costs referred to in this report were calculated on the basis of net kilowatt-hours. The net kilowatt-hour output is found by subtracting the power used for plant auxiliaries and station lights from the total gross output of the plant.

Figures given for power output per gallon of fuel oil and of lubricating oil were calculated on the basis of the gross outputs of the individual units and plants.

Load factors and capacity factors were calculated on the basis of the gross outputs.

Formulas defining running engine capacity factor, running plant capacity factor, annual plant load factor, and plant service factor are as follows:

$$\text{Running engine capacity factor, per cent} = \frac{\text{Engine output in gross kwhr} \times 100}{\text{Kw rating} \times \text{number of hours operated}}$$

$$\text{Running plant capacity factor, per cent} = \frac{\text{Plant output in gross kwhr} \times 100}{\text{Total rated kwhr of individual units}}$$

$$\text{Annual plant load factor, per cent (see below)} = \frac{\text{Plant output in gross kwhr} \times 100}{\text{Peak load in kw} \times \text{number of hours in period}}$$

$$\text{Plant service factor, per cent (see below)} = \frac{\text{Total rated kwhr of individual units} \times 100}{\text{Total installed kw} \times \text{number of hours in period}}$$

The expression "rated kwhr" refers to the kilowatt rating of an engine-generator set multiplied by the number of hours operated. For example, if a unit having a rating of 200 kw was operated 4000 hr, the rated kwhr equals 800,000, no matter what the actual output may have been. Thus the denominator of the expression for "Running plant capacity factor" and likewise the numerator for the "Plant service factor" are arrived at by totaling the rated kilowatt-hours of all plant units. In this report, the kilowatt rating of an engine-generator set is considered equal to: Rated bhp $\times$ 0.746 $\times$ 0.9. In this, 0.9 is the assumed generator efficiency.

In the strict sense of its definition, the annual plant load factor cannot be correctly applied to data covering any period other than one year. However, the committee extended the application of this to plants operated 8760 hr plus or minus 2 per cent, using the actual number of hours in the denominator in each case.

The formula for "Plant service factor" requires further explanation for special cases. The expression is an index of the actual number of hours of operation as compared to the total number of hours installed for operation. Therefore, when some units have been installed for longer periods than have others in the same plant, account must be taken of the fact in the calculation. For example, assume that a plant is reported for a twelve-month period, during which time one unit rated at 200 kw, installed before the start of the period, was operated 5000 hr. Six months after the start of the period, a unit rated at 300 kw was installed, and subsequently operated 2500 hr. The plant service factor in per cent is therefore:

$$\frac{200 \times 5000 + 300 \times 2500}{200 \times 8760 + 300 \times 4380} \times 100$$

Since the 300-kw unit was not installed during the entire 8760 hr, but only for 4380 hr, this adjustment must be made.

Fuel-Oil and Lubricating-Oil Data. Lubricating-oil economies of the 90 plants for which running capacity factors could be calculated are shown graphically in Fig. 1, in which the gross kilowatt-hour output per gallon of lubricating oil is plotted against running capacity factor. (Of the 119 plants, 29 did not report unit hours of operation, and running capacity factors could not be calculated for these.) Fuel-oil economies of the same 90 plants are shown graphically in Fig. 2, in which the gross kilowatt-hour output per gallon of fuel oil is likewise plotted against running capacity factor. The values plotted in Fig. 2 are corrected neither for the heat content of the fuel nor for altitude. The first of these possible corrections has been made for Fig. 3, in which the Btu consumption per gross kilowatt-hour is plotted against running capacity factor. In arriving at the values shown, the committee used the following formula:

$$\text{Higher heat content (Btu per gal)} = \frac{1180\,[40(\text{deg A.P.I.} - 10) + 18{,}650]}{\text{deg A.P.I.} + 131.5}$$

Inasmuch as not all of the 90 plants plotted on Figs. 1 and 2 reported the gravity of the fuel oil used, only 74 plants are shown in Fig. 3.

The type of the plant was judged to be that of the engines generating 95 per cent or more of the gross output. The following types of plants are illustrated:

Diesel four-stroke cycle, air injection
Diesel four-stroke cycle, mechanical injection
Diesel two-stroke cycle, air injection
Diesel two-stroke cycle, mechanical injection, separate scavenging
Diesel two-stroke cycle, mechanical injection, crankcase scavenging
Mixed-type full Diesel
Mixed Diesel and semi-Diesel.

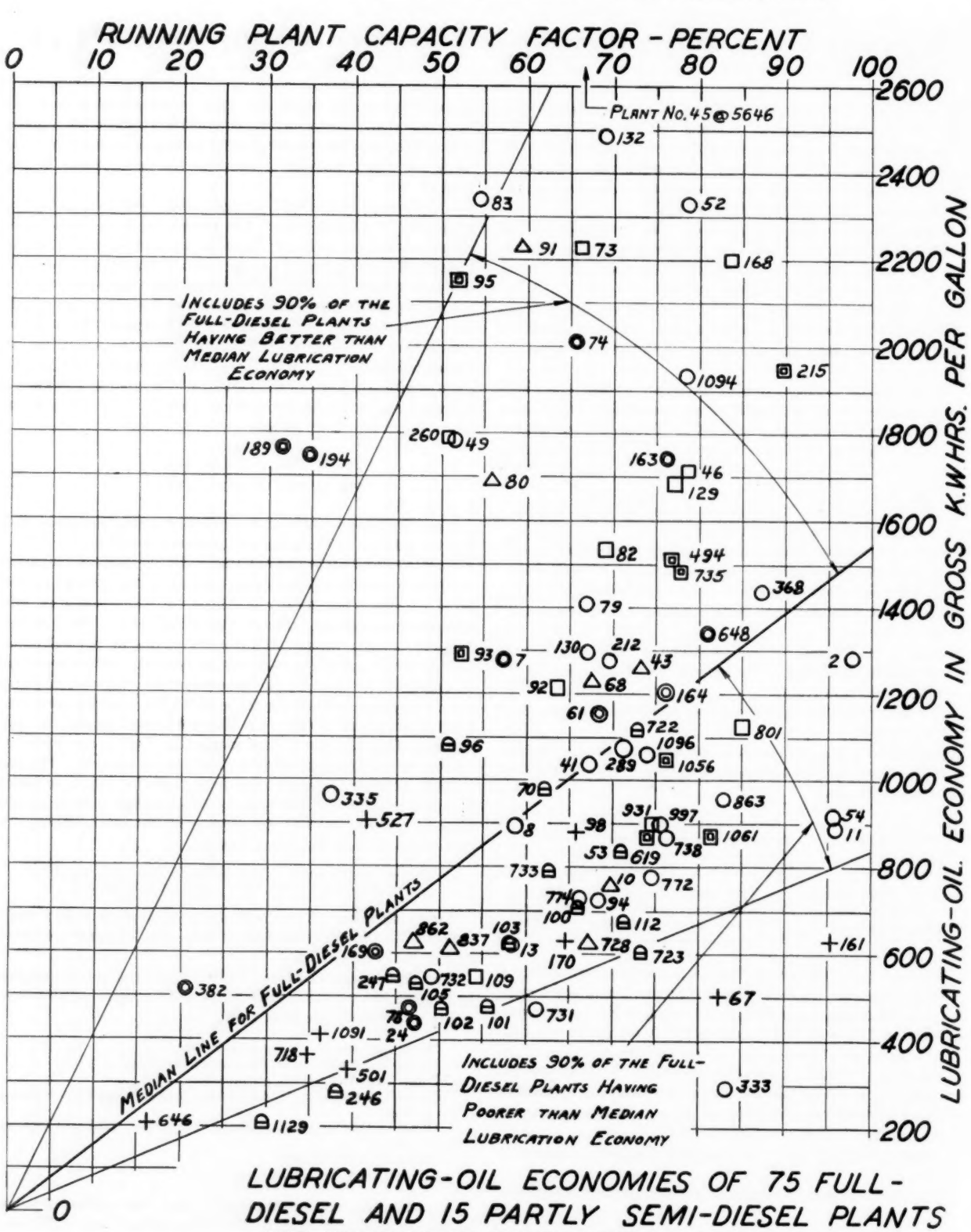
A median line was drawn on each chart, the number of points above the line being equal to the number below—only full-Diesel plants being considered. High and low boundary lines were drawn to include all but 10 per cent of the full-Diesel plants on the high and low sides, respectively.

Unless it is otherwise noted in Table 4, plants received fuel oil in tank cars.

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[1] Submitted by the Sub-Committee on Oil Engine Power Cost, Oil and Gas Power Division, A.S.M.E., Franz Eder, Chairman, Engineer, Robert W. Hunt Company, New York, N. Y.

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LUBRICATING-OIL ECONOMIES OF 75 FULL-DIESEL AND 15 PARTLY SEMI-DIESEL PLANTS -NOT INCLUDING 29 PLANTS, RUNNING CAPACITY FACTORS FOR WHICH ARE NOT AVAILABLE

FOR KEY TO PLANT SYMBOLS SEE FIG. TWO

FIG. ONE

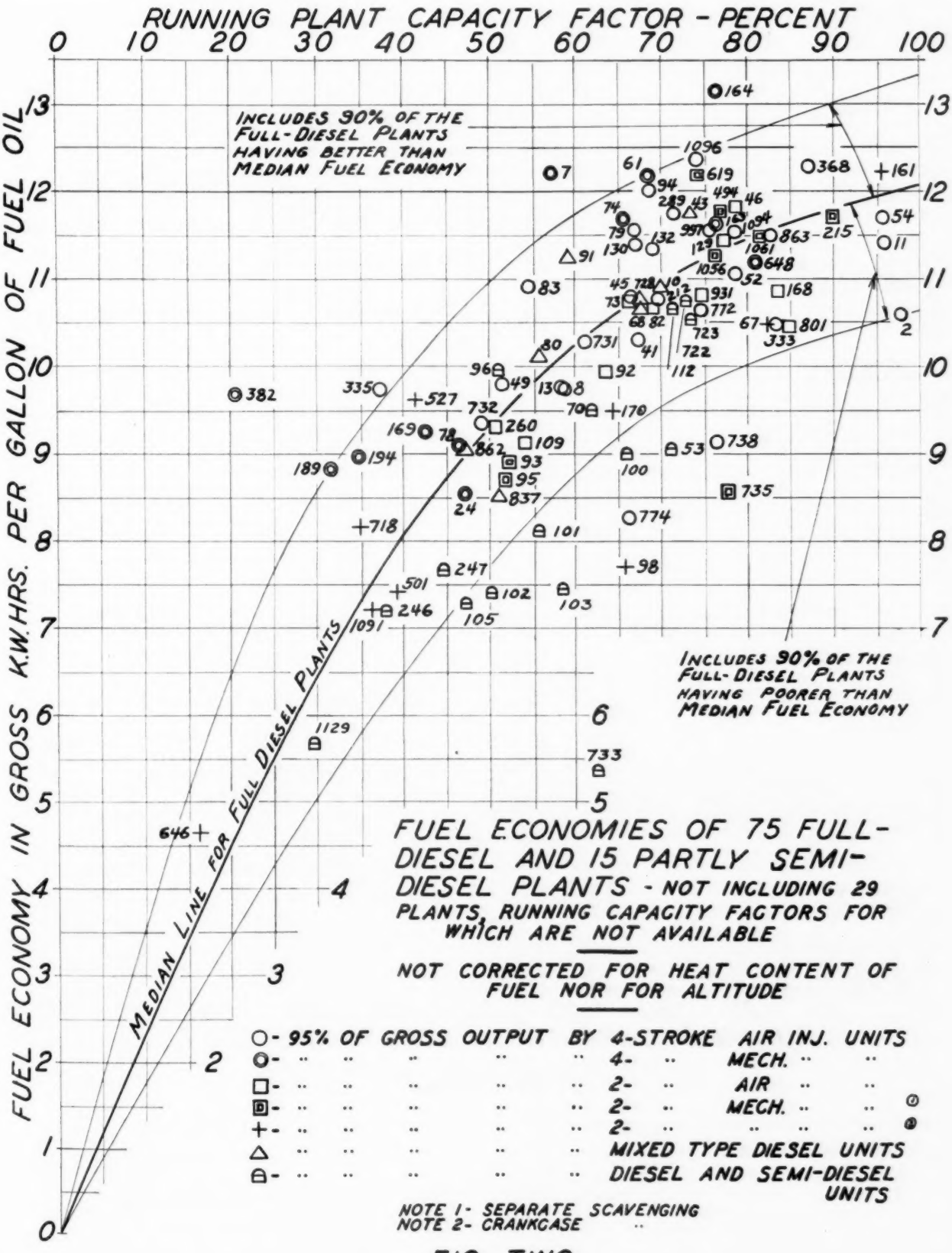

RUNNING PLANT CAPACITY FACTOR - PERCENT
0 10 20 30 40 50 60 70 80 90 100
FUEL ECONOMY IN GROSS K.W.HRS. PER GALLON OF FUEL OIL
0 1 2 3 4 5 6 7 8 9 10 11 12 13
INCLUDES 90% OF THE FULL-DIESEL PLANTS HAVING BETTER THAN MEDIAN FUEL ECONOMY
INCLUDES 90% OF THE FULL-DIESEL PLANTS HAVING POORER THAN MEDIAN FUEL ECONOMY
MEDIAN LINE FOR FULL DIESEL PLANTS
FUEL ECONOMIES OF 75 FULL-DIESEL AND 15 PARTLY SEMI-DIESEL PLANTS - NOT INCLUDING 29 PLANTS, RUNNING CAPACITY FACTORS FOR WHICH ARE NOT AVAILABLE
NOT CORRECTED FOR HEAT CONTENT OF FUEL NOR FOR ALTITUDE
○ - 95% OF GROSS OUTPUT BY 4-STROKE AIR INJ. UNITS
◎ - " " " " " 4- " MECH. " "
□ - " " " " " 2- " AIR " "
▣ - " " " " " 2- " MECH. " " ①
+ - " " " " " 2- " " " " ②
△ " " " " " MIXED TYPE DIESEL UNITS
⊟ - " " " " " DIESEL AND SEMI-DIESEL UNITS
NOTE 1- SEPARATE SCAVENGING
NOTE 2- CRANKCASE "

FIG. TWO

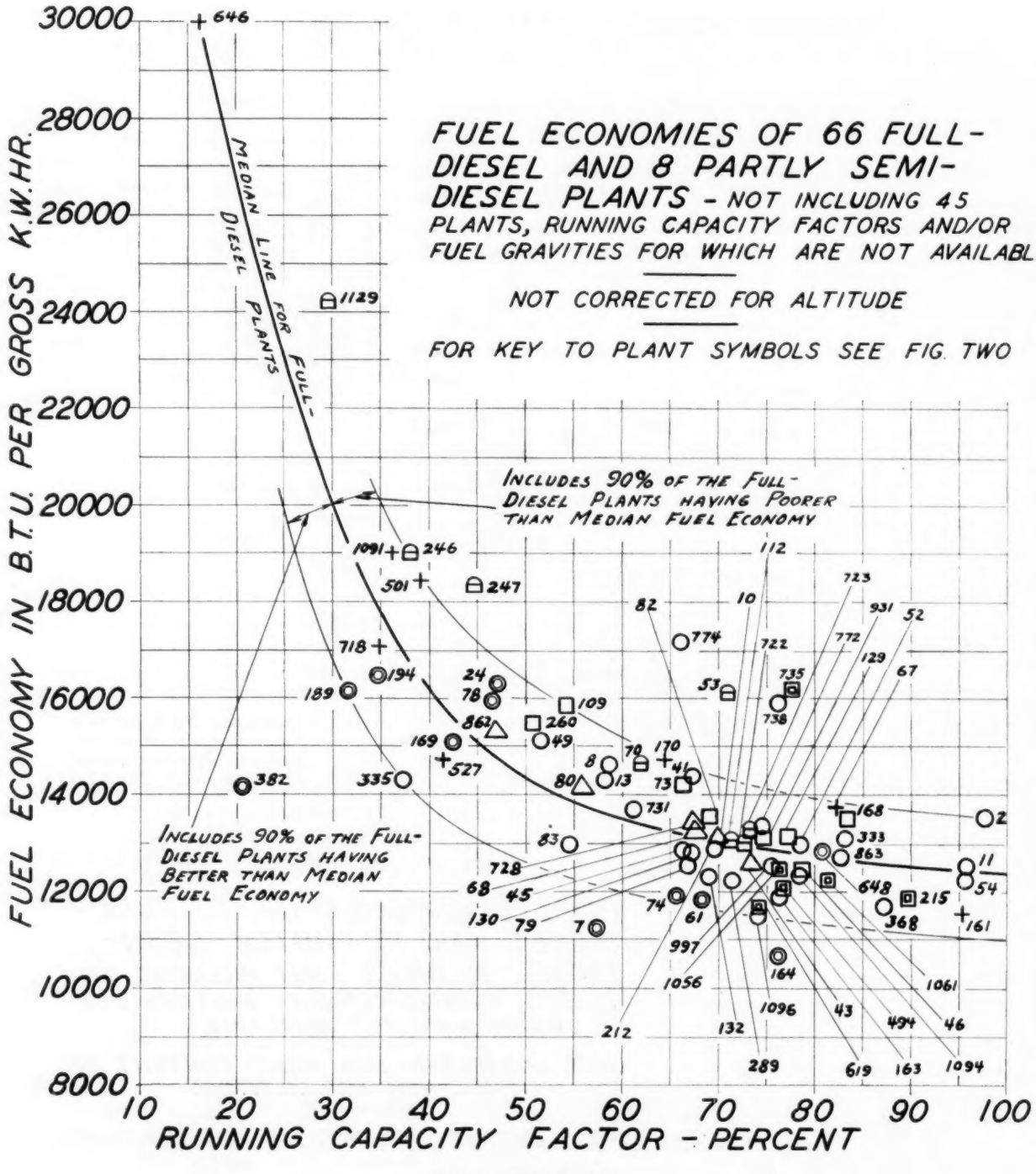

FUEL ECONOMIES OF 66 FULL-DIESEL AND 8 PARTLY SEMI-DIESEL PLANTS - NOT INCLUDING 45 PLANTS, RUNNING CAPACITY FACTORS AND/OR FUEL GRAVITIES FOR WHICH ARE NOT AVAILABLE
NOT CORRECTED FOR ALTITUDE
FOR KEY TO PLANT SYMBOLS SEE FIG. TWO
MEDIAN LINE FOR FULL-DIESEL PLANTS
INCLUDES 90% OF THE FULL-DIESEL PLANTS HAVING POORER THAN MEDIAN FUEL ECONOMY
INCLUDES 90% OF THE FULL-DIESEL PLANTS HAVING BETTER THAN MEDIAN FUEL ECONOMY
FUEL ECONOMY IN B.T.U. PER GROSS K.W.HR.
30000
28000
26000
24000
22000
20000
18000
16000
14000
12000
10000
8000
10
20
30
40
50
60
70
80
90
100
RUNNING CAPACITY FACTOR - PERCENT

FIG. THREE

TABLE I - INFORMATION ON PRODUCTION COST (Page 1)

PLANT NUMBER	CHARACTER OF PLANT (SEE NOTES)	TYPE OF LOAD (SEE NOTES)	NUMBER OF ENGINES	TOTAL INSTALLED B.H.P.	TOTAL INSTALLED K.V.A.	TOTAL PLANT HOURS OPERATED IN REPORTED PERIOD	NUMBER OF MONTHS IN REPORTED PERIOD	TOTAL GROSS OUTPUT-K.W. HRS.	TOTAL NET OUTPUT-K.W. HRS.	PER CENT OF GROSS K.W. HRS. FOR PLANT PURPOSES	ANNUAL PLANT LOAD FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	PLANT SERVICE FACTOR (SEE TEXT)	AVERAGE COST OF FUEL OIL-CENTS PER GALLON	AVERAGE COST OF LUBRICATING OIL- CENTS PER GALLON	COSTS PER NET K.W. HR. - MILLS								PLANT NUMBER
																FUEL COST	LUBRICATING OIL COST	ATTENDANCE COST, INCLUDING SUPERINTENDANCE	COST OF SUPPLIES AND MISCELLANEOUS, INCLUDING WATER	COST OF ENGINE REPAIRS	COST OF ALL OTHER PLANT REPAIRS	COMBINED COST OF ALL REPAIRS, SUPPLIES & MISC.	TOTAL PRODUCTION COST	
43	U	C	8	11,540	11,100	8,760	12	32,634,548	29,849,848	8.5	52.5	73.2	65.7	3.78	53.6	3.51	0.47	0.70	0.28	0.63	0.29	1.20	5.88	43
46	I	B	5	9,250	7,700	6,552	12	19,927,910	18,957,568	4.8	42.1	78.6	46.6	3.73	35.9	3.32	0.22	0.64	0.23	0.99	0.35	1.57	5.75	46
168	I	B	6	8,500	6,250	8,760	12	23,703,300	23,015,138	2.9	54.1	83.5	56.8	4.00	57.2	3.80	0.27	0.49	0.18	0.91	0.12	1.21	5.77	168
82	M-W	C	6	5,660	5,142	8,784	12	8,989,025	8,778,880	2.4	33.2	69.0	39.1	5.13	68.3	4.93	0.45	2.94	0.92	1.96	1.22	4.10	12.42	82
91	U	C	5	5,250	---	8,760	12	8,241,000	7,734,325	6.2	48.2	59.3	45.0	3.20	55.9	3.03	0.27	1.21	0.26	0.44		0.70	5.21	91
73	M	C	4	5,000	4,252	8,760	12	8,815,500	8,341,500	5.3	41.9	66.3	45.1	0.95	55.0	0.93	0.26	1.93	0.13	0.22	0.05	0.40	3.52	73
92	U	C	3	4,900	---	8,760	12	9,731,200	9,378,900	3.6	52.9	63.5	53.1	2.03	41.7	2.12	0.36	1.08	0.20	0.50		0.70	4.26	92
129	I	B	3	4,750	3,700	6,552	12	13,808,100	13,193,230	4.5	50.8	77.1	64.1	3.50	69.1	3.20	0.43	0.63	0.13	0.80	0.07	1.00	5.26	129
52	M	C	4	4,580	3,880	8,784	12	8,572,600	8,417,102	1.8	33.7	78.6	40.4	4.83	68.3	4.46	0.30	2.55	0.63	0.49	0.38	1.50	8.81	52
45	M	C	5	4,250	3,625	8,760	12	6,099,950	5,537,000	9.2	46.4	66.5	36.6	4.79	48.3	4.89	0.09	2.04	0.07	1.28	0.55	1.90	8.92	45
54	I	B	7	4,200	3,150	8,750	12	23,323,000	22,518,000	3.5	92.6	95.6	98.8	1.03	32.0	0.91	0.36	1.19	0.38	1.37		1.75	4.21	54
864	U	S[a]	5	4,000	3,561	3,799	12	5,438,130	5,069,410	6.8	26.9	--	--	2.11	65.0	2.34	0.57	2.53	0.43	1.52		1.95	7.39	864
130	M	C	4	3,600	3,149	8,760	12	4,453,200	4,168,750	6.4	42.4	67.0	31.4	4.19	68.0	3.93	0.56	1.58	0.71	1.96	0.18	2.85	8.92	130
2	W	B	3	3,450	3,000	8,760	12	17,823,240	17,745,325	0.5	86.5	97.8	89.8	2.11	32.8	2.00	0.26	1.09	0.08	1.16	0.09	1.33	4.68	2
164	I	B	4	3,360	4,000	8,729	12	11,589,300	11,037,481	4.8	59.5	76.3	76.9	3.34	32.1	2.66	0.28	1.73	0.35	1.52	0.30	2.17	6.84	164
720	U	C	5	3,080	2,500	8,760	12	3,280,300	3,124,300	4.8	34.1	--	--	3.07	42.1	4.42	1.10	1.43	0.22	0.05	0.35	0.62	7.57	720
109	M	F	3	2,925	2,450	4,380	12	1,902,340	1,858,860	2.3	--	54.1	30.3	5.60	66.5	6.29	1.25	2.32	0.27	0.80	0	1.07	10.93	109
7	M	C	5	2,760	2,625	8,760	12	3,382,400	3,088,885	8.7	43.9	57.3	36.4	3.43	45.0	3.08	0.39	3.79	0.68	1.28	0.51	2.47	9.73	7
732	U	B	5	2,520	2,150	8,760	12	3,212,915	2,632,385	18.1	25.0	49.1	44.1	2.82	50.8	3.68	1.14	2.19	0.59	1.33	0.77	2.69	9.70	732
978	U	V	3	2,520	2,100	---	12	2,303,700	2,247,295	2.4	--	--	--	3.33	54.6	3.11	0.47	1.23	0.42	0.29		0.71	5.52	978
41	M-W	C	3	2,370	---	8,760	12	4,142,730	4,092,730	1.2	39.4	67.3	44.2	4.57	52.4	4.48	0.51	3.85	0.25	0.24	0.13	0.62	9.46	41
723	U	S	4	2,370	2,010	2,769	12	1,649,900	1,556,510	5.7	--	73.3	--	4.21	54.1	4.23	0.95	3.56	0.53	0.86	0.70	2.09	10.83	723
79	M-W	C	5	2,170	1,858	8,760	12	3,378,600	3,204,390	5.2	38.6	66.8	39.6	2.62	40.0	2.38	0.30	3.31	0.13	0.20	0.13	0.46	6.45	79
5	U	C	3	2,160	---	---	12	1,918,770	1,789,653	6.7	10.0	--	--	3.33	50.1	3.86	0.80	2.15	0.79	1.39		2.18	8.99	5
80	M	B	4	2,150	---	8,760	12	2,587,100	2,463,900	4.8	39.9	55.8	36.7	2.41	40.9	2.50	0.25	3.07	0.30	0.48	0	0.78	6.60	80
6	U	C	4	2,100	---	---	12	1,651,300	1,612,957	2.3	11.8	--	--	4.12	49.9	3.53	0.62	1.89	0.31	0.82		1.13	7.17	6
68	M-W	C	4	2,060	1,744	8,760	12	3,485,700	3,336,860	4.4	59.4	67.5	42.7	3.37	52.0	3.30	0.44	1.41	0.11	0.31	0.37	0.79	5.94	68
728	U	C	3	2,040	1,724	8,760	12	3,601,330	3,275,810	9.0	39.1	67.1	44.8	5.20	41.0	5.30	0.72	2.01	0.60	0.39	0.34	1.33	9.36	728
93	U	C	3	1,960	---	8,664	12	3,651,800	3,403,390	6.9	29.8	52.3	60.6	2.79	46.0	3.36	0.38	2.14	0.13	0.31		0.44	6.32	93
215	U	S	2	1,960	1,690	2,389	12	1,601,100	1,568,700	2.1	13.1	89.9	15.5	3.97	39.2	3.45	0.21	1.40	0.38	0.53	0.08	0.99	6.05	215
837	U	B	5	1,875	1,625	8,760	12	2,264,234	2,101,374	7.2	35.9	51.1	40.2	2.72	37.3	3.43	0.65	6.01	0.49	3.36	0.92	4.77	14.86	837
997	I	B	3	1,800	1,425	7,976	12	7,210,320	7,061,000[b]	2.1	78.4	75.5	90.2	5.94	41.0	5.25	0.47	1.22	0.19	0.35		0.54	7.48	997
212	W	S	2	1,750	1,342	8,760	12	229,400	228,700[b]	0.5	3.6	69.6	3.2	4.43	65.0	4.13	0.51	2.21	0.28	0.09	0	0.37	7.22	212
722	U	S	4	1,705	1,450	3,062	12	966,930	826,340	14.6	10.1	72.8	13.2	4.26	54.0	4.63	0.56	8.48	1.03	0.14	0.44	1.61	15.28	722
735	U	P	2	1,680	1,400	4,451	12	2,859,400	2,760,680	3.5	29.7	77.8	37.2	3.33	69.9	4.03	0.49	1.43	0.09	0.52	0.36	0.97	6.92	735
18	U	C	3	1,650	---	---	12	2,502,301	2,406,609	3.8	26.0	--	--	3.94	50.1	3.93	0.58	1.98	0.41	1.83		2.24	8.73	18
8	M-W	C	3	1,560	---	8,760	12	1,902,660	1,876,410	1.3	33.4	58.7	35.3	2.81	35.9	2.92	0.41	3.61	--	--	--	0.54	7.48	8
10	W	C	2	1,500	1,250	8,598	12	3,025,700	3,000,700[b]	0.9	78.5	69.9	49.0	2.45	50.0	2.27	0.66	5.38	0.13	0.84	0.52	1.49	9.80	10
132	M	C	3	1,500	1,289	8,760	12	1,905,300	1,655,300	13.1	38.5	69.1	31.3	3.63	55.7	3.68	0.26	4.30	0.34	0	0	0.34	8.58	132
289	M	C	4	1,465	1,232	8,760	12	2,623,740	2,523,740	3.8	38.4	71.3	42.7	4.15	52.4	3.67	0.51	4.71	0.32	1.12		1.44	10.33	289
865	U	S	4	1,450	1,297	849	12	380,370	364,360	4.2	5.2	--	--	1.53	65.7	1.53	0.90	19.33	2.95	7.41		10.36	32.12	865
70	M-W	C	3	1,420	1,190	8,760	12	2,043,100	1,868,725	8.5	41.7	62.0	39.4	2.83	41.5	3.25	0.46	3.48	0.32	0.85	0.38	1.55	8.74	70
154	U	C	3	1,400	---	---	12	1,641,605	1,608,939	2.0	29.3	--	--	3.59	41.8	2.94	0.22	4.22	0.44	1.14		1.58	8.96	154
163	I	B-S	3	1,400	1,311	8,760	12	4,005,500	3,793,300	5.3	43.6	76.3	63.7	1.92	29.5	1.75	0.16	1.60	0.21	1.95		2.16	5.67	163
11	W	C	2	1,350	1,138	3,398	12	1,336,540	1,328,740[b]	0.6	30.5	95.9	17.5	1.80	36.9	1.59	0.42	5.05	0.14	3.02	0.38	3.54	10.60	11

TABLE I - INFORMATION ON PRODUCTION COST (Page 2)

PLANT NUMBER	CHARACTER OF PLANT (SEE NOTES)	TYPE OF LOAD (SEE NOTES)	NUMBER OF ENGINES	TOTAL INSTALLED B.H.P.	TOTAL INSTALLED K.V.A.	TOTAL PLANT HOURS OPERATED IN REPORTED PERIOD	NUMBER OF MONTHS IN REPORTED PERIOD	TOTAL GROSS OUTPUT-K.W. HRS.	TOTAL NET OUTPUT-K.W. HRS.	PER CENT OF GROSS K.W. HRS. FOR PLANT PURPOSES	ANNUAL PLANT LOAD FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	PLANT SERVICE FACTOR (SEE TEXT)	AVERAGE COST OF FUEL OIL-CENTS PER GALLON	AVERAGE COST OF LUBRICATING OIL--CENTS PER GALLON	COSTS PER NET K.W. HR. - MILLS: FUEL COST	LUBRICATING OIL COST	ATTENDANCE COST, INCLUDING SUPERINTENDANCE	COST OF SUPPLIES AND MISCELLANEOUS, INCLUDING WATER	COST OF ENGINE REPAIRS	COST OF ALL OTHER PLANT REPAIRS	COMBINED COST OF ALL REPAIRS, SUPPLIES & MISC.	TOTAL PRODUCTION COST	PLANT NUMBER
725	U	B	4	1,310	1,097	6,636	12	---	2,640,671	---	---	---	---	3.35	69.2	3.16	0.58	1.78	0.40	0.21	0.15	0.76	6.28	725
13	M-W	C	3	1,125	1,000	8,760	12	1,430,000	1,259,958	11.9	38.4	58.3	---	5.41	39.7	6.29	0.73	4.69	0.24	0.31	0.05	0.60	12.31	13
17	U	C	2	1,200	---	---	12	2,258,800	2,207,300	2.3	27.2	---	---	3.97	51.1	3.83	0.40	1.82	0.42	0.41	1.47	1.89	7.94	17
94	U	S	2	1,200	---	2,683	12	775,410	717,642	7.5	9.0	66.4	16.1	2.26	45.6	2.04	0.68	3.57	5.00	2.47	0.65	1.06	7.35	94
333	U	P	1	1,200	1,000	950	12	637,000	606,740	4.8	9.1	83.3	10.8	5.40	49.6	5.40	1.80	5.00	4.00	2.47	0	6.47	18.87	333
451	M	C	2	1,200	1,024	5,040	12	935,260	833,560	10.9	---	---	---	3.25	36.2	3.47	0.62	4.02	0.36	0.12	0.27	0.75	9.06	451
980	M	C	2	1,200	1,000	8,760	12	2,159,800	2,143,170	0.8	36.5	---	---	5.32	40.0	5.06	0.46	4.39	0.44	0.56	0	1.00	10.91	980
616	M	C	3	1,150	982	8,760	12	1,768,150	1,609,550	9.0	38.8	---	---	4.50	41.5	4.35	0.76	4.32	---	2.48	---	2.79	12.22	616
95	U	S	3	1,120	---	4,662	12	936,650	869,940	7.1	21.4	51.9	27.4	2.35	45.2	2.91	0.23	4.05	0.42	0.72	1.27	1.14	8.33	95
96	U	C	3	1,100	---	8,758	12	1,037,900	967,500	6.9	19.7	50.9	31.5	2.74	45.2	2.95	0.45	5.03	0.20			1.47	9.90	96
721	U	C	3	1,080	900	8,760	12	716,290	716,290	7.9	29.4	---	---	3.22	43.0	6.22	2.40	4.85	0.79	7.36	0.69	8.84	21.71	721
78	I	C	2	1,060	800	8,760	12	2,005,834	1,906,347	4.9	31.4	46.5	69.2	5.07	50.4	5.28	1.13	4.69	0.23	0.15	0.19	0.57	12.27	78
83	M-W	C	2	1,000	860	8,760	12	1,788,400	1,720,560	3.8	33.5	54.4	55.8	6.00	60.6	5.71	0.31	6.46	0.81	1.05	0	1.86	14.34	83
98	U	C	4	1,000	---	8,736	12	1,472,200	1,352,480	8.1	35.4	65.8	38.0	3.44	40.0	4.85	0.64	3.67	0.64	1.78		2.42	11.44	98
738	U	P	1	1,000	720	2,249	12	1,151,600	1,091,000	5.3	21.9	76.3	25.7	3.00	49.5	3.47	0.60	3.73	0.46	1.26	0.35	2.09	9.89	738
801	U	B	1	1,000	937	8,760	12	4,778,300	4,616,205	3.4	78.0	84.9	95.8	3.31	32.9	3.28	0.30	1.04	1.20	0.28	0.02	1.50	6.12	801
1061	U	B	1	980	975	3,822	12	2,048,030	1,974,459	3.6	33.4	81.4	43.7	3.98	46.8	3.59	0.56	1.98	0.18	0.63	0.10	0.91	5.98	1061
863	U	C-B	2	975	875	8,718	12	2,854,500	2,780,400[b]	2.6	59.3	82.7	60.3	3.50	62.0	3.13	0.66	3.19	0.19	0.55	0.05	0.80	6.57	863
49	U	C	2	970	840	8,760	12	1,503,440	1,468,939	2.3	44.0	51.5	51.1	3.97	64.5	4.14	0.37	3.74	1.31	1.26	0.05	2.62	10.87	49
53	U	B	3	900	758	8,742	12	2,453,620	2,355,470[b]	4.0	52.8	71.0	65.2	2.65	42.3	3.06	0.53	2.00	0.42	2.94	0.05	3.41	9.00	53
260	M	S	1	900	750	263	12	80,593	38,740	51.9[c]	1.5	50.6	3.0	5.00	55.5	11.17	0.65	6.24	4.44	0	0	4.44	22.50	260
74	M-W	C	3	885	750	8,760	12	1,193,892[b]	1,172,892	1.7	28.7	65.7	34.8	3.59	58.9	3.12	0.30	5.97	0.22	0.07	0.30	0.59	9.98	74
189	M	C	3	850	730	8,760	12	1,007,351	951,426	5.6	38.4	31.5	64.0	3.68	59.8	4.40	0.36	5.99	0.78	1.66		2.44	13.19	189
153	M	S	1	840	700	---	12	594,940	557,470	6.3	12.1	---	---	3.41	49.9	3.46	0.61	4.71	0.90	0.93		1.83	10.61	153
494	U	B	1	840	700	2,164	12	937,565	915,865	2.3	21.4	76.8	24.7	3.56	50.0	3.09	0.34	1.59	0.70	0.61	0.28	1.59	6.61	494
619	U	B	1	840	700	2,090	12	873,880	828,880	5.1	18.1	74.1	23.9	3.72	53.1	3.21	0.65	1.66	0.85	1.70	0	2.55	8.07	619
1056	U	B	1	840	700	4,822	12	2,073,460	1,906,182	8.0	41.5	76.2	55.0	3.84	49.0	3.70	0.51	1.55	0.50	0.66	0.28	1.44	7.21	1056
731	U	S	1	825	750	1,825	12	618,602	592,895	4.2	11.8	61.2	20.8	4.04	45.1	4.91	1.00	6.58	0.23	0.59	0	0.82	13.31	731
648	U	B	2	800	666	4,774	12	1,208,893	1,180,517	2.4	32.1	80.9	31.7	3.53	50.0	3.23	0.38	2.97	0.75	0.02	0.06	0.84	7.42	648
100	U	S	3	740	---	4,800	12	535,530	498,640	6.9	19.1	65.0	18.6	2.78	40.1	3.31	0.61	6.30	0.69	0.12		0.81	11.03	100
112	M	C	3	740	620	8,760	12	1,294,500	1,195,128	7.7	34.8	71.3	41.8	3.42	46.2	3.47	0.74	5.38	0.18	0.35	0.25	0.78	10.37	112
67	A	C	2	720	500	4,482	12	891,600	891,200[b]	0.1	48.6	82.3	25.6	4.58	43.5	4.38	0.67	2.09	0.24	1.05	0.10	1.39	8.73	67
170	I	C	2	720	600	2,876	12	863,979	835,219	3.3	22.4	64.5	31.6	3.76	47.5	3.78	0.78	2.20	0.20	2.64	0	2.84	12.59	170
527	M	C	2	720	600	8,760	12	1,062,000	933,200	12.2	31.9	41.4	60.5	2.83	64.9	3.35	0.82	3.48	0.29	0.13	0.01	0.43	8.08	527
718	U	C	2	720	600	8,760	12	761,700	691,477	9.2	38.6	34.9	51.4	3.45	44.0	4.86	1.33	5.04	0.68	6.28	0.14	7.10	18.13	718
19	U	C	3	700	---	---	12	747,510	694,060	7.1	34.8	---	---	3.73	54.7	5.27	1.47	5.15	0.80	4.79		5.59	17.48	19
368	I	B	2	700	590	6,250	12	2,297,100	2,253,000	2.1	55.9	87.2	63.9	5.00	66.0	4.15	0.47	3.70	0.89	0.90	0	1.79	10.11	368
61	I	C	1	600	550	8,760	12	1,559,700	1,555,200[b]	0.3	38.7	68.4	64.7	4.20	42.0	3.46	0.36	5.37	0.11	1.07	0	1.18	10.37	61
106	U	C	1	600	---	---	12	1,549,330	1,504,859	2.9	28.1	---	---	3.97	52.0	3.71	0.24	3.05	0.66	1.08		1.76	8.76	106
1094	U	B	1	600	513	2,213	12	700,574	678,530	3.0	19.5	78.5	25.3	3.64	50.0	3.43	0.27	2.65	0.91	1.80	0.37	3.08	9.43	1094
1096	U	B	1	600	513	2,570	12	766,386	726,486	5.2	20.8	74.0	29.3	3.68	50.0	3.13	0.50	4.46	0.76	0.45	0.41	1.65	9.74	1096
103	U	C	3	590	---	8,784	12*	1,062,730	1,015,336	4.4	34.6	58.3	52.4	3.31	40.2	4.63	0.67	4.24	0.69	1.34		2.03	11.57	103
862	U	C	4	588	495	8,760	12	676,320	640,360	5.3	25.7	46.9	41.7	3.27	41.4	3.81	0.70	10.77	0.40	1.24	0.97	2.61	17.89	862
102	U	C	3	580	---	8,760	12	661,730	592,210	10.5	26.0	50.1	38.8	3.97	52.0	3.75	1.05	7.53	0.21	0.46		1.76	13.11	102
101	U	C	3	540	---	8,640	12	761,615	706,879	7.2	48.3	55.7	43.0	2.94	45.1	3.11	1.01	4.34	0.34	0.69		1.03	9.49	101

TABLE I - INFORMATION ON PRODUCTION COST (Page 3)

PLANT NUMBER	CHARACTER OF PLANT (SEE NOTES)		NUMBER OF ENGINES	TOTAL INSTALLED B.H.P.	TOTAL INSTALLED K.V.A.	TOTAL PLANT HOURS OPERATED IN REPORTED PERIOD	NUMBER OF MONTHS IN REPORTED PERIOD	TOTAL GROSS OUTPUT-K.W. HRS.	TOTAL NET OUTPUT-K.W. HRS.	PER CENT OF GROSS K.W. HRS. FOR PLANT PURPOSES	ANNUAL PLANT LOAD FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	PLANT SERVICE FACTOR (SEE TEXT)	AVERAGE COST OF FUEL OIL-CENTS PER GALLON	AVERAGE COST OF LUBRICATING OIL- CENTS PER GALLON	COSTS PER NET K.W. HR. - MILLS								PLANT NUMBER
																FUEL COST	LUBRICATING OIL COST	ATTENDANCE COST, INCLUDING SUPERINTENDANCE	COST OF SUPPLIES, MISCELLANEOUS, INCLUDING WATER	COST OF ENGINE REPAIRS	COST OF ALL OTHER PLANT REPAIRS	COMBINED COST OF ALL REPAIRS, SUPPLIES & MISC.	TOTAL PRODUCTION COST	
246	M	C	3	540	430	8,760	12	427,085	407,085[b]	4.7	33.6	38.0	35.4	6.29	58.7	9.15	2.19	8.78	0.41	0.35	0	0.76	20.88	246
540	M	C	2	540	425	8,760	12	750,000	713,333	4.9	32.9	--	--	3.30	45.7	4.10	1.72	5.52	0.35	0.38	0.06	0.79	12.13	540
772	U	B	2	500	513	1,260	12	233,531	222,100	4.9	9.5	74.5	10.7	3.72	50.3	3.67	0.68	9.55	3.41	3.67	0.56	7.64	21.54	772
501	M	C	3	480	380	8,760	12	473,713	452,970	4.4	37.3	39.3	42.6	3.93	50.9	5.53	1.60	8.50	0.49	0.62	0.33	1.44	17.07	501
615	M-W	C	3	440	369	8,760	12	---	380,000	--	--	--	--	6.00	50.0	8.12	1.53	9.48	0.13	0.26	0.26	0.65	19.78	615
931	I	P-B	1	440	---	3,706	12	814,915	790,715[b]	3.0	--	74.6	42.3	4.60	60.0	4.38	0.69	6.97	0.30	0.13	0.02	0.35	12.39	931
335	M	C	2	430	369	8,760	12	504,000	489,44c	2.8	39.1	37.2	53.6	3.82	60.0	4.03	0.64	9.74	4.06	0.40	0.05	4.51	18.92	335
24	M	C	2	400	326	8,760	12	572,700	548,700[b]	4.2	43.6	47.1	51.9	2.72	51.5	3.10	1.24	7.77	0.66	0.47	0	0.93	13.04	24
194	B	C	1	400	---	4,115	12	384,250	355,490	7.5	23.3	34.7	47.0	3.26	67.3	3.93	0.41	6.34	0.02	0	0	0.02	10.70	194
247	M-W	C	3	390	315	8,760	12	481,150	480,204	0.3	25.1	44.6	47.1	4.62	55.0	6.03	1.01	9.62	0.05	0.40	0.26	0.71	17.37	247
105	U	C	3	370	---	8,664	12	405,580	349,820	13.7	33.1	47.4	39.2	2.97	45.5	4.25	1.01	8.66	0.38	0.97		1.35	15.27	105
152	U	S	1	360	---	---	12	54,600	52,080	4.6	4.2	--	--	4.75	50.0	5.47	0.79	32.57	4.80	7.41		12.21	51.04	152
161	I	C	1	360	300	4,335	12	1,001,000	981,240[b]	2.1	44.8	95.4	49.5	5.33	46.5	4.45	0.76	2.72	0.22	0.10	0	0.32	8.25	161
1091	U	C	2	360	270	8,760	12	382,800	349,169	8.7	33.6	36.2	49.8	5.33	56.0	8.11	1.50	12.02	2.10	0.69		2.79	24.42	1091
688	M	C	3	350	---	8,759	12	---	437,000	--	--	--	--	5.00	31.7	6.69	0.65	6.86	0.51	0.17	0.69	1.37	15.57	688
365	M-W	C	2	345	230	---	12	553,500	493,893	10.8	36.1	--	--	3.95	--	4.51	1.33	11.28	1.17	3.18		4.35	21.47	365
774	U	B	2	345	270	1,000	12	113,247	109,735	3.1	7.2	66.2	8.4	3.79	50.0	4.74	0.71	9.97	3.03	6.09	1.07	10.19	25.61	774
265	M	C	3	320	249	8,760	12	---	216,000	--	--	--	37.0	7.79	87.8	11.81	1.83	11.57	0.32	0.46	0.58	1.36	26.57	265
27	U	C	2	315	---	---	12	229,080	208,880	8.8	30.8	--	--	4.06	50.1	7.72	1.52	10.58	0.95	6.34		7.29	27.18	27
382	I	C	2	300	---	4,414	8	91,300	86,867[b]	4.8	--	20.5	37.4	3.35	39.9	3.62	0.82	4.47	0.43	0	0	0.43	9.34	382
1129	U	C	3	300	269	8,760	12	212,800	195,850	8.0	30.4	29.7	40.7	4.02	45.3	7.65	2.38	16.28	0.92	5.52	0.83	7.27	33.62	1129
156	U	C	2	270	215	8,760	12	199,500	158,236	20.7	29.2	--	--	3.18	42.7	8.34	4.10	16.55	3.23	2.10	0.51	5.89	34.88	156
1097	W	C	2	270	215	---	12	298,264	297,534[b]	0.3	31.0	--	--	5.50	44.8	6.46	0.89	1.52	0.69	0.41	0.02	1.12	10.01	1097
169	I	C	1	240	200	2,793	12	191,340	184,360	3.7	13.7	42.5	31.9	5.56	51.5	6.23	0.89	2.11	0.87	4.39	0	5.26	14.49	169
516	M	C	2	240	188	8,740	12	---	185,000	--	--	--	--	4.30	48.0	7.21	1.37	13.12	0.38	0	0	0.38	22.08	516
222	U	C	2	180	138	8,760	12	44,000	35,224	19.8	33.5	--	--	3.46	42.5	19.78	7.23	33.83	4.87	5.69	1.42	11.98	72.82	222
733	U	S	2	175	120	52	12	3,360	3,293	2.0	0.3	62.5	0.5	5.78	70.6	10.93	0.91	10.93	9.72	0	0	9.72	32.49	733
645	M	S	1	120	90	8,758	12	23,340	21,035	9.9	5.3	16.1	20.4	7.00	47.4	16.70	2.57	16.97	3.28	0	0	3.28	39.52	646
201	U	C	2	100	103	8,760	12	71,640	62,464	12.8	37.2	--	--	7.45	46.7	23.48	5.80	36.70	4.74	14.50	0.70	19.94	85.72	201

NOTES: Character of Plant
U - Private Power Company
P - Private Pumping Company
M - Municipal Power Plant
W - Municipal Pumping Plant
I - Industrial Power Plant
B - City Building Power Plant

Type of Load
C - Complete Power
B - Base Load
S - Standby
P - Peak Load
V - Type varied during period

Lettered Notes
a - Standby for last 8 months of period
b - Estimated by Operator
c - High plant use due to electric heaters for fuel and lubricating oil operated continuously.

TABLE II - COMPARATIVE COSTS - 1929, 1930 and 1931 REPORTS (Page 1)

PLANT NUMBER	YEAR	TOTAL INSTALLED B.H.P.	TOTAL NET OUTPUT - K.W. HRS.	ANNUAL PLANT LOAD FACTOR	RUNNING PLANT CAPACITY FACTOR	PLANT SERVICE FACTOR	AVERAGE COST OF FUEL OIL-CENTS PER GALLON	COSTS PER NET K.W. HR. - MILLS								YEAR	PLANT NUMBER
								FUEL COST	LUBRICATING OIL COST	ATTENDANCE & SUPERINTENDANCE COST	COST OF SUPPLIES & MISC., INCLUDING WATER	COST OF ENGINE REPAIRS	COST OF ALL OTHER PLANT REPAIRS	TOTAL OF ALL SUPPLY, REPAIR, & MISC. COSTS	TOTAL PRODUCTION COST		
43	1929	11,540	22,591,027	--	69.9	--	4.25	3.82	0.69	0.60	--	--	--	1.04	6.15	1929	43
	1930	11,540	26,221,176	46.1	70.4	57.8	4.37	3.95	0.53	0.94	0.21	0.67	0.05	0.93	6.35	1930	
	1931	11,540	29,849,848	52.5	73.2	65.7	3.78	3.51	0.47	0.70	0.28	0.63	0.29	1.20	5.88	1931	
46	1929	6,900	24,909,500	69.4	83.3	76.3	4.92	4.48	0.19	0.83	0.32	1.33	0.22	1.87	7.38	1929	46
	1930	9,250	27,444,480	60.7	83.6	67.2	3.62	3.35	0.23	0.61	0.19	0.95	0.25	1.39	5.58	1930	
	1931	9,250	18,957,568	42.1	78.6	46.6	3.73	3.32	0.22	0.64	0.23	0.99	0.35	1.57	5.75	1931	
82	1930	5,660	7,958,851	31.3	65.0	35.9	5.71	5.22	0.74	0.73	1.36	1.91	1.08	4.37	13.86	1930	82
	1931	5,660	8,778,880	33.2	69.0	39.1	5.13	4.93	0.45	2.94	0.92	1.96	1.22	4.10	12.42	1931	
91	1929	3,450	7,585,725	46.6	52.0	72.8	3.90	3.72	0.48	1.18	0.16	0.22	0.07	0.45	5.82	1929	91
	1930	5,250	8,167,025	53.7	53.1	52.4	4.09	3.95	0.29	1.06	0.16	0.33	0.04	0.53	5.83	1930	
	1931	5,250	7,734,325	46.2	59.3	45.0	3.20	3.03	0.27	1.21	0.26	--	--	0.70	5.21	1931	
73	1930	5,000	6,016,000	37.9	64.0	44.4	1.30	1.34	0.31	2.62	0.01	0.10	0.01	0.12	4.39	1930	73
	1931	5,000	8,341,500	41.9	66.3	45.1	0.95	0.93	0.26	1.93	0.13	0.22	0.05	0.40	3.52	1931	
92	1929	2,500	6,662,389	44.1	71.6	61.6	2.15	2.20	0.55	1.24	0.17	1.85	0.12	2.14	6.13	1929	92
	1930	4,900	10,027,739	56.8	65.0	61.1	2.09	2.25	0.33	0.92	0.26	0.26	0.06	0.58	4.08	1930	
	1931	4,900	9,378,900	52.9	63.3	53.1	2.03	2.12	0.36	1.08	0.20	--	--	0.70	4.26	1931	
52	1929	3,330	5,722,032	31.1	65.3	45.2	5.58	5.38	0.55	3.83	0.88	0.74	0.40	2.02	11.77	1929	52
	1930	4,580	7,257,703	33.5	75.2	45.4	5.70	5.30	0.38	2.83	0.74	0.56	0.48	1.78	10.29	1930	
	1931	4,580	8,417,102	33.7	78.6	40.4	4.83	4.46	0.30	2.55	0.63	0.49	0.38	1.50	8.81	1931	
45	1929	4,250	7,442,350	36.8	72.5	43.6	5.23	5.04	0.09	1.65	0.06	--	--	0.92	7.70	1929	45
	1930	4,250	6,908,500	34.9	69.5	42.4	5.29	5.18	0.09	1.77	0.04	1.01	0.10	1.15	8.20	1930	
	1931	4,250	5,537,000	46.4	66.5	36.6	4.79	4.89	0.09	2.04	0.07	1.28	0.55	1.90	8.92	1931	
54	1930	4,200	22,910,000	94.0	99.9	98.9	2.53	2.16	0.55	1.18	1.16	--	--	1.97	5.86	1930	54
	1931	4,200	22,518,000	92.6	95.6	98.8	1.03	0.91	0.36	1.19	0.38	--	--	1.75	4.21	1931	
130	1930	3,600	4,484,700	41.8	54.1	50.7	4.72	4.59	0.89	1.36	0.77	0.82	0.17	1.76	8.60	1930	130
	1931	3,600	4,168,750	42.4	67.0	31.4	4.19	3.93	0.56	1.58	0.71	1.96	0.18	2.85	8.92	1931	
2	1929	3,450	15,479,500	83.2	94.4	84.5	2.71	2.62	0.33	1.40	0.13	--	--	0.47	4.82	1929	2
	1930	3,450	16,077,095	78.8	94.3	77.1	2.18	1.98	0.45	1.14	0.09	1.56	0.09	1.74	5.31	1930	
	1931	3,450	17,745,325	86.5	97.8	89.8	2.11	2.00	0.26	1.09	0.08	1.16	0.09	1.33	4.68	1931	
164	1930	3,360	14,709,450	81.4	90.7	72.7	3.66	2.91	0.31	1.38	0.16	1.12	0.19	1.47	6.07	1930	164
	1931	3,360	11,037,481	59.5	76.3	76.9	3.34	2.66	0.28	1.73	0.35	1.52	0.30	2.17	6.84	1931	
81	1930	3,250	No Reply To Inquiries For 1931														81
3	1930	3,125	No Reply To Inquiries For 1931														3
720	1930	3,080	3,226,050	34.7	52.5	35.7	3.99	5.21	0.94	1.37	0.37	0.26	0.35	0.98	8.50	1930	720
	1931	3,080	3,124,300	34.1	--	--	3.07	4.42	1.10	1.43	0.22	0.05	0.35	0.62	7.57	1931	
109	1930	1,800	885,399	--	48.1	16.3	5.53	5.33	1.26	3.78	0.88	0.60	0.11	1.59	12.96	1930	109
	1931	2,925	1,858,860	--	54.1	30.3	5.60	6.29	1.25	2.32	0.27	0.80	0	1.07	10.93	1931	
7	1929	1,830	3,231,938	38.0	64.8	53.2	5.77	5.31	0.36	3.06	1.34	--	--	3.98	12.70	1929	7
	1930	2,760	3,211,821	38.0	61.8	53.1	4.79	4.52	0.35	3.17	1.19	1.57	0.31	3.07	11.11	1930	
	1931	2,760	3,068,685	43.9	57.3	36.4	3.43	3.09	0.39	3.79	0.68	1.28	0.51	2.47	9.73	1931	
732	1930	2,520	4,848,480	37.4	59.7	61.1	3.65	3.73	0.91	1.61	--	--	--	2.08	8.33	1930	732
	1931	2,520	2,632,325	25.0	49.1	44.1	2.82	3.68	1.14	2.19	0.59	1.33	0.77	2.69	9.70	1931	
60	1930	2,400	No Reply To Inquiries For 1931														60
41	1929	2,370	3,798,610	39.6	--	--	5.51	5.43	0.48	4.11	0.24	0.22	0.07	0.53	10.54	1929	41
	1930	2,370	4,068,290	39.2	59.0	44.2	5.19	5.09	0.48	4.08	0.37	0.63	0.15	1.15	10.80	1930	
	1931	2,370	4,092,730	39.4	67.3	44.2	4.57	4.48	0.51	3.85	0.25	0.24	0.13	0.62	9.46	1931	
723	1930	1,320	1,270,520	19.7	68.0	26.3	5.62	6.46	1.15	4.01	0.39	2.37	0.97	3.73	15.35	1930	723
	1931	2,370	1,355,510	--	73.3	--	4.21	4.23	0.95	3.56	0.53	0.86	0.70	2.09	10.83	1931	
79	1930	2,170	3,184,025	42.6	67.3	36.8	3.33	3.12	0.46	4.13	0.08	0.49	0.31	0.88	8.59	1930	79
	1931	2,170	3,204,390	38.6	66.8	39.6	2.62	2.38	0.30	3.31	0.13	0.20	0.13	0.46	6.45	1931	
5	1929	2,160	319,120	3.7	--	--	4.15	5.45	1.77	11.61	3.56	--	--	17.99	36.81	1929	5
	1930	2,160	1,013,269	8.1	--	--	4.04	4.86	1.25	5.56	1.14	--	--	8.17	19.84	1930	
	1931	2,160	1,789,653	10.0	--	--	3.33	3.86	0.80	2.15	0.79	--	--	2.18	8.99	1931	
80	1930	2,150	2,280,120	41.9	--	--	3.49	3.54	0.55	4.09	0.74	1.18	0	1.92	10.10	1930	80
	1931	2,150	2,463,900	39.9	55.8	36.7	2.41	2.50	0.25	3.07	0.30	0.48	0	0.78	6.60	1931	
6	1929	2,100	1,318,898	10.0	--	--	4.70	4.17	0.72	2.67	0.47	--	--	1.61	9.16	1929	6
	1930	2,100	1,576,234	18.8	--	--	4.51	3.93	0.57	2.07	0.40	--	--	2.03	8.60	1930	
	1931	2,100	1,612,957	11.8	--	--	4.12	3.53	0.62	1.89	0.31	--	--	1.13	7.17	1931	
728	1930	2,040	3,104,140	41.6	66.0	42.4	5.90	6.44	0.73	2.62	0.84	1.09	0.10	2.03	11.82	1930	728
	1931	2,040	3,275,810	39.1	67.1	44.8	5.20	5.30	0.72	2.01	0.60	0.39	0.34	1.33	9.36	1931	

TABLE II - COMPARATIVE COSTS - 1929, 1930 and 1931 REPORTS (Page 2)

PLANT NUMBER	YEAR	TOTAL INSTALLED B.H.P.	TOTAL NET OUTPUT - K.W. HRS.	ANNUAL PLANT LOAD FACTOR	RUNNING PLANT CAPACITY FACTOR	PLANT SERVICE FACTOR	AVERAGE COST OF FUEL OIL-CENTS PER GALLON	COSTS PER NET K.W. HR. - MILLS: FUEL COST	LUBRICATING OIL COST	ATTENDANCE & SUPERINTENDANCE COST	COST OF SUPPLIES & MISC., INCLUDING WATER	COST OF ENGINE REPAIRS	COST OF ALL OTHER PLANT REPAIRS	TOTAL OF ALL SUPPLY, REPAIR, & MISC. COSTS	TOTAL PRODUCTION COST	YEAR	PLANT NUMBER
716	1930	1,965	Not Operated In 1931 - Territory Served By High Line														716
93	1929	1,960	2,836,950	27.4	44.8	59.4	4.10	5.25	0.52	2.50	0.15	0.38	0.09	0.62	8.88	1929	93
	1930	1,960	2,659,910	30.9	48.4	58.3	4.09	5.35	0.39	2.55	0.16	0.51	0.02	0.69	9.08	1930	
	1931	1,960	3,403,360	29.8	52.3	60.6	2.79	3.36	0.38	2.14	0.13	--	--	0.44	6.32	1931	
215	1930	1,960	1,372,941	--	78.5	17.5	5.40	4.68	0.58	1.12	0.30	0.01	0.10	0.44	6.82	1930	215
	1931	1,960	1,568,700	13.1	89.9	15.5	3.97	3.45	0.21	1.40	0.38	0.53	0.08	0.99	6.05	1931	
729	1930	1,925	Incomplete Data Furnished for 1931														
837	1930	1,875	2,016,465	33.2	49.0	38.8	3.07	3.80	0.68	7.39	0.66	3.30	1.12	5.10	16.97	1930	837
	1931	1,875	2,101,374	35.9	51.1	40.2	2.72	3.43	0.65	6.01	0.49	3.36	0.92	4.77	14.86	1931	
176	1930	1,800	Foreign Installation - Data Not Solicited for 1931														
722	1930	1,705	796,450	9.3	82.1	11.2	5.61	5.97	0.69	7.99	1.49	1.19	0.90	3.58	18.23	1930	722
	1931	1,705	828,340	10.1	72.8	13.2	4.26	4.63	0.56	8.48	1.03	0.14	0.44	1.61	15.28	1931	
18	1929	1,650	2,351,123	26.4	--	--	4.71	4.52	0.41	2.58	0.38	--	--	2.30	9.81	1929	18
	1930	1,650	2,406,690	27.2	--	--	4.49	4.51	0.61	2.08	0.21	--	--	4.80	12.00	1930	
	1931	1,650	2,406,609	26.0	--	--	3.94	3.93	0.58	1.98	0.41	--	--	2.24	8.73	1931	
8	1930	1,560	2,049,000	34.2	57.0	39.1	3.37	3.40	0.58	4.68	0.70	0.27	0	0.97	9.63	1930	8
	1931	1,560	1,876,410	33.4	58.7	35.3	2.81	2.92	0.41	3.61	--	--	--	0.54	7.48	1931	
10	1929	1,500	3,154,085	91.2	74.7	48.9	2.97	3.00	0.97	3.28	0.19	--	--	0.46	7.71	1929	10
	1930	1,500	3,058,452	--	72.1	48.5	2.42	2.33	0.78	3.09	0.07	0.78	0.42	1.27	7.47	1930	
	1931	1,500	3,000,700	78.5	69.9	49.0	2.45	2.27	0.66	5.38	0.13	0.84	0.52	1.49	9.80	1931	
132	1930	1,500	1,446,160	33.9	63.5	29.9	4.64	4.98	0.32	4.51	0.34	0	0	0.34	10.15	1930	132
	1931	1,500	1,655,300	38.5	69.1	31.3	3.63	3.68	0.26	4.30	0.34	0	0	0.34	8.58	1931	
865	1930	1,450	3,544,230	49.1	--	--	1.94	1.93	0.53	2.04	1.00	--	--	3.55	8.05	1930	865
	1931	1,450	364,360	5.2	--	--	1.53	1.53	0.90	19.33	2.95	--	--	10.36	32.12	1931	
70	1930	1,420	1,980,400	--	--	--	3.68	3.81	0.44	3.84	0.13	0.12	0.05	0.30	8.39	1930	70
	1931	1,420	1,868,725	41.7	62.0	39.4	2.83	3.25	0.46	3.48	0.32	0.85	0.38	1.55	8.74	1931	
154	1930	1,400	1,571,594	26.8	--	--	4.48	3.79	0.32	3.95	0.35	--	--	0.71	8.77	1930	154
	1931	1,400	1,606,939	29.3	--	--	3.59	2.94	0.22	4.22	0.44	--	--	1.58	8.96	1931	
163	1930	1,400	3,125,835	60.4	71.4	82.1	2.44	2.13	0.21	1.12	0.29	--	--	3.44	6.90	1930	163
	1931	1,400	3,793,300	43.6	76.3	63.7	1.92	1.75	0.16	1.60	0.21	--	--	2.16	5.67	1931	
11	1929	1,370	1,061,480	24.5	97.1	14.0	2.38	2.12	0.64	4.36	0.22	--	--	0.84	7.96	1929	11
	1930	1,080	954,080	22.1	90.7	13.5	2.09	1.82	0.49	3.65	0.16	1.88	0.83	2.87	8.83	1930	
	1931	1,080	1,328,740	30.5	95.9	17.5	1.30	1.59	0.42	5.05	0.14	3.02	0.38	3.54	10.60	1931	
12	1930	1,350	Foreign Installation - Data Not Solicited for 1931														
725	1930	1,310	1,976,000	--	--	--	4.40	4.39	0.84	2.58	0.38	0.46	0.20	1.04	8.85	1930	725
	1931	1,310	2,840,671	--	--	--	3.35	3.15	0.58	1.78	0.40	0.21	0.15	0.76	6.28	1931	
13	1929	1,225	1,169,971	33.8	55.9	34.1	6.00	7.44	0.57	4.74	1.00	0.89	0.77	2.66	15.41	1929	13
	1930	1,225	1,131,661	32.7	59.2	31.2	5.69	6.30	0.65	4.88	0.54	0.39	0.08	1.01	13.14	1930	
	1931	1,225	1,259,958	38.4	58.3	34.1	5.41	6.29	0.73	4.69	0.24	0.31	0.05	0.60	12.51	1931	
17	1929	1,200	1,611,876	20.7	--	--	4.52	4.69	0.53	2.64	0.67	--	--	2.11	9.96	1929	17
	1930	1,200	2,148,910	27.5	--	--	4.45	4.23	0.50	2.06	0.36	--	--	1.96	8.75	1930	
	1931	1,200	2,207,300	27.2	--	--	3.97	3.83	0.40	1.89	0.42	--	--	1.89	7.94	1931	
94	1929	1,200	2,982,340	39.6	67.0	64.9	3.17	2.83	0.55	1.93	0.36	0.25	0.19	0.80	6.11	1929	94
	1930	1,200	3,315,550	39.9	66.4	71.1	3.12	2.76	0.59	1.69	0.30	0.51	0.05	0.86	5.90	1930	
	1931	1,200	717,642	9.0	38.4	16.1	2.26	2.04	0.68	3.57	0.41	--	--	1.06	7.35	1931	
616	1930	1,150	1,553,500	37.2	--	--	5.10	4.83	0.85	4.81	--	--	--	0.39	10.88	1930	616
	1931	1,150	1,609,550	38.8	--	--	4.50	4.35	0.76	4.32	--	2.48	--	2.79	12.22	1931	
95	1929	1,120	1,675,810	26.0	48.0	57.8	3.23	3.76	0.32	3.60	0.32	0.36	0.15	0.83	8.51	1929	95
	1930	1,120	1,408,900	23.3	50.7	45.4	3.12	3.83	0.21	3.54	0.38	0.68	0.02	1.08	8.66	1930	
	1931	1,120	869,940	21.4	51.9	27.4	2.36	2.91	0.23	4.05	0.42	--	--	1.14	8.33	1931	
96	1929	1,100	1,133,027	17.0	49.1	37.7	3.81	4.30	0.99	3.87	0.54	4.39	0.19	5.12	14.26	1929	96
	1930	1,100	1,215,181	24.3	61.5	32.2	3.55	3.54	0.66	4.26	0.43	0.78	0.21	1.42	9.88	1930	
	1931	1,100	967,500	19.7	50.9	31.5	2.74	2.95	0.45	5.03	0.20	--	--	1.47	9.90	1931	
182	1930	1,090	No Reply to Inquiries for 1931														
721	1930	1,080	749,476	23.1	32.9	39.0	6.52	11.48	1.60	4.61	1.05	4.16	1.69	6.90	24.59	1930	721
	1931	1,080	716,290	29.4	--	--	3.22	6.22	2.40	4.25	0.79	7.36	0.69	8.84	21.71	1931	

TABLE II - COMPARATIVE COSTS - 1929, 1930 and 1931 REPORTS (Page 3)

PLANT NUMBER	YEAR	TOTAL INSTALLED B.H.P.	TOTAL NET OUTPUT - K.W. HRS.	ANNUAL PLANT LOAD FACTOR	RUNNING PLANT CAPACITY FACTOR	PLANT SERVICE FACTOR	AVERAGE COST OF FUEL OIL-CENTS PER GALLON	COSTS PER NET K.W. HR. - MILLS								YEAR	PLANT NUMBER
								FUEL COST	LUBRICATING OIL COST	ATTENDANCE & SUPERINTENDANCE COST	COST OF SUPPLIES & MISC., INCLUDING WATER	COST OF ENGINE REPAIRS	COST OF ALL OTHER PLANT REPAIRS	TOTAL OF ALL SUPPLY, REPAIR, & MISC. COSTS	TOTAL PRODUCTION COST		
78	1928/30	1,060	4,867,305	--	55.5	88.7	5.08	4.92	0.82	2.79	0.08	0.61	0.13	0.82	9.35	1928/30	78
	1931	1,060	1,906,347	31.4	46.5	69.2	5.07	5.88	1.13	4.69	0.23	0.15	0.19	0.57	12.27	1931	
97	1930	1,020	Not Operated in 1931 - Territory Served by High Line													1931	97
83	1930	1,000	1,512,520	33.6	51.6	53.1	6.75	6.37	0.46	7.34	0.54	1.00	0	1.54	15.71	1930	83
	1931	1,000	1,720,560	33.5	54.4	55.3	6.00	5.71	0.31	6.46	0.81	1.05	0	1.86	14.34	1931	
98	1929	1,000	1,677,817	--	63.2	--	5.57	7.31	1.05	3.05	0.45	1.68	0.21	2.34	13.75	1929	98
	1930	1,000	1,774,521	39.9	64.5	56.7	4.73	6.37	0.90	3.28	0.60	1.00	0.31	1.91	12.46	1930	
	1931	1,000	1,352,480	35.4	65.8	38.0	3.44	4.85	0.50	3.67	0.64	--	--	2.42	11.44	1931	
801	1930	1,000	4,110,052	74.3	74.6	94.9	3.82	4.19	0.43	1.25	0.79	0.89	0.17	1.95	7.72	1930	801
	1931	1,000	4,616,205	78.0	84.9	95.8	3.31	3.28	0.30	1.04	1.20	0.28	0.02	1.50	6.12	1931	
856	1930	1,000	No Reply to Inquiries for 1931														856
1061	1930	980	1,405,473	--	96.7	49.4	4.32	3.22	0.56	0.68	0.10	--	--	0.55	5.01	1930	1061
	1931	980	1,974,459	33.4	81.4	43.7	3.98	3.59	0.56	0.92	0.18	0.63	0.10	0.91	5.98	1931	
863	1930	975	2,136,540	41.0	69.4	52.7	3.55	3.29	0.91	1.76	0.46	2.32	0.06	2.84	8.80	1930	863
	1931	975	2,780,400	59.3	82.7	60.3	3.50	3.13	0.66	1.98	0.19	0.55	0.06	0.80	6.57	1931	
69	1930	960	No Reply to Inquiries for 1931													1930	69
53	1930	900	2,300,000	50.4	70.6	63.7	3.26	3.68	0.73	2.12	0.43	1.76	0.05	2.24	8.77	1930	53
	1931	900	2,355,470	52.8	71.0	62.5	2.65	3.06	0.53	2.00	0.42	2.94	0.05	3.41	9.00	1931	
74	1930	885	1,010,212	--	--	--	4.53	5.32	0.80	3.98	0.29	--	--	1.23	11.33	1930	74
	1931	885	1,172,892	28.7	65.7	34.8	3.59	3.12	0.30	5.97	0.22	0.07	0.30	0.59	9.98	1931	
189	1930	850	800,000	32.4	27.4	59.5	4.10	5.22	0.93	8.63	1.03	--	--	2.53	17.31	1930	189
	1931	850	951,426	38.4	31.5	64.0	3.68	4.40	0.36	5.99	0.78	--	--	2.44	13.19	1931	
153	1930	840	49,340	2.2	--	--	4.08	7.70	1.01	44.30	11.92	--	--	24.43	77.44	1930	153
	1931	840	557,470	12.1	--	--	3.41	3.46	0.61	4.71	0.90	--	--	1.83	10.61	1931	
1056	1930	840	2,122,057	46.5	92.5	50.2	4.10	4.13	0.56	1.61	0.15	--	--	0.37	6.67	1930	1056
	1931	840	1,906,182	41.5	76.2	55.0	3.84	3.70	0.51	1.56	0.50	0.66	0.28	1.44	7.21	1931	
731	1930	825	454,041	9.2	49.2	18.7	5.75	5.96	1.84	9.43	0.37	1.56	0	1.93	19.16	1930	731
	1931	825	592,895	11.8	61.2	20.8	4.84	4.91	1.00	6.58	0.23	0.59	0	0.82	13.31	1931	
100	1929	740	1,247,150	49.7	59.9	50.2	3.36	4.14	0.73	4.69	0.41	0.37	0.20	0.98	10.53	1929	100
	1930	740	1,459,540	35.1	60.9	58.1	3.50	4.32	0.68	3.70	0.47	0.82	0.06	1.35	10.05	1930	
	1931	740	498,640	19.1	66.0	18.6	2.78	3.31	0.61	6.30	0.69	--	--	0.81	11.03	1931	
112	1930	740	1,165,300	29.7	68.1	42.2	4.43	5.19	0.83	5.46	0.19	0.54	0.12	0.85	12.33	1930	112
	1931	740	1,195,128	34.8	71.3	41.8	3.42	3.47	0.74	5.38	0.18	0.35	0.25	0.78	10.37	1931	
67	1929	720	1,163,600	64.0	79.1	33.7	5.94	6.03	0.55	1.60	0.21	0.04	0	0.25	8.43	1929	67
	1930	720	1,098,900	58.4	82.9	30.3	4.97	4.73	0.71	1.70	0.23	1.03	0.02	1.28	8.42	1930	
	1931	720	891,200	48.5	82.3	25.6	4.58	4.38	0.87	2.09	0.24	1.05	0.10	1.39	8.73	1931	
170	1930	720	860,110	19.5	64.7	32.7	6.51	7.28	0.89	2.50	0.12	1.30	0	1.42	12.09	1930	170
	1931	720	835,219	22.4	64.5	31.6	6.20	6.73	0.78	2.19	0.20	2.64	0	2.84	12.59	1931	
527	1930	720	673,300	--	37.0	56.2	3.87	5.02	1.59	3.97	0.25	0.04	0.04	0.33	10.91	1930	527
	1931	720	933,200	31.9	41.4	60.5	2.83	3.35	0.82	3.48	0.29	0.13	0.01	0.43	8.08	1931	
718	1930	720	604,312	--	36.9	51.5	4.27	6.05	1.33	6.50	0.83	1.70	0.58	3.11	16.99	1930	718
	1931	720	691,477	38.6	34.9	51.4	3.45	4.66	1.33	5.04	0.68	6.28	0.14	7.10	18.13	1931	
19	1929	700	603,260	29.8	--	--	4.50	6.32	1.40	5.40	1.07	--	--	2.24	15.36	1929	19
	1930	700	710,280	34.9	--	--	4.13	5.72	1.26	4.96	1.07	--	--	4.05	15.99	1930	
	1931	700	694,060	34.8	--	--	3.73	5.27	1.47	5.15	0.80	--	--	5.59	17.48	1931	
61	1930	600	1,539,900	--	52.3	79.6	6.39	5.31	0.59	5.53	0.33	0.61	0.06	1.00	12.43	1930	61
	1931	600	1,555,200	38.7	68.4	64.7	4.20	3.46	0.36	5.37	0.11	1.07	0	1.18	10.37	1931	
106	1929	600	750,419	--	--	--	4.66	4.75	0.66	2.66	0.61	--	--	2.62	10.69	1929	106
	1930	600	1,435,434	35.6	--	--	4.49	4.18	0.27	3.09	0.65	--	--	2.52	10.06	1930	
	1931	600	1,504,859	28.1	--	--	3.97	3.71	0.24	3.05	0.68	--	--	1.76	8.76	1931	

TABLE II - COMPARATIVE COSTS - 1929, 1930 and 1931 REPORTS (Page 4)

PLANT NUMBER	YEAR	TOTAL INSTALLED B.H.P.	TOTAL NET OUTPUT - K.W. HRS.	ANNUAL PLANT LOAD FACTOR	RUNNING PLANT CAPACITY FACTOR	PLANT SERVICE FACTOR	AVERAGE COST OF FUEL OIL - CENTS PER GALLON	COSTS PER NET K.W. HR. - MILLS								YEAR	PLANT NUMBER
								FUEL COST	LUBRICATING OIL COST	ATTENDANCE & SUPERINTENDANCE COST	COST OF SUPPLIES & MISC., INCLUDING WATER	COST OF ENGINE REPAIRS	COST OF ALL OTHER PLANT REPAIRS	TOTAL OF ALL SUPPLY, REPAIR, & MISC. COSTS	TOTAL PRODUCTION COST		
103	1929	390	772,985	40.8	53.3	67.8	4.63	6.04	0.99	5.82	0.56	0.85	0.28	1.69	14.54	1929	103
	1930	590	995,145	38.0	60.2	61.2	4.20	5.43	0.93	4.60	0.49	1.73	0.43	2.65	13.61	1930	
	1931	590	1,015,336	34.6	58.3	52.4	3.31	4.63	0.67	4.24	0.69	--	--	2.03	11.57	1931	
862	1930	588	627,820	25.3	47.7	39.7	4.43	4.62	0.83	8.84	0.42	1.63	1.88	3.93	18.22	1930	862
	1931	588	640,360	25.7	46.9	41.7	3.27	3.81	0.70	10.77	0.40	1.24	0.97	2.61	17.89	1931	
102	1929	480	577,920	24.7	39.8	58.1	3.41	5.22	1.61	8.51	0.57	0.56	0.30	1.43	16.77	1929	102
	1930	580	591,620	27.4	42.7	55.1	3.25	4.78	1.36	8.32	0.34	0.28	0.35	0.97	15.43	1930	
	1931	580	592,210	26.0	50.1	38.8	2.49	3.75	1.06	7.63	0.21	--	--	0.67	13.11	1931	
101	1929	540	666,871	36.5	53.7	40.3	3.56	4.85	1.43	6.39	0.83	2.67	0.57	4.07	16.74	1929	101
	1930	540	768,521	49.1	60.5	40.6	3.79	4.56	0.90	5.62	0.35	0.74	0.03	1.12	12.20	1930	
	1931	540	706,879	48.3	55.7	43.0	2.34	3.11	1.01	4.34	0.34	--	--	1.03	9.49	1931	
246	1930	540	364,500	28.8	--	--	6.47	11.42	2.69	9.80	0.69	0.14	0	0.83	24.74	1930	246
	1931	540	407,085	33.6	38.0	35.4	6.29	9.15	2.19	8.78	0.41	0.35	0	0.76	20.88	1931	
501	1930	480	403,022	32.5	40.6	37.5	5.50	7.58	1.80	10.58	0.40	0.14	0.31	0.85	20.81	1930	501
	1931	480	452,970	37.3	39.3	42.6	3.93	5.53	1.60	8.50	0.49	0.62	0.33	1.44	17.07	1931	
335	1930	430	432,940	38.5	38.2	57.0	4.85	5.45	0.57	11.01	1.25	0.19	0.07	1.51	18.54	1930	335
	1931	430	489,446	39.1	37.2	53.6	3.82	4.03	0.64	9.74	4.06	0.40	0.05	4.51	18.92	1931	
24	1929	400	563,300	45.9	44.9	58.4	4.00	5.27	1.01	12.14	1.10	2.39	0.07	3.56	21.98	1929	24
	1930	400	No Report Made for 1930													1930	
	1931	400	548,700	43.6	47.1	51.9	2.52	3.10	1.24	7.74	0.66	0.27	0	0.93	13.04	1931	
194	1930	400	897,210	41.4	52.3	77.9	3.73	4.03	0.75	7.93	0.19	0.04	0	0.23	12.94	1930	194
	1931	400	355,490	23.3	34.7	47.0	3.26	3.93	0.41	6.34	0.02	0	0	0.02	10.70	1931	
99	1930	400	Not Operated in 1931 Territory Served by High Line													1930	99
247	1930	390	428,290	28.9	--	--	6.00	8.12	1.10	10.79	0.11	0	0	0.11	20.12	1930	247
	1931	390	480,204	25.1	44.6	47.1	4.62	6.03	1.01	9.62	0.05	0.40	0.26	0.71	17.37	1931	
353	1930	390	No Reply to Inquiries for 1931													1930	353
105	1929	370	272,138	--	51.7	--	3.20	5.97	3.69	14.60	1.76	6.59	0.90	9.25	33.51	1929	105
	1930	370	372,330	38.4	56.5	35.7	3.14	4.77	1.09	11.00	0.97	0.44	0.17	1.58	18.44	1930	
	1931	370	349,820	33.1	47.4	39.2	2.67	4.25	1.01	8.66	0.38	--	--	1.35	15.27	1931	
152	1930	360	25,948	2.2	--	--	5.35	6.55	0.69	81.00	12.28	--	--	22.28	110.52	1930	
	1931	360	52,080	4.2	--	--	4.75	5.47	0.79	32.57	4.80	--	--	12.21	51.04	1931	
857	1930	330	No Reply to Inquiries for 1931													1930	857
265	1930	180	184,000	--	--	--	9.14	13.48	1.76	13.58	1.10	0.71	0	1.81	30.63	1930	265
	1931	320	216,000	--	--	37.0	7.79	11.81	1.83	11.57	0.32	0.46	0.58	1.36	26.57	1931	
27	1930	315	190,375	30.1	--	--	4.93	9.00	2.42	12.68	0.77	--	--	6.68	30.78	1930	27
	1931	315	208,890	30.8	--	--	4.06	7.77	1.52	10.58	0.95	--	--	7.29	27.16	1931	
515	1930	305	No Reply to Inquiries for 1931													1930	515
169	1930	240	151,850	11.3	41.5	27.1	5.98	6.23	0.94	2.56	0.55	0.51	0	1.06	10.79	1930	169
	1931	240	184,360	13.7	42.5	31.9	5.56	6.23	0.89	2.11	0.87	4.39	0	5.26	14.49	1931	
858	1930	180	No Reply to Inquiries for 1931													1930	858
733	1930	175	14,800	1.6	65.0	1.6	4.82	5.54	2.91	8.92	2.91	0	0	2.91	20.28	1930	733
	1931	175	3,293	0.3	62.5	0.5	5.78	10.93	0.91	10.93	9.72	0	0	9.72	32.49	1931	
646	1930	120	15,893	3.8	13.7	7.0	7.00	12.65	3.15	147.25	11.26	0	0	11.26	174.31	1930	646
	1931	120	21.038	5.3	16.1	20.4	7.00	16.70	2.57	16.97	3.28	0	0	3.28	39.52	1931	

TABLE III - INFORMATION COVERING LUBRICATING OIL (Page 1)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM - SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	TRUNK PISTON OR CROSS HEAD	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING - K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	RATED HORSEPOWER HOURS IN REPORTED PERIOD	GALLONS OF NEW LUBRICATING OIL USED	GALLONS OF UNFIT LUBRICATING OIL DISCARDED	RATED H.P. HRS. PER GALLON OF NEW LUBRICATING OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	GROSS OUTPUT - K.W. HRS.	GROSS K.W. HRS. PER GALLON OF NEW LUBRICATING OIL	LUBRICATING OIL TREATMENT (SEE NOTES)	AVERAGE COST OF LUBRICATING OIL - CENTS PER GALLON
43	1	4	A	-	TP	500	336	3	22	29.5	150	375	1915	813	406,500	639	---	636	69.3		189,480	296		
	2	4	A	-	TP	500	336	3	22	29.5	150	375	1916	901	450,500	557	---	609	68.4		206,928	372		
	3	4	A	-	TP	520	349	4	19	24.5	200	500	1919	1,775	923,000	1,008	---	915	76.6		474,370	471	SC &	
	4	4	A	-	TP	520	349	4	19	24.5	200	500	1920	1,543	646,360	679	---	952	76.8		333,150	491	CC &	
	5	4	A	-	CH	1,000	671	6	20.5	36	150	925	1922	2,273	2,273,000	1,771	---	1,283	63.7		972,250	549	C	
	6	4	A	-	CH	1,000	671	6	20.5	36	150	925	1925	2,383	2,383,000	1,874	---	1,271	67.3		1,077,270	575		
	7	2	A	S	CH	3,750	2,517	6	30	42	124	3,750	1928	7,651	28,681,250	8,641	---	3,320	86.8		12,862,700	1,488		
	8	2	A	S	CH	3,750	2,517	6	30	42	124	3,750	1929	8,156	30,585,000	10,730	---	2,850	80.5		16,518,400*	1,5399		
		Plant				11,840	7,746					11,100			66,348,610	25,899	---	2,560		73.2	32,634,548*	1,260*		33.0
46	1	2	A	S	CH	1,250	840	5	20.75	26	180	850	1922	1,650	2,062,500	---	---	---	72.0		997,760	---		
	2	2	A	S	CH	1,250	840	5	20.75	26	180	850	1922	2,635	3,295,000	---	---	---	70.3		1,556,550	---		
	3	2	A	S	CH	2,250	1,510	4	28	44	124	2,000	1922	4,398	9,895,500	---	---	---	78.1		5,182,300	---	SC &	
	4	2	A	S	CH	2,250	1,510	4	28	44	124	2,000	1922	4,094	9,211,500	---	---	---	79.8		4,931,900	---	S &	
	5	2	A	S	CH	2,250	1,510	4	28	44	124	2,000	1930	5,912	13,302,000	---	---	---	81.4		7,259,400	---	F	
		Plant				9,250	6,210					7,700			37,766,500	11,616	---	3,250		78.6	19,927,910	1,715		35.9
168	1	2	A	S	CH	1,250	840	5	20.75	26	180	850	1918	3,825	4,781,250	---	---	---	76.5		2,456,000	---		
	2	2	A	S	CH	1,250	840	5	20.75	26	180	850	1918	4,026	5,035,000	---	---	---	73.5		2,554,800	---		
	3	2	A	S	CH	1,250	840	5	20.75	26	180	850	1919	4,819	6,023,750	---	---	---	78.5		3,179,600	---	SC &	
	4	2	A	S	CH	1,250	840	5	20.75	26	180	850	1916	3,995	4,993,750	---	---	---	77.5		2,502,100	---	S &	
	5	2	A	S	CH	1,250	840	5	20.75	26	180	850	1918	6,338	7,922,500	---	---	---	73.1		4,114,900	---	F	
	6	2	A	S	CH	2,250	1,510	4	28	44	124	2,000	1929	6,003	13,506,750	---	---	---	97.1		8,796,000	---		
		Plant				8,500	5,710					6,250			42,263,000	10,772	---	3,925		83.5	23,703,300	2,200		57.0
82	1	4	A	-	TP	360	242	4	16.25	21	225	312	1921	963	346,680	---	0	---	65.9		140,260	---		
	2	2	A	S	CH	750	504	4	17	27	180	740	1923	1,410	1,057,500	---	0	---	57.5		408,930	---		
	3	2	A	S	CH	750	504	4	17	27	180	740	1924	999	749,250	---	0	---	58.3		293,335	---		
	4	2	A	S	CH	1,150	772	6	17	27	180	1,000	1927	3,077	3,538,550	---	0	---	68.5		1,326,800	---	BC	
	5	2	A	S	CH	1,150	772	6	17	27	180	1,000	1927	3,489	4,012,350	---	0	---	67.2		1,810,100	---		
	6	2	A	S	CH	1,500	1,007	8	17	27	180	1,350	1929	6,466	9,699,000	---	0	---	72.2		4,701,300	---		
		Plant				5,660	3,801					5,142			19,403,330	5,842	0	3,320		69.0	8,989,025	1,538		58.3
91	1	4	M	-	TP	1,125	755	6	22	30	180	---	1927	4,859	5,466,375	---	---	---	58.0		2,185,400	---		
	2	4	M	-	TP	1,125	755	6	22	30	180	---	1928	6,826	7,679,250	---	---	---	58.4		3,013,700	---		
	3	4	M	-	TP	600	403	6	17	24	225	---	1927	3,705	2,223,000	---	---	---	57.6		859,900	---	CC	
	4	4	M	-	TP	600	403	6	17	24	225	---	1927	3,641	2,184,600	---	---	---	57.0		835,000	---		
	5	4	A	-	TP	1,800	1,208	6	24	32	225	---	1930	1,743	3,137,400	---	---	---	65.8		1,406,200	---		
		Plant				5,250	3,524					---			20,690,625	3,695	---	5,600		59.3	8,241,000	2,230		55.9
73	1	2	A	S	CH	1,250	840	5	20.75	26	180	1,063	1923	5,572	6,965,000	1,406	0	4,955	64.0		2,996,600	2,130		
	2	2	A	S	CH	1,250	840	5	20.75	26	180	1,063	1924	6,345	7,931,250	1,392	0	5,695	66.9		3,566,800	2,560	CC &	
	3	2	A	S	CH	1,250	840	5	20.75	26	180	1,063	1928	3,167	3,958,750	910	0	4,350	69.8		1,855,500	2,040	F	
	4	2	A	S	CH	1,250	840	5	20.75	26	180	1,063	1928	736	920,000	241	0	3,815	64.0		395,800	1,640		
		Plant				5,000	3,360					4,252			19,775,000	3,949	0	5,010		66.3	8,815,500	2,230		55.0
92	1	2	A	S	CH	1,250	840	5	20.75	26	180	---	1928	3,252	4,065,000	---	---	---	65.2		1,780,700	---	BC &	
	2	2	A	S	CH	1,250	840	5	20.75	26	180	---	1928	2,929	3,661,250	---	---	---	71.1		1,747,900	---	S	
	3	2	A	S	CH	2,400	1,612	8	21	24	225	---	1930	6,291	15,098,400	---	---	---	61.2		6,202,800	---		
		Plant				4,900	3,292					---			22,824,650	8,025	---	2,845		63.5	9,731,500	1,213		41.7
129	1	2	A	S	CH	1,250	840	5	20.75	26	180	850	1918	5,178	6,472,500	---	---	---	67.1		2,918,125	---	CC &	
	2	2	A	S	CH	1,250	840	5	20.75	26	180	850	1921	5,204	6,505,000	---	---	---	69.2		3,023,725	---	S & F	
	3	2	A	S	CH	2,250	1,510	4	28	44	124	2,000	1930	6,081	13,682,250	---	---	---	85.6		7,856,250	---		
		Plant				4,750	3,190					3,700			26,659,750	8,207	---	3,245		77.1	13,808,100	1,682		39.1
52	1	4	A	-	CH	630	557	4	23	32	164	700	1928	2,143	1,778,690	464	0	3,830	69.8		833,400	1,795		
	2	4	A	-	CH	1,250	840	6	23	32	164	1,060	1928	3,890	4,862,500	1,090	0	4,460	80.1		2,618,300	2,400	SC	
	3	4	A	-	CH	1,250	840	6	23	32	164	1,060	1928	3,943	4,928,750	1,123	0	4,390	80.2		2,555,800	2,365	& S	
	4	4	A	-	CH	1,250	840	6	23	32	164	1,060	1930	3,749	4,686,250	1,002	0	4,680	78.3		2,465,200	2,460		
		Plant				4,580	3,077					3,880			16,256,190	3,679	0	4,420		73.5	8,572,800	2,328		38.3
45	1	4	A	-	TP	600	403	6	16.5	24	200	500	1923	843	505,800	157	55	3,220	42.5		153,575	979		
	2	4	A	-	TP	600	403	6	16.5	24	200	500	1923	777	466,200	161	25	2,895	47.4		148,500	923	SC &	
	3	4	A	-	CH	750	504	4	23	32	150	625	1923	440	333,000	26	34	12,690	60.5		134,275	5,165	S &	
	4	4	A	-	CH	1,150	772	6	23	32	150	1,000	1926	5,326	6,124,900	366	0	16,730	38.7		2,722,100	7,710	C & F	
	5	4	A	-	CH	1,150	772	6	23	32	150	1,000	1926	5,394	6,203,100	369	0	16,800	68.3		2,841,500	7,700		
		Plant				4,250	2,854					3,625			13,630,000	1,079	114	12,650		86.5	6,099,550	5,345		45.3
54	1	4	A	-	TP	600	403	6	17	25	200	450		---	----	---	---	---	--		----	---		
	2	4	A	-	TP	600	403	6	17	25	200	450		---	----	---	---	---	--		----	---		
	3	4	A	-	TP	600	403	6	17	25	200	450		---	----	---	---	---	--		----	---		
	4	4	A	-	TP	600	403	6	17	25	200	450	1925	---	----	---	---	---	--		----	---	CC	
	5	4	A	-	TP	600	403	6	17	25	200	450	-25	---	----	---	---	---	--		----	---		
	6	4	A	-	TP	600	403	6	17	25	200	450		---	----	---	---	---	--		----	---		
	7	4	A	-	TP	600	403	6	17	25	200	450		---	----	---	---	---	--		----	---		
		Plant				4,200	2,821					3,150		60,548	36,328,800	25,553	---	1,440		95.6	23,323,000	914		32.0
864	1	4	A	-	TP	500	336	4	18.875	28.375	164	437	1915	---	----	---	---	---	--		863,310	---		
	2	4	A	-	TP	500	336	4	18.875	28.375	164	437	1915	---	----	---	---	---	--		784,270	---		
	3	4	A	-	TP	500	336	4	18.875	28.375	164	437	1915	---	----	---	---	---	--		744,250	---	CC	
	4	2	A	S	CH	1,250	840	5	20.75	26	180	1,125	1928	---	----	---	---	---	--		1,592,530	---	& F	
	5	2	A	S	CH	1,250	840	5	20.75	26	180	1,125	1928	---	----	---	---	---	--		1,653,500	---		
		Plant				4,000	2,688					3,561				4,426	---	---		--	5,438,130	1,227		35.0
130	1	4	A	-	TP	600	403	6	16.25	23	225	513	1924	3,557	2,134,200	628	0	3,400	68.2		978,000	1,556		
	2	4	A	-	TP	600	403	6	17	24	200	513	1926	2,670	1,602,000	965	0	1,660	65.1		700,000	725	BC	
	3	4	A	-	TP	1,200	806	8	20	24	300	1,062	1929	2,294	2,752,800	780	150	3,530	65.9		1,818,700	1,563	& CC	
	4	4	A	-	TP	1,800	806	6	20	24	300	1,062	1930	2,841	3,409,200	1,067	0	3,190	68.0		1,554,500	1,456		
		Plant				3,600	2,418					3,150			9,898,200	3,440	150	2,875		67.0	4,453,200	1,294		38.0
2	1	4	A	-	TP	1,150	772	6	23	29.5	166.7	1,000	1925	8,018	9,220,700	---	0	---	97.7		6,053,930	---		
	2	4	A	-	TP	1,150	772	6	23	29.5	166.7	1,000	1925	7,597	8,736,550	---	0	---	97.5		5,738,380	---	CC	
	3	4	A	-	TP	1,150	772	6	23	29.5	166.7	1,000	1925	7,978	9,174,700	---	0	---	98.0		6,033,930	---		
		Plant				3,450	2,316					3,000			27,131,950	13,910	0	1,950		97.8	17,823,240	1,281		32.8
164	1	4	M	-	TP	840	564	6	19.5	24	214	1,000	1926	6,153	5,168,520	---	---	---	79.9		2,772,800	---		
	2	4	M	-	TP	840	564	6	19.5	24	214	1,000	1926	6,503	5,462,520	---	---	---	75.3		2,762,200	---	SC &	
	3	4	M	-	TP	840	564	6	19.5	24	214	1,000	1926	7,646	6,422,640		---	---	69.9		3,013,400	---	S & F	
	4	4	M	-	TP	840	564	6	19.5	24	214	1,000	1926	6,646	5,582,640		---	---	81.0		3,038,600	---		
		Plant				3,360	2,256					4,000			22,636,320	9,620	100	2,353		76.3	11,589,300	1,205		32.1

TABLE III - INFORMATION COVERING LUBRICATING OIL (Page 2)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM-SEE NOTES	SCAVENGING SYSTEM-SEE NOTES	TRUNK PISTON OR CROSS HEAD	RATED B.H.P.	EQUIVALENT K.W.-90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING - K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	RATED HORSEPOWER HOURS IN REPORTED PERIOD	GALLONS OF NEW LUBRICATING OIL USED	GALLONS OF UNFIT LUBRICATING OIL DISCARDED	RATED H.P. HRS. PER GALLON OF NEW LUBRICATING OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	GROSS OUTPUT - K.W. HRS.	GROSS K.W. HRS. PER GALLON OF NEW LUBRICATING OIL	LUBRICATING OIL TREATMENT (SEE NOTES)	AVERAGE COST OF LUBRICATING OIL - CENTS PER GALLON
720	1	4	M	-	TP	380[b]	255	2	22.5	36	150	250	1922	---	-----	---	---	---	---		-----	---	CC
	2	4	M	-	TP	300	202	3	17.75	22	225	250	1926	---	-----	---	---	---	---		-----	---	
	3	2	M	S	TP	840	564	6	16	20	257	700	1928	---	-----	---	---	---	---		-----	---	
	4	2	M	S	TP	720	484	6	16	20	257	600	1927	---	-----	---	---	---	---		-----	---	
	5	2	M	S	TP	840	564	6	16	20	257	700	1928	---	-----	---	---	---	---		-----	---	
	Plant					3,080	2,069					2,500				8,150	---	---	--	3,280,300	403		42.1
109	1	2	A	S	CH	675	453	4	16.75	22	240	500	1927	3,066	2,069,550	1,240	---	1,668	44.2	614,280	495		
	2	2	A	S	TP	1,125	755	8	14.75	21	257	975	1929	2,159	2,428,875	1,725	---	1,407	58.8	959,500	556	S	
	3	2	A	S	TP	1,125	755	8	14.75	21	257	975	1931[c]	652	733,500	525	---	1,397	66.7	328,560	626		
	Plant					2,925	1,963					2,450			5,231,925	3,490	---	1,498		1,902,340	545		66.5
7	1	4	M	-	TP	360	242	6	11.5	16	277	375	1925	3,669	1,320,840	---	---	---	57.7	512,400	---		
	2	4	M	-	TP	360	242	6	11.5	16	277	375	1925	3,255	1,171,800	---	---	---	54.6	430,100	---		
	3	4	M	-	TP	360	242	6	11.5	16	277	375	1925	3,413	1,228,680	---	---	---	57.1	471,400	---	CC	
	4	4	M	-	TP	750	504	6	16	22	240	750	1927	1,667.5	1,250,625	---	---	---	57.3	481,200	---		
	5	4	M	-	TP	930	624	6	18	22	257	750	1930	4,101.5	3,814,395	---	---	---	58.1	1,487,300	---		
	Plant					2,760	1,954					2,625			8,786,340	2,645	400	3,320		3,382,400	1,278		45.0
732	1	4	A	-	TP	250	168	4	13.75	17.5	257	200	1915	0	0	---	---	---	--	-----	---		
	2	4	A	-	TP	250	168	4	13.75	17.5	257	200	1916	1,925	456,250	---	---	---	--	-----	---		
	3	4	A	-	TP	520	349	4	19	24.5	200	500	1921	3,650	1,898,000	---	---	---	--	-----	---	CC	
	4	4	A	-	CH	750	504	4	23	32	150	625	1923	4,745	3,558,750	---	---	---	--	-----	---		
	5	4	A	-	CH	750	504	4	23	32	150	625	1925	5,110	3,832,500	---	---	---	--	-----	---		
	Plant					2,520	1,693					2,150			9,745,500	5,900	---	1,650		3,212,915	545		50.8
978	1	2	M	S	TP	840	564	6	16	20	257	700	1931	---	-----	---	---	---	--	-----	---		
	2	2	M	S	TP	840	564	6	16	20	257	700	1931	---	-----	---	---	---	--	-----	---	BC	
	3	2	M	S	TP	840	564	6	16	20	257	700	1931	---	-----	---	---	---	--	-----	---		
	Plant					2,520	1,692					2,100			-----	1,942	---	---	--	2,303,700	1,186		54.6
41	1	4	A	-	TP	520	349	3	22	29.5	164	---	1922	4,023	2,091,960	---	---	---	63.3	888,530	---		
	2	4	A	-	TP	700	470	4	22	29.5	164	---	1923	3,765	2,635,500	---	---	---	60.5	1,071,500	---	CC &	
	3	4	A	-	TP	1,150	772	6	23	29.5	164	---	1928	3,862	4,441,300	---	---	---	73.2	2,182,700	---	S & F	
	Plant					2,370	1,591								9,168,760	3,934	---	2,300		4,142,730	1,038		52.4
723	1	2	M	CC	TP	300[d]	202	6	14	17	257	250	1925	460	138,000	---	---	---	79.8	74,100	---		
	2	2	M	CC	TP	300[d]	202	6	14	17	257	250	1925	440	132,000	---	---	---	74.3	66,100	---		
	3	2	M	S	TP	720	484	6	16	20	257	600	1927	1,382	995,040	---	---	---	73.0	488,100	---	BC	
	4	2	M	S	TP	1,080	705	7	16	20	257	910	1931	1,998	2,087,400	---	---	---	72.9	1,021,600	---		
	Plant					2,370	1,593					2,010			3,352,440	2,730	---	1,227		1,649,900	604		54.1
79	1	4	A	-	TP	225	151	3	16	24	164	198	1911	940	211,500	231	---	916	31.7	45,004	195		
	2	4	A	-	TP	225	151	3	16	24	164	198	1912	77	17,325	28	---	619	31.8	3,696	132		
	3	4	A	-	TP	600	403	6	16.25	23	225	512	1929	4,891	2,928,600	751	---	3,900	68.2	1,342,100	1,787	S & F	
	4	4	A	-	TP	520	349	4	19	24.5	200	438	1923	2,805	1,458,600	528	---	2,760	70.3	688,300	1,302		
	5	4	A	-	TP	600	403	6	16.25	23	225	512	1925	4,861	2,916,600	861	---	3,385	66.3	1,299,500	1,510		
	Plant					2,170	1,457					1,858			7,532,625	2,399	578	3,140		3,378,600	1,407		40.0
5	1	2	M	S	TP	720	484	6	16	20	257	---	1926	---	-----	---	---	---	--	-----	---	--	
	2	2	M	S	TP	720	484	6	16	20	257	---	1927	---	-----	---	---	---	--	-----	---	--	
	3	2	M	S	TP	720	484	6	16	20	257	---	1927	---	-----	---	---	---	--	-----	---	--	
	Plant					2,160	1,452					---			-----	2,861	---	---		1,918,770	671		50.1
80	1	4	A	-	TP	365	245	4	16.25	21	225	---	1921	2,105	768,325	---	---	---	--	-----	---		
	2	4	A	-	TP	520	349	4	19	24.5	200	---	1921	2,105	1,094,600	---	---	---	--	-----	---	S & C	
	3	4	A	-	TP	600	403	6	16.25	23	225	---	1929	3,272	1,963,200	---	---	---	--	-----	---		
	4	4	M	-	TP	665	447	4	21	30	180	---	1926	4,625	3,075,625	---	---	---	--	-----	---		
	Plant					2,150	1,444					---			6,901,750	1,531	---	4,505		2,587,100	1,689		40.9
6	1	2	M	CC	TP	300[d]	202	6	14	17	257	---	1926	---	-----	---	---	---	--	-----	---	--	
	2	4	A	-	TP	600	403	6	17	25	200	---	1926	---	-----	---	---	---	--	-----	---	--	
	3	4	A	-	TP	600	403	6	17	25	200	---	1927	---	-----	---	---	---	--	-----	---	--	
	4	4	A	-	TP	600	403	6	17	25	200	---	1928	---	-----	---	---	---	--	-----	---	--	
	Plant					2,100	1,411					---			-----	1,987	---	---	--	1,651,300	831		49.9
68	1	2	M	CC	TP	240	161	4	14	17	257	200	1925	2,635	632,400	---	0	---	54.5	231,300	---		
	2	2	M	CC	TP	360	242	6	14	17	257	300	1925	3,125	1,125,000	---	0	---	58.5	442,800	---	BC	
	3	2	M	S	TP	560	376	4	16	20	257	470	1928	4,428	2,479,680	---	0	---	70.5	1,174,400	---	& S	
	4	2	M	S	TP	900	604	6	16	20	257	774	1930	3,834	3,450,600	---	0	---	70.7	1,637,400	---		
	Plant					2,060	1,383					1,744			7,687,680	2,840	0	2,706		3,485,700	1,227		62.0
728	1	4	A	-	TP	600	403	6	16.25	23	225	512	1926	4,100	2,460,000	---	---	---	77.1	1,274,700	---		
	2	4	A	-	TP	600	403	6	16.25	23	225	512	1926	3,900	2,340,000	---	---	---	77.4	1,216,900	---	BC	
	3	2	M	S	TP	840	564	6	16	20	257	700	1929	3,800	3,192,000	---	---	---	51.8	1,109,830	---		
	Plant					2,040	1,370					1,724			7,992,000	5,775	---	1,382		3,601,330	624		41.0
93	1	2	M	S	TP	700	470	5	16	20	257	---	1928	6,115	4,280,500	---	---	---	51.5	1,478,600	---		
	2	2	M	S	TP	700	470	5	16	20	257	---	1928	6,002	4,201,400	---	---	---	51.7	1,457,100	---	CC	
	3	2	M	S	TP	560	376	4	16	20	257	---	1928	3,406	1,907,360	---	---	---	55.9	716,100	---		
	Plant					1,960	1,316					---			10,389,260	2,839	---	3,660		3,651,800	1,286		46.0
215	1	2	M	S	TP	980	658	7	16	20	257	845	1930	1,566	1,534,680	---	---	---	89.0	917,000	---	BC	
	2	2	M	S	TP	980	658	7	16	20	257	845	1930	1,141	1,118,180	---	---	---	91.2	684,100	---		
	Plant					1,960	1,316					1,690			2,652,860	822	---	3,230		1,601,100	1,947		39.2
837	1	4	A	-	CH	750	504	4	23	32	150	625	1922	7,636	5,726,750	2,161	---	2,650	50.2	1,933,280	895		
	2	4	A	-	TP	450	302	6	16	24	164	413	1913	492	221,400		---						
	3	4	A	-	TP	225	151	3	16	24	164	200	1904	1,024	230,400	1,528	---	572	56.3	330,954	216	F	
	4	4	A	-	TP	225	151	3	16	24	164	200	1904	1,101	247,725		---						
	5	4	A	-	TP	225	151	3	16	24	164	187	1906	774	174,150		---						
	Plant					1,875	1,259					1,625			6,600,425	3,689	---	1,789		2,264,234	614		37.3
997	1	4	A	-	TP	600	403	6	16.25	23	225	475	1929	7,839	4,733,400	2,676	200	1,768	--	-----	---		
	2	4	A	-	TP	600	403	6	16.25	23	5	475	1929	7,976	4,785,600	2,886	200	1,658	--	-----	---	CC	
	3	4	A	-	TP	600	403	6	16.25	23	225	475	1929	7,826	4,695,600	2,493	200	1,883	--	-----	---		
	Plant					1,800	1,209					1,425			14,214,600	8,055	600	1,764		7,210,320	895		41.0
212	1	4	A	-	TP	875	588	8	17.5	24	200	671	1929	245	214,375	94	---	2,280	73.3	105,500	1,123		
	2	4	A	-	TP	875	588	8	17.5	24	200	671	1929	316	276,500	86	---	3,215	66.7	123,900	1,440	S & F	
	Plant					1,750	1,176					1,342			490,875	180	---	2,725		229,400	1,275		65.0

TABLE III - INFORMATION COVERING LUBRICATING OIL (Page 3)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM - SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	TRUNK PISTON OR CROSS HEAD	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING - K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	RATED HORSEPOWER HOURS IN REPORTED PERIOD	GALLONS OF NEW LUBRICATING OIL USED	GALLONS OF UNFIT LUBRICATING OIL DISCARDED	RATED H.P. HRS. PER GALLON OF NEW LUBRICATING OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	GROSS OUTPUT - K.W. HRS.	GROSS K.W. HRS. PER GALLON OF NEW LUBRICATING OIL	LUBRICATING OIL TREATMENT (SEE NOTES)	AVERAGE COST OF LUBRICATING OIL - CENTS PER GALLON
722	1	2	M	CC	TP	200[d]	134	4	14	17	257	170	1923	282	56,400	---	---	---	73.7		27,870	---		
	2	2	M	CC	TP	300[d]	202	6	14	17	257	250	1923	392	117,600	---	---	---	79.3		62,760	---		
	3	2	M	S	TP	840	564	6	16	20	257	700	1929	1,830	1,537,200	---	---	---	70.8		731,400	---	BC	
	4	4	A	-	TP	365	245	4	16.25	21	225	330	1916	729	266,085	---	---	---	81.1		144,900	---		
	Plant					1,705	1,145					1,450			1,977,285	864		2,290		72.8	966,930	1,119		54.0
735	1	2	M	S	TP	840	564	6	16	20	257	700	1928	3,256	2,735,040	---	0	---	82.6		1,517,600	---		
	2	2	M	S	TP	840	564	6	16	20	257	700	1928	3,256	2,735,040	---	0	---	73.1		1,341,800	---	BC	
	Plant					1,680	1,128					1,400			5,470,080	1,924	0	2,842		77.8	2,859,400	1,485		69.9
18	1	2	M	CC	CH	450	302	5	15.25	16	275	---	1922	---	-----	---	---	---	--		-----	---	---	
	2	4	A	-	TP	600	403	6	17	25	200	---	1926	---	-----	---	---	---	--		-----	---	---	
	3	4	A	-	TP	600	403	6	17	25	200	---	1928	---	-----	---	---	---	--		-----	---	---	
	Plant					1,650	1,108					---			-----	2,780	---	---		--	2,802,301	900		50.1
8	1	4	A	-	TP	400	269	4	16.5	24	200	---	1921	4,958	1,983,200	---	0	---	44.9		898,430	---	CC & F	
	2	4	A	-	TP	560	376	3	22.5	22.5	225	---	1922	335	187,600	---	0	---	43.7		55,070	---	CC	
	3	4	A	-	TP	600	403	6	17	25	200	---	1926	4,430	2,658,000	---	0	---	70.7		1,249,160	---	CC	
	Plant					1,560	1,048					---			4,828,800	2,136	0	2,260		58.7	1,902,660	891		35.9
10	1	4	A	-	TP	750	504	4	22	29.5	166.7	625	1924	3,717	2,787,750	1,545	0	1,803	66.9		1,253,000	811		
	2	2	A	S	CH	750	504	4	17	27	176.5	625	1925	4,881	3,660,750	2,437	0	1,502	72.0		1,772,700	727	CC	
	Plant					1,500	1,008					1,250			6,448,500	3,982	0	1,619		69.9	3,025,700	760		50.0
132	1	4	A	-	TP	375	252	3	17.5	24.5	225	319	1929	4,479	1,679,625	334	---	5,030	62.9		709,600	2,123		
	2	4	A	-	TP	500	336	4	17.5	24.5	225	432	1929	3,164	1,582,000	278	---	5,690	74.1		788,100	2,833	CC	
	3	4	A	-	TP	625	420	5	17.5	24.5	225	538	1929	1,351	844,375	155	---	5,445	71.8		407,600	2,630		
	Plant					1,500	1,008					1,289			4,106,000	767	---	5,350		69.1	1,905,300	2,483		55.7
289	1	4	A	-	TP	250	168	4	13.75	17.5	257	210	1920	5,991	1,497,750	714	0	2,098	70.0		704,190	986		
	2	4	A	-	TP	250	168	4	13.75	17.5	257	210	1920	5,594	1,398,500	674	0	2,075	69.3		651,560	968		
	3	4	A	-	TP	365	245	4	16.25	21	225	300	1915	3,175	1,158,875	369	0	3,140	67.8		527,890	1,430	BC	
	4	4	A	-	TP	600	403	6	16.25	23	225	512	1927	2,367	1,420,200	687	0	2,070	77.6		740,100	1,077		
	Plant					1,465	984					1,232			5,475,325	2,444	0	2,240		71.3	2,623,740	1,073		52.4
865	1	4	A	-	TP	225	151	3	16	24	164	160	1908[e]	---	-----	---	---	---	--		4,970	---		
	2	4	A	-	TP	225	151	3	16	24	164	187	1911[e]	---	-----	---	---	---	--		5,920	---		
	3	4	A	-	TP	500	336	4	18.875	28.375	164	475	1921	---	-----	---	---	---	--		197,970	---	S	
	4	4	A	-	TP	500	336	4	18.875	28.375	164	475	1917	---	-----	---	---	---	--		171,510	---		
	Plant					1,450	974					1,297			-----	499	---	---		--	380,370	762		65.7
70	1	2	M	S	TP	560	376	4	16	20	257	470	1928	3,451	1,932,560	---	---	---	65.1		844,865	---	CC & S	
	2	2	M	S	TP	560	376	4	16	20	257	470	1928	4,129	2,312,240	---	---	---	59.6		925,650	---	CC & S	
	3	2	M	CC	TP	300[d]	202	6	14	17	257	250	1920	2,200	660,000	---	---	---	61.3		272,585	---	F	
	Plant					1,420	954					1,190			4,904,800	2,083	---	2,350		62.0	2,043,100	981		41.5
154	1	4	A	-	TP	600	403	6	17	24	200	---	1927	---	-----	---	---	---	--		-----	---	---	
	2	4	A	-	TP	400	269	4	17	24	200	---	1926	---	-----	---	---	---	--		-----	---	---	
	3	4	A	-	TP	400	269	4	17	24	200	---	1925	---	-----	---	---	---	--		-----	---	---	
	Plant					1,400	941					---			-----	835	---	---		--	1,641,605	1,965		41.8
163	1	4	M	-	TP	400	269	4	17.75	22	225	343	1924	5,306	2,122,400	337	144	6,300	73.9		1,054,400	3,130		
	2	4	M	-	TP	400	269	4	17.75	22	225	343	1927	6,521	2,608,400	783	125	3,330	76.9		1,349,500	1,722	BC	
	3	4	M	-	TP	600	403	6	17.75	22	225	625	1929	5,154	3,092,400	941	150	3,285	77.2		1,601,600	1,702		
	Plant					1,400	941					1,311			7,823,200	2,061	419	3,795		76.3	4,005,500	1,942		29.8
11	1	4	A	-	TP	750	504	4	22	29.5	166.7	625	1924	233	174,750	---	0	---	88.5		103,940	---		
	2	4	A	-	TP	600	403	6	16.5	24	200	513	1926	3,165	1,899,000	---	0	---	96.7		1,232,600	---	CC	
	Plant					1,350	907					1,138			2,073,750	1,508	0	1,373		95.9	1,336,540	886		36.9
725	1	4	A	-	TP	600	403	6	17	25	200	500	1926	---	-----	---	---	---	--		-----	---		
	2	4	A	-	TP	400	269	4	17	25	200	337	1925	---	-----	---	---	---	--		-----	---	CC	
	3	2	M	CC	TP	200[d]	134	4	14	17	257	170	1924	---	-----	---	---	---	--		-----	---	& BC	
	4	2	M	CC	TP	110[d]	74	2	14	17.5	257	90	1923	---	-----	---	---	---	--		-----	---	& S & C	
	Plant					1,310	880					1,097			-----	2,220	65	---		--	-----	---		69.2
13	1	4	A	-	TP	275	185	4	13.5	19.5	257	219	1923	6,509	1,789,975	1,006	300	1,779	61.7		743,100	738		
	2	4	A	-	TP	200	134	3	13.5	19.5	257	156	1923	3,402	680,400	503	150	1,352	65.5		298,700	594	CC &	
	3	4	A	-	TP	750	504	4	23	29.5	164	625	1925	1,574	1,180,500	802	344	1,472	49.0		388,200	484	F	
	Plant					1,225	823					1,000			3,650,875	2,311	794	1,580		58.3	1,430,000	619		39.7
17	1	4	A	-	TP	600	403	6	17	25	200	---	1926	---	-----	---	---	---	--		-----	---		
	2	4	A	-	TP	600	403	6	17	25	200	---	1926	---	-----	---	---	---	--		-----	---	---	
	Plant					1,200	806					---			-----	1,745	---	---		--	2,258,800	1,293		51.1
94	1	4	A	-	TP	600	403	6	17	24	200	---	1927	1,437	862,200	---	---	---	64.6		374,330	---	CC &	
	2	4	A	-	TP	600	403	6	17	24	200	---	1927	1,374	824,400	---	---	---	72.4		401,080	---	S	
	Plant					1,200	806					---			1,686,600	1,071	---	1,573		68.4	775,410	724		45.6
333	1	4	A	-	TP	1,200	806	6	20	24	300	950	1929	950	1,140,000	2,200	750	518	83.3	83.3	637,000	290	S	49.6
451	1	2	M	S	TP	600	403	4	16	20	257	512	1931[f]	---	-----	---	---	---	--		469,230	---		
	2	2	M	S	TP	600	403	4	16	20	257	512	1931[f]	---	-----	---	---	---	--		466,030	---	CC	
	Plant					1,200	806					1,024			-----	1,887[z]	---	---		---	935,260	495		36.2
980	1	4	A	-	TP	450	302	4	17	24.5	225	375	1927	---	-----	---	---	---	--		744,300	---		
	2	4	A	-	TP	750	504	6	17.5	24.5	225	625	1927	---	-----	---	---	---	--		1,415,500	---	CC	
	Plant					1,200	806					1,000			-----	2,478	---	---		--	2,159,800	871		40.0
616	1	4	A	-	TP	300	202	3	16.5	24	200	250	1923	---	-----	---	0	---	--		681,440	---		
	2	2	M	CC	TP	100[d]	67	2	14	17	257	75	1920	---	-----	---	0	---	--		10,000	---	CC &	
	3	4	A	-	CH	750	504	4	23	32	150	657	1925	---	-----	---	0	---	--		1,076,710	---	F	
	Plant					1,150	773					982			-----	1,976	0	---		--	1,768,150	895		61.5
95	1	2	M	S	TP	560	376	4	16	20	257	---	1929	2,468	1,382,080	---	---	---	51.4		477,100	---		
	2	2	M	S	TP	560	376	4	16	20	257	---	1929	2,334	1,307,040	---	---	---	52.4		459,550	---	CC	
	Plant					1,120	752					---			2,689,120	436	---	6,170		51.9	936,650	2,147		45.2

TABLE III - INFORMATION COVERING LUBRICATING OIL (Page 4)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM - SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	TRUNK PISTON OR CROSS HEAD	RATED B.H.P.	EQUIVALENT K.W. - 90% OVERALL EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING - K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	RATED HORSEPOWER HOURS IN REPORTED PERIOD	GALLONS OF NEW LUBRICATING OIL USED	GALLONS OF UNFIT LUBRICATING OIL DISCARDED	RATED H.P. HRS. PER GALLON OF NEW LUBRICATING OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	GROSS OUTPUT - K.W. HRS.	GROSS K.W. HRS PER GALLON OF NEW LUBRICATING OIL	LUBRICATING OIL TREATMENT (SEE NOTES)	AVERAGE COST OF LUBRICATING OIL - CENTS PER GALLON
96	1 4 M	-	TP	600	403	6	17	24	225	---	1926	1,220	732,000	---	---	---	46.8		230,100	---	None	
	2 4 M	-	TP	300	202	3	17	24	225	---	1923	6,984	2,095,200	---	---	---	53.2		751,600	---		
	3 2 M	CC	TP	200[d]	134	4	14	17	257	---	1921	1,041	208,200	---	---	---	40.3		56,200	---		
	Plant			1,100	739					---			3,035,400	961	---	3,160		50.9	1,037,900	1,080		45.2
721	1 2 M	CC	TP	360	242	6	14	17	257	300	1925	---	-----	---	---	---	--		-----	---	CC	
	2 2 M	CC	TP	360	242	6	14	17	257	300	1925	---	-----	---	---	---	--		-----	---	& F	
	3 2 M	CC	TP	360	242	6	14	17	257	300	1926	---	-----	---	---	---	--		-----	---		
	Plant			1,080	726					900			-----	4,005	---	---		--	777,700	194		43.0
78	1 4 M	-	TP	400	269	4	17	24	225	300	1928	6,134	2,453,600	3,059	250	803	34.0		560,474	183	CC	
	2 4 M	-	TP	660	443	6	17	24	240	500	1930	6,016	3,970,560	1,200	100	3,308	54.3		1,445,360	1,204		
	Plant			1,060	712					800			6,424,160	4,259	350	1,508		46.5	2,005,834	471		50.4
83	1 4 A	-	TP	600	403	6	16.25	23	225	520	1929	4,790	2,874,000	431	0	6,670	59.8		1,155,900	2,680	BC	
	2 4 A	-	TP	400	269	4	16.25	23	225	340	1927	5,042	2,016,800	333	0	6,050	46.6		632,500	1,900		
	Plant			1,000	672					860			4,890,800	764	0	6,405		54.4	1,788,400	2,340		68.6
98	1 2 M	CC	TP	360	242	6	14	17	257	---	1926	3,328	1,198,080	---	---	---	65.4		526,800	---	BC	
	2 2 M	CC	TP	360	242	6	14	17	257	---	1929	3,329	1,198,440	---	---	---	65.0		523,500	---		
	3 2 M	CC	TP	180	121	3	14	17	257	---	1926	5,180	932,400	---	---	---	67.3		421,900	---		
	4 2 M	CC	TP	100[d]	67	2	14	17	257	---	--	0	0	---	---	---	0		0	---		
	Plant			1,000	672					---			3,328,920	1,676	---	1,985		65.8	1,472,200	878		40.0
738	1 4 A	-	CH	1,000	672	6	22	32	150	720	1918[h]	2,249	2,249,000	1,334	0	1,685	76.3	76.3	1,151,600	863	CC	49.5
801	1 2 A	S	CH	1,000	672	4	20.75	26	180	937	1927	8,390	8,390,000	4,240	0	1,977	84.9	84.9	4,778,300	1,126	BC & S & F	32.9
1061	1 2 M	S	TP	980	658	7	16	20	257	975	1930	3,822	3,745,560	2,355	660	1,590	81.4	81.4	2,048,030	870	BC & S	46.8
883	1 4 A	-	TP	300	202	3	17	24	200	250	1925	4,806	1,441,800	421	110	3,425	78.1		753,800	1,790	CC	
	2 4 A	-	TP	675	453	6	17	24	225	625	1927	5,484	3,701,700	2,560	50	1,445	84.5		2,100,700	821	& F	
	Plant			975	655					875			5,143,500	2,981	160	1,725		82.7	2,854,500	958		62.0
49	1 4 A	-	TP	485	326	4	18	24	200	420	1924	4,156	2,015,660	---	---	---	50.9		689,270	---	CC	
	2 4 A	-	TP	485	326	4	18	24	200	420	1924	4,816	2,338,760	---	---	---	51.9		814,170	---		
	Plant			970	652					840			4,351,420	842	310	5,170		51.5	1,503,440	1,785		64.5
53	1 2 M	CC	TP	200[d]	134	4	14	17	257	170	1923	2,581	516,200	621	40	831	69.6		240,560	387		
	2 2 M	CC	TP	300[d]	202	6	14	17	257	250	1923	4,857	1,457,100	1,373	80	1,061	72.0		706,180	514	S	
	3 4 A	-	TP	400	269	4	16.25	23	225	338	1925	7,927	3,170,800	948	80	3,345	70.7		1,506,880	1,589		
	Plant			900	605					758			5,144,100	2,942	200	1,748		71.0	2,453,620	834		42.3
260	1 2 A	S	TP	900	605	6	14.75	21	257	750	1928	263	236,700	45	0	5,260	60.6	60.6	80,593	1,790	CC	55.5
74	1 2 M	CC	CH	225	151	3	15.25	16	277	187	1924	378	85,050	95	0	895	58.0		33,088	348		
	2 4 M	-	TP	260	175	4	14.5	18	277	219	1928	5,768	1,499,680	354	174	4,235	71.5		721,602	2,035	C	
	3 4 M	-	TP	400	269	6	14.5	18	277	344	1928	2,794	1,117,600	144	86	7,760	58.4		439,202	3,050		
	Plant			885	595					760			2,702,330	593	260	4,555		65.7	1,193,892	2,010		58.9
189	1 4 M	-	TP	550	369	5	17	24	240	470	1929	8,606	4,733,300	---	0	---	31.3		993,708	---	B &	
	2 4 M	-	TP	200[b]	134	2	17	24	200	170	1922	154	30,800	---	0	---	66.1		13,645	---	C	
	3 4 M	-	TP	100[b]	67	1	17	24	200	90	1915	0	0	---	0	---	--		0	---		
	Plant			850	570					730			4,764,100	570	0	8,360		31.5	1,007,351	1,765		59.8
153	1 2 M	S	TP	840	564	6	16	20	257	700	1929	---	-----	700	---	---	--	--	594,940	850	--	49.9
494	1 2 M	S	TP	840	564	6	16	20	257	700	1930	2,164	1,817,760	620	150	2,930	76.8	76.8	937,565	1,511	CC	50.0
619	1 2 M	S	TP	840	564	6	16	20	257	700	1929	2,090	1,755,600	1,007	300	1,743	74.1	74.1	873,880	867	CC	53.1
1056	1 2 M	S	TP	840	564	6	16	20	257	700	1929	4,822	4,050,480	1,975	530	2,050	76.2	76.2	2,073,460	1,049	BC & S	49.0
731	1 4 A	-	TP	825	554	4	23	32	164	750	1926	1,825	1,505,625	1,320	---	1,140	61.2	61.2	618,502	468	BC,S&F	45.1
648	1 4 M	-	TP	400	269	4	17	24	225	333	1928	2,555	1,022,000	407	60	2,510	80.5		553,000	1,358		
	2 4 M	-	TP	400	269	4	17	24	225	333	1928	3,006	1,202,400	496	70	2,425	81.2		655,893	1,321	C	
	Plant			800	538					666			2,224,400	903	130	2,464		80.9	1,208,893	1,339		50.0
100	1 2 M	CC	TP	300[d]	202	6	14	17	257	---	1923	115	34,500	---	---	---	38.6		8,970	---		
	2 2 M	CC	TP	240	161	4	14	17	257	---	1924	2,688	645,120	---	---	---	62.6		271,120	---	BC	
	3 2 M	CC	TP	200[d]	134	4	14	17	257	---	1924	2,646	529,200	---	---	---	72.0		255,440	---		
	Plant			740	497								1,208,820	754	---	1,603		66.0	535,530	710		40.1
112	1 2 M	CC	TP	200[d]	134	4	14	17	257	170	1925	1,798	359,600	241	---	1,491	74.8		180,200	748		
	2 2 M	CC	TP	240	161	4	14	17	257	200	1927	6,930	1,663,200	1,081	---	1,538	69.7		777,900	720	BC	
	3 2 M	CC	TP	300[d]	202	6	14	17	257	250	1925	2,283	684,900	586	---	1,168	73.3		336,400	574		
	Plant			740	497					620			2,707,700	1,908	---	1,417		71.3	1,294,500	678		46.2
67	1 2 M	CC	TP	360	242	6	14	17	257	250	1925	1,860	669,600	740	---	905	82.2		369,790	500		
	2 2 M	CC	TP	360	242	6	14	17	257	250	1925	2,622	943,920	1,044	---	903	82.3		522,010	500	F & C	
	Plant			720	484					500			1,613,520	1,784	---	904		82.3	891,800	500		43.3
170	1 2 M	CC	TP	360	242	6	14	17	257	300	1929	2,700	972,000	687	119	1,414	--		-----	---		
	2 2 M	CC	TP	360	242	6	14	17	257	300	1929	2,830	1,018,800	693	119	1,470	--		-----	---	C & F	
	Plant			720	484					600			1,990,800	1,380	238	1,442		64.5	883,979	626		47.5
527	1 2 M	CC	TP	360	242	6	14	17	257	300	1930	5,000	1,800,000	---	0	---	37.2		450,000	---	B & C	
	2 2 M	CC	TP	360	242	6	14	17	257	300	1930	5,602	2,016,720	---	0	---	45.2		612,000	---	& F	
	Plant			720	484					600			3,816,720	1,174	0	3,250		41.4	1,062,000	904		64.9
718	1 2 M	CC	TP	360	242	6	14	17	257	300	1926	4,593	1,653,480	---	---	---	40.4		448,950	---	B &	
	2 2 M	CC	TP	360	242	6	14	17	257	300	1926	4,467	1,608,120	---	---	---	28.9		312,750	---	F & C	
	Plant			720	484					600			3,261,600	2,090	---	1,560		34.9	761,700	365		44.0
19	1 2 M	CC	TP	100[d]	67	2	14	17	257	---	--	---	-----	---	---	---	--		-----	---	--	
	2 2 M	CC	CH	240	161	4	12.5	13.25	325	---	1927	---	-----	---	---	---	--		-----	---	--	
	3 2 M	CC	CH	360	242	4	15.25	16	275	---	1927	---	-----	---	---	---	--		-----	---	--	
	Plant			700	470					---			-----	1,863	---	---		--	747,510	401		54.7
368	1 4 A	-	TP	400	269	4	16.25	23	225	340	1925	6,250	2,500,000	980	---	2,605	81.8		1,375,600	1,433	CC	
	2 4 A	-	TP	300	202	3	16.25	23	225	250	1929	4,750	1,425,000	638	---	2,234	96.0		921,500	1,444		
	Plant			700	471					590			3,925,000	1,598	---	2,455		87.2	2,297,100	1,437		66.0

TABLE III - INFORMATION COVERING LUBRICATING OIL (Page 5)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM - SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	TRUNK PISTON OR CROSS HEAD	RATED B.H.P.	EQUIVALENT K.W. 90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	RATED BHP33POWER HOURS IN REPORTED PERIOD	GALLONS OF NEW LUBRICATING OIL USED	GALLONS OF UNFIT LUBRICATING OIL DISCARDED	RATED H.P. HRS. PER GALLON OF NEW LUBRICATING OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	GROSS OUTPUT - K.W. HRS.	GROSS K.W. HRS. PER GALLON OF NEW LUBRICATING OIL	LUBRICATING OIL TREATMENT (SEE NOTES)	AVERAGE COST OF LUBRICATING OIL - CENTS PER GALLON
61	1 | 4 | M	-	TP	300	202	3	17.75	22	225	275	1928	7,759	2,327,700	---	---	---	69.5		1,083,600	---	CC & F	
	2 | 4 | M	-	TP	300	202	3	17.75	22	225	275	1928	3,541	1,062,300	---	---	---	66.9		476,100	---		
	Plant			600	404					550			3,390,000	1,348	50	2,515		68.4	1,559,700	1,157		42.0
106	1 | 4 | A	-	TP	600	403	6	17	25	200	---	1929	---	-----	685	---	---	--	--	1,849,330	2,260	--	52.0
1094	1 | 4 | A	-	TP	600	403	6	16.25	23	225	513	1928	2,213	1,327,800	362	100	3,670	78.5	78.5	700,574	1,934	C	50.0
1096	1 | 4 | A	-	TP	600	403	6	16.25	23	225	513	1928	2,570	1,542,000	723	180	2,133	74.0	74.0	766,386	1,069	C	50.0
103	1 | 2 | M	CC	TP	240	161	4	14	17	257	---	1927	5,497	1,319,280	---	---	---	57.1		505,150	---		
	2 | 2 | M	CC	TP	200d	134	4	14	17	257	---	1920i	3,731	746,200	---	---	---	65.1		325,500	---	None	
	3 | 2 | M	CC	TP	150d	101	3	14	17	257	---	1923	4,353	652,950	---	---	---	52.8		232,080	---		
	Plant			590	396					---			2,718,430	1,694	---	1,604		58.3	1,062,730	627		40.2
962	1 | 2 | M	CC	TP	75d	50	2	12	15	300	60	1920	89	6,675	419	---	2,130	--		-----	---	F	
	2 | 2 | M	CC	TP	75d	50	2	12	15	300	60	--	242	18,150		---		--		-----	---		
	3 | 2 | M	CC	TP	180	121	3	14	17	257	150	1925	4,823	868,140		---		--		-----	---		
	4 | 4 | A	-	TP	258	173	3	16.5	24.5	180	225	1918j	4,947	1,250,526	662	---	1,889	--		-----	---		
	Plant			588	394					495			2,143,491	1,081	---	1,982		46.9	676,320	625		41.4
102	1 | 2 | M	CC	TP	240	161	4	14	17	257	---	1928	3,778	906,720	---	---	---	46.9		285,570	---	BC & S	
	2 | 2 | M	CC	TP	240	161	4	14	17	257	---	1928	3,517	844,080	---	---	---	49.9		282,220	---		
	3 | 2 | M	CC	TP	100d	67	2	14	17	257	---	1922j	2,158	215,800	---	---	---	65.0		93,940	---		
	Plant			580	389					---			1,966,600	1,400	---	1,404		50.1	661,730	473		44.8
101	1 | 2 | M	CC	TP	240	161	4	14	17	257	---	1926	5,024	1,205,760	---	---	---	53.8		435,515	---	BC & S	
	2 | 2 | M	CC	TP	150d	101	3	14	17	257	---	1919	1,600	240,000	---	---	---	53.4		86,420	---		
	3 | 2 | M	CC	TP	150d	101	3	14	17	257	---	1923	3,933	589,950	---	---	---	60.3		239,680	---		
	Plant			540	363					---			2,035,710	1,591	---	1,279		55.7	761,615	478		45.1
246	1 | 2 | M	CC	TP	100d	67	2	14	17	257	60	1918	1,990	199,000	---	0	---	--		-----	---	S & C & F	
	2 | 2 | M	CC	TP	200d	134	4	14	17	257	170	1922	4,720	944,000	---	0	---	--		-----	---		
	3 | 2 | M	CC	TP	240	161	4	14	17	257	200	1929	2,200	528,000	---	0	---	--		-----	---		
	Plant			540	362					430			1,671,000	1,518	0	1,100		38.0	427,085	281		58.7
540	1 | 2 | A	CC	TP	360	242	6	14	17	257	300	1929	---	-----	---	---	---	--		-----	---	F	
	2 | 2 | M	CC	TP	180	121	3	14	17	257	125	1926	---	-----	---	---	---	--		-----	---		
	Plant			540	363					425				2,674	---	---		--	750,000	280		45.7
772	1 | 4 | A	-	TP	250	168	4	13.75	17.5	275	250	1916	889	222,250	150	40	1,482	74.5		111,231	742	F	
	2 | 4 | A	-	TP	250	168	4	13.75	17.5	275	263	1916	977	244,250	150	60	1,627	74.5		122,300	815		
	Plant			500	336					513			466,500	300	100	1,554		74.5	233,531	778		50.3
501	1 | 2 | M	CC	TP	120	81	2	14	17	257	90	1925	4,896	587,520	499	42	1,176	32.3		128,038	257	CC & S & F	
	2 | 2 | M	CC	TP	120	81	2	14	17	257	90	1925	4,829	579,480	541	40	1,071	33.3		128,975	239		
	3 | 2 | M	CC	TP	240	161	4	14	17	257	200	1929	2,625	630,000	380	50	1,658	51.3		216,700	570		
	Plant			480	323					380			1,797,000	1,420	132	1,265		39.3	473,713	335		50.9
615	1 | 2 | M	CC	TP	100d	67	2	14	17	257	75	1922	4,130	413,000	---	---	---	--		-----	---	S & F	
	2 | 2 | M	CC	TP	100d	67	2	14	17	257	75	1922	490	49,000	---	---	---	--		-----	---		
	3 | 2 | M	CC	TP	240	161	4	14	17	257	219	1930	4,995	1,198,800	---	---	---	--		-----	---		
	Plant			440	295					369			1,660,800	1,160	---	1,432		--	-----	---		50.0
931	1 | 2 | A	S	OH	440	295	4	15	20	225	---	1922	3,706	1,630,640	908	108	1,795	74.6	74.6	814,915	898	CC & S	60.0
335	1 | 4 | A	-	TP	250	168	4	13.75	17.5	257	219	1924	6,310	1,577,500	368	56	4,290	34.2		362,040	984	S	
	2 | 4 | A	-	TP	180	121	4	11.75	15	300	150	1919	2,450	441,000	154	55	2,865	47.9		141,960	922		
	Plant			430	289					369			2,019,500	522	111	3,865		37.3	504,000	965		60.0
24	1 | 4 | M	-	TP	200	134	3	14.5	18	277	163	1925	4,531	906,200	---	---	---	47.3		287,300	---	None	
	2 | 4 | M	-	TP	200	134	3	14.5	18	277	163	1925	4,531	906,200	---	---	---	47.0		285,400	---		
	Plant			400	268					326			1,812,400	1,320	200	1,372		47.1	572,700	437		51.6
194	1 | 4 | F	-	TP	400	269	4	17	24	225	---	1928	4,115	1,646,000	220	---	7,480	34.7	34.7	384,250	1,747	C	67.3
247	1 | 2 | M	CC	TP	50d	34	1	14	17	257	40	1915	320	16,000	---	0	---	--		-----	---	C & F	
	2 | 2 | M	CC	TP	100d	67	2	14	17	257	75	1920	3,650	365,000	---	0	---	--		-----	---		
	3 | 2 | M	CC	TP	240	161	4	14	17	257	200	1928	5,115	1,227,600	---	0	---	--		-----	---		
	Plant			390	262					318			1,608,600	880	0	1,828		44.6	491,150	547		55.0
105	1 | 2 | M	CC	TP	150d	101	3	14	17	257	---	1922	4,580	687,000	---	---	---	55.1		255,060	---	BC	
	2 | 2 | M	CC	TP	120	81	2	14	17	257	---	1926	2,668	320,160	---	---	---	34.7		74,960	---		
	3 | 2 | M	CC	TP	100d	67	2	14	17	257	---	1923	2,670	267,000	---	---	---	42.2		75,560	---		
	Plant			370	249					---			1,274,160	763	---	1,669		47.4	405,580	531		46.5
152	1 | 2 | M	CC	TP	360	242	6	14	17	257	---	1925	---	-----	82	---	---	--	--	54,600	666	--	50.0
161	1 | 2 | M	CC	TP	360	242	6	14	17	257	300	1929	4,335	1,560,600	1,598	---	977	95.4	95.4	1,001,000	626	--	46.5
1091	1 | 2 | M	CC	TP	180	121	3	14	17	257	135	1929	4,380	788,400	---	---	---	36.1		191,300	---	None	
	2 | 2 | M	CC	TP	180	121	3	14	17	257	135	1930	4,380	788,400	---	---	---	36.2		191,500	---		
	Plant			360	242					270			1,576,800	935	---	1,686		36.2	382,800	409		56.0
688	1 | 2 | M	CC	TP	50d	34	1	14	17	257	---	1923	---	-----	---	---	---	--		-----	---	S & F	
	2 | 2 | M	CC	TP	120	81	2	14	17	257	---	1926	---	-----	---	---	---	--		-----	---		
	3 | 2 | M	CC	TP	180	121	3	14	17	257	---	1927	---	-----	---	---	---	--		-----	---		
	Plant			550	236					---			-----	900	100	---		--	-----	---		31.7
365	1 | 4 | A	-	TP	165	111	4	11.75	15	277	110	1917	---	-----	---	---	---	--		262,700	---	C	
	2 | 4 | A	-	TP	180	121	4	11.75	15	300	120	1922	---	-----	---	---	---	--		290,800	---		
	Plant			345	232					230			-----	---	---	---		--	553,500	---		--
774	1 | 4 | A	-	TP	165	111	4	11.75	15	277	135	1917	972	160,380	90	50	1,782	68.3		73,600	818	None	
	2 | 4 | A	-	TP	180	121	4	11.75	15	300	135	1920	525	94,500	66	20	1,432	62.4		39,647	601		
	Plant			345	232					270			254,880	156	50	1,632		56.2	113,247	726		50.0
265	1 | 2 | M	CC	TP	60	40	1	14	17	257	48	1927	2,555	153,300	---	---	---	--		-----	---	S & F	
	2 | 2 | M	CC	TP	120	81	2	14	17	257	90	1928	6,908	828,600	---	---	---	--		-----	---		
	3 | 2 | M	CC	TP	140	94	2	14	17	300	111	1931k	400	56,000	---	---	---	--		-----	---		
	Plant			320	215					249			1,037,900	450	201	2,307		--	-----	---		87.8

TABLE III - INFORMATION COVERING LUBRICATING OIL (Page 6)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM - SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	TRUNK PISTON OR CROSS HEAD	RATED B.H.P.	EQUIVALENT K.W. 90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	RATED HORSEPOWER HOURS IN REPORTED PERIOD	GALLONS OF NEW LUBRICATING OIL USED	GALLONS OF UNFIT LUBRICATING OIL DISCARDED	RATED H. P. HRS. PER GALLON OF NEW LUBRICATING OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	GROSS OUTPUT - K.W. HRS.	GROSS K.W. HRS. PER GALLON OF NEW LUBRICATING OIL	LUBRICATING OIL TREATMENT (SEE NOTES)	AVERAGE COST OF LUBRICATING OIL - CENTS PER GALLON
27	1 2 M			CC	TP	75[d]	50	2	12	15	300	---	1926	---	-----	-----	--	---	--		---	---	--	
	2 2 M			CC	CH	240	161	4	12.5	13.25	325	---	1926	---	-----	-----	--	---	--		---	---	--	
	Plant					315	211					---			-----	633	--	---		--	229,080	362		50.1
382	1 4 M			-	TP	150	101	4	9.5	14	300	---	1927	3,279	491,850	-----	--	---	20.5		67,900	---	BC	
	2 4 M			-	TP	150	101	4	9.5	14	300	---	1927	1,135	170,250	-----	--	---	20.4		23,400	---		
	Plant					300	202					---			662,100	178	--	3,720		20.5	91,300	513		39.9
1129	1 2 M			CC	TP	b 75[d]	50	2	10.5	14	257	72	--	3,337	250,275	-----	--	---	31.5		52,515	---		
	2 2 M			CC	TP	b 75[d]	50	2	10.5	14	257	72	--	75	5,625	-----	--	---	27.6		1,035	---	None	
	3 2 M			CC	TP	150[d]	101	3	14	17	257	125	1922	5,402	810,300	-----	--	---	29.2		159,250	---		
	Plant					300	201					269			1,066,200	1,031	--	1,034		29.7	212,800	206		45.3
156	1 2 M			CC	TP	150[d]	101	3	14	17	257	125	1929[1]	---	-----	-----	--	---	--		---	---	F	
	2 2 M			CC	TP	120	81	2	14	17	257	90	1929	---	-----	-----	--	---	--		---	---		
	Plant					270	182					215			-----	1,520	--	---		--	199,500	131		42.7
1097	1 2 M			CC	TP	150[d]	101	3	14	17	257	125	1924	---	-----	426	--	---	--		193,776	455	F	
	2 2 M			CC	TP	120	81	2	14	17	257	90	1928	---	-----	167	--	---	--		104,488	625		
	Plant					270	182					215			-----	593	--	---		--	298,264	503		44.8
169	1 4 M			-	TP	240	161	6	10.75	13.75	327	200	1928	2,793	670,320	318	55	2,105	42.5	42.5	191,340	601	CC	51.5
516	1 2 M			CC	TP	120	81	2	14	17	257	94	1926	---	-----	-----	--	---	--		---	---	F	
	2 2 M			CC	TP	120	81	2	14	17	257	94	1930	---	-----	-----	--	---	--		---	---		
	Plant					240	162					188			-----	530	--	---		--	---	---		48.0
222	1 2 M			CC	TP	60	40	1	14	17	257	48	1930	---	-----	-----	--	---	--		---	---	F	
	2 2 M			CC	TP	120	81	2	14	17	257	90	1930	---	-----	-----	--	---	--		---	---		
	Plant					180	121					138			-----	600	--	---		--	44,000	73		42.5
733	1 2 M			CC	CH	100	67	2	12.5	13.25	327	60	1922	41	4,100	3-3/4	0	1,093	67.7		1,860	496	None	
	2 2 M			CC	TP	75[d]	50	2	12	15	300	60	1914	52	3,900	1/2	0	7,800	57.7		1,500	3000	P	
	Plant					175	117					120			8,000	4-1/4	0	1,882		62.5	3,360	791		70.6
646	1 2 M			CC	TP	120	81	2	14	17	257	90	1927	1,789	214,680	114	--	1,882	16.1	16.1	23,340	205	BC & P	47.4
201	1 2 M			CC	TP	50[d]	34	2	10.5	12	375	63	1924	---	-----	-----	--	---	--		---	---	F	
	2 2 M			CC	TP	50[d]	34	1	14	17	257	40	1927	---	-----	-----	--	---	--		---	---		
	Plant					100	68					103			-----	750	--	---		--	71,640	96		46.7

NOTES:

INJECTION SYSTEM
- A - Air
- M - Mechanical

SCAVENGING SYSTEM (2 STROKE CYCLE ONLY)
- S - Separate (i.e., Pump or Blower)
- CC - Crank Case

LUBRICATING OIL TREATMENT
- CC - Continuous Centrifuging
- BC - "Batch" Centrifuging
- C - Chemically
- F - Filtering
- S - Settling

LETTERED NOTES
- a - No deduction made for Motor-driven scavenging blower
- b - Horizontal Engine
- c - November 5th
- d - Semi diesel
- e - Date of original installation; date of installation in this plant not known
- f - May 18th
- g - Includes first filling
- h - Date of original installation; installed in this plant in 1924
- i - Date of original installation; installed in this plant in 1930
- j - Date of original installation; installed in this plant in 1928
- k - November 9th
- l - Date of installation in this plant; date of original installation not known

TABLE IV - INFORMATION COVERING FUEL OIL (Page 1)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM-SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	GROSS OUTPUT - K.W. HRS.	FUEL OIL USED - GALLONS	GROSS K.W. HRS. PER GALLON OF FUEL OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	NATURE OF FUEL OIL USED - (SEE NOTES)	AVERAGE COST OF FUEL OIL-CENTS PER GALLON	IS FUEL CENTRIFUGED?	PLANT ALTITUDE - FEET ABOVE SEA LEVEL	PURPOSE FOR WHICH JACKET WATER HEAT IS UTILIZED (SEE NOTES)	PURPOSE FOR WHICH EXHAUST HEAT IS UTILIZED (SEE NOTES)
43	1	4	A	-	500	336	3	22	29.5	150	375	1915	813	189,480	18,529	10.21	69.3		24° A P I					
	2	4	A	-	500	336	3	22	29.5	150	375	1916	901	206,928	20,382	10.14	68.4		12° S.P. @ 77°F					
	3	4	A	-	520	349	4	19	24.5	200	500	1919	1,775	474,370	50,531	9.39	76.6		0.7% S ; 0.1% BS & W					
	4	4	A	-	520	349	4	19	24.5	200	500	1920	1,243	333,150	35,989	9.26	76.8		0.3% Asph. by Holde Method		NO			
	5	4	A	-	1,000	671	6	20.5	36.0	150	925	1922	2,273	972,250	100,892	9.63	63.7							
	6	4	A	-	1,000	671	6	20.5	36.0	150	925	1925	2,383	1,077,270	112,818	9.55	67.3		19° A P I; 1.05% S					
	7	2	A	S	3,750	2,517	6	30	42	124	3,750	1928	7,851	12,824,700	1,168,975	11.00	66.8		61.3° SF @ 770 F					
	8	2	A	S	3,750	2,517	6	30	42	124	3,750	1929	8,156	16,518,400[a]	1,263,435	13.07[a]	80.5		0.023% ASH; 0.03% BS & W					
			Plant		11,540	7,746					11,100			32,634,548[a]	2,771,551	11.77[a]		73.2	33.55% Asph.-120 hrs.@ 2200 C	3.78		2500	N	N
46	1	2	A	S	1,250	840	5	20.75	26	180	850	1922	1,650	997,760	99,302	10.04	72.0		21° A P I					
	2	2	A	S	1,250	840	5	20.75	26	180	850	1922	2,636	1,556,550	148,875	10.45	70.3		20° S.P. @ 122°F					
	3	2	A	S	2,250	1,510	4	28	44	124	2,000	1922	4,398	5,182,300	454,290	11.40	78.1		1.70% S		NO			
	4	2	A	S	2,250	1,510	4	28	44	124	2,000	1922	4,094	4,931,900	427,870	11.52	79.8		0.2% BS & W					
	5	2	A	S	2,250	1,510	4	28	44	124	2,000	1930	5,912	7,259,400	554,120	13.09	81.4							
			Plant		9,250	6,210					7,700			19,927,910	1,684,463	11.82		78.6		3.75		5000	F	N
168	1	2	A	S	1,250	840	5	20.75	26	180	850	1918	3,825	2,456,000	241,318	10.18	76.5		21°A P I					
	2	2	A	S	1,250	840	5	20.75	26	180	850	1918	4,028	2,554,800	261,350	9.78	79.5		20° S.P. @ 122°F					
	3	2	A	S	1,250	840	5	20.75	26	180	850	1919	4,819	3,179,600	301,486	10.54	78.5		1.70% S					
	4	2	A	S	1,250	840	5	20.75	26	180	850	1916	3,995	2,602,100	258,932	10.04	77.5		0.2% BS & W		NO			
	5	2	A	S	1,250	840	5	20.75	26	180	850	1918	6,338	4,114,800	393,895	10.44	73.1							
	6	2	A	S	2,250	1,510	4	28	44	124	2,000	1929	6,003	8,796,000	723,156	12.16	97.1							
			Plant		8,500	5,710					6,250			23,703,300	2,180,137	10.87		83.5		4.00		4700	F	N
82	1	4	A	-	360	242	4	10.25	21	225	312	1921	963	148,860	14,308	10.40	63.9		25.0° A P I					
	2	2	A	S	750	504	4	17	27	180	740	1923	1,410	408,630	45,82[illegible]	8.92	57.5		42° S U @ 100°F					
	3	2	A	S	750	504	4	17	27	180	740	1924	996	293,335	37,261	9.09	58.3		1.1% S		NO			
	4	2	A	S	1,150	772	6	17	27	180	1,000	1927	3,077	1,626,800	151,169	10.76	68.5		Trace BS & W					
	5	2	A	S	1,150	772	6	17	27	180	1,000	1927	3,489	1,810,100	170,890	10.59	67.1		By Tank Truck					
	6	2	A	S	1,500	1,007	8	17	27	180	1,350	1928	6,466	4,701,300	427,986	10.98	72.2		For 10 months outof 12					
			Plant		5,060	3,801					5,142			8,989,025	842,442	10.67		69.0		5.13		23	N	N
91	1	4	M	-	1,125	755	6	22	30	180	--	1927	4,80[illegible]	2,125,400	-- --	--	58.0		-- --					
	2	4	M	-	1,125	755	6	22	30	180	--	1928	6,826	3,013,700	-- --	--	58.4		-- --					
	3	4	M	-	600	403	6	17	24	225	--	1927	3,705	859,900	-- --	--	57.5		-- --		NO			
	4	4	M	-	600	403	6	17	24	225	--	1927	3,641	835,800	-- --	--	57.0		-- --					
	5	4	A	-	1,800	1,208	6	24	32	225	--	1930	1,743	1,406,200	-- --	--	66.6		-- --					
			Plant		5,250	3,524					--			8,241,000	730,943	11.27		59.3		3.80		6000	N	P
73	1	2	A	S	1,250	840	5	20.75	26	180	1,063	1923	5,572	2,9[illegible]	248,8[illegible]	12.04	64.0		11° - 14° A P I					
	2	2	A	S	1,250	840	5	20.75	26	180	1,063	1924	6,345	3,56[illegible],800	344,158	10.3[illegible]	66.9		750° S U @ 100°F		YES			
	3	2	A	S	1,250	840	5	20.75	26	180	1,063	1928	3,167	1,856,500	185,497	10.00	69.8							
	4	2	A	S	1,250	840	5	20.75	26	180	1,063	1928	736	396,600	39,065	10.12	64.0							
			Plant		5,000	3,360					4,252			8,815,500	817,803	10.78		68.3		0.95		600	N	B
98	1	2	A	S	1,250	840	5	20.75	26	180	--	1928	3,252	1,780,700	-- --	--	65.2		-- --					
	2	2	A	S	1,250	840	5	20.75	26	180	--	1928	2,929	1,747,900	-- --	--	71.1		-- --		NO			
	3	2	A	S	2,400	1,612	8	21	24	225	--	1930	6,291	6,202,800	-- --	--	81.2		-- --					
			Plant		4,900	3,292					--			9,731,200	978,007	9.94		63.8		2.03		2580	N	N
129	1	2	A	S	1,250	840	5	20.75	26	180	850	1918	5,178	2,918,125	287,881	10.13	67.1		16° A P I ; 0.8% S					
	2	2	A	S	1,250	840	5	20.75	26	180	850	1921	5,204	3,023,725	289,304	10.45	69.2		103° SF @ 122°F		NO			
	3	2	A	S	2,250	1,510	4	28	44	124	2,000	1930	6,081	7,866,250	628,735	12.50	85.6		0.4% B S & W					
			Plant		4,750	3,190					3,700			13,808,100	1,205,920	11.45		77.1		3.50		3300	F	N
52	1	4	A	-	830	557	4	23	32	164	700	1928	2,143	833,400	76,959	10.82	69.5		27° A P I					
	2	4	A	-	1,250	840	6	23	32	164	1,060	1928	3,890	2,018,200	207,085	11.03	80.1		70° S U @ 100°F		NO			
	3	4	A	-	1,250	840	6	23	32	164	1,060	1928	3,943	2,655,800	235,702	11.27	80.2		0.52% S					
	4	4	A	-	1,250	840	6	23	32	164	1,060	1930	3,749	2,465,200	225,404	10.93	78.3							
			Plant		4,580	3,077					3,880			8,592,600	775,150	11.05		78.6		4.83		30	N	B
45	1	4	A	-	600	403	6	16.5	24	200	500	1923	843	153,575	15,960	9.62	45.2		0.35% S 35° A P I					
	2	4	A	-	600	403	6	16.5	24	200	500	1923	777	148,500	16,609	8.95	47.4		0.05% CC 43° S U @ 20C					
	3	4	A	-	750	504	4	23	32	150	625	1923	440	134,275	14,134	9.50	60.5		0.04% S		NO			
	4	4	A	-	1,150	772	6	23	32	150	1,000	1926	5,326	2,822,100	255,992	11.02	68.7		.02%CC Trace B S & W					
	5	4	A	-	1,150	772	6	23	32	150	1,000	1926	5,394	2,841,500	261,704	10.86	68.3							
			Plant		4,250	2,854					3,625			6,099,950	564,399	10.80		66.5	Tank Wagon	4.79		50	N	L
54	1	4	A	-	600	403	6	17	25	200	450		--	-- --	-- --	--	--		27.8° A P I					
	2	4	A	-	600	403	6	17	25	200	450	1925	--	-- --	-- --	--	--		280° S U @ 60°F					
	3	4	A	-	600	403	6	17	25	200	450	-26	--	-- --	-- --	--	--		0.7% S					
	4	4	A	-	600	403	6	17	25	200	450		--	-- --	-- --	--	--		1.0% B S & W		YES			
	5	4	A	-	600	403	6	17	25	200	450		--	-- --	-- --	--	--		Crude Oil, By					
	6	4	A	-	600	403	6	17	25	200	450		--	-- --	-- --	--	--		Pipe Line					
	7	4	A	-	600	403	6	17	25	200	450		--	-- --	-- --	--	--							
			Plant		4,200	2,821					3,150		60,548	23,323,000	1,991,451	11.72		95.6		1.03		400	N	F
864	1	4	A	-	500	336	4	18.875	28.375	164	437	1915	--	663,310	-- --	--	--		14°-20°A P I					
	2	4	A	-	500	336	4	18.875	28.375	164	437	1915	--	784,270	-- --	--	--		200° - 1000° S U @ 100°F		NO			
	3	4	A	-	500	336	4	18.875	28.375	164	437	1915	--	744,250	-- --	--	--							
	4	2	A	S	1,250	840	5	20.75	26	180	1,125	1928	--	1,592,800	-- --	--	--		1.8% S					
	5	2	A	S	1,250	840	5	20.75	26	180	1,125	1928	--	1,653,500	-- --	--	--							
			Plant		4,000	2,688					3,561			5,438,130	561,524	9.69		--		2.11		2900	N	F-L
130	1	4	A	-	600	403	6	10.25	23	225	513	1924	3,557	978,000	80,171	12.19	68.2		16°-30° A P I					
	2	4	A	-	600	403	6	17	24	200	513	1926	2,670	700,000	60,233	11.61	65.1		45°-200° S U @ 100°F		NO			
	3	4	A	-	1,200	806	6	20	24	300	1,068	1929	2,294	1,218,700	110,123	11.07	65.9		0.5% S					
	4	4	A	-	1,200	806	6	20	24	300	1,068	1930	2,841	1,556,500	139,81[illegible]	11.12	68.0							
			Plant		3,600	2,418					3,150			4,453,200	390,343	11.40		67.0		4.19		750	N	L
2	1	4	A	-	1,150	772	6	23	29.5	166.7	1,000	1925	8,018	6,653,930	574,120	10.53	97.7		27°A P I					
	2	4	A	-	1,150	772	6	23	29.5	166.7	1,000	1925	7,597	5,735,380	533,998	10.74	97.8		40° S U @ 100°F		NO			
	3	4	A	-	1,150	772	6	23	29.5	166.7	1,000	1925	7,978	6,033,930	572,352	10.53	98.0		0.5% S ; 1.5% CC					
			Plant		3,450	2,316					3,000			17,823,240	1,680,470	10.60		97.8	0.004% Ash.	2.11		735	N	L
164	1	4	M	-	840	564	6	19.5	24	214	1,000	1926	6,153	2,772,800	-- --	--	79.9		32.7° A P I ; 0.82%S					
	2	4	M	-	840	564	6	19.5	24	214	1,000	1926	6,503	2,762,500	-- --	--	75.3		40.8° S U @ 100°F		NO			
	3	4	M	-	840	564	6	19.5	24	214	1,000	1926	7,64[illegible]	3,015,400	-- --	--	69.9		0.021% CC					
	4	4	M	-	840	564	6	19.5	24	214	1,000	1926	6,64[illegible]	3,038,600	-- --	--	81.0		0.0018% Ash.					
			Plant		3,360	2,256					4,000			11,589,300	881,071	13.15		76.3	Trace B S & W byPipe Line	3.34		30	F	N
780	1	4	M	-	380[b]	255	2	22.5	36	150	250	1922	--	--- --	-- --	--	--		28°-32° A P I					
	2	4	M	-	300	202	3	17.75	22	225	250	1926	--	--- --	-- --	--	--		1.5% S		NO			
	3	2	M	S	840	564	6	16	20	257	700	1928	--	--- --	-- --	--	--							
	4	2	M	S	720	484	6	16	20	257	600	1927	--	--- --	-- --	--	--							
	5	2	M	S	840	564	6	16	20	257	700	1928	--	--- --	-- --	--	--							
			Plant		3,080	2,069					2,500			3,280,300	450,000	7.29		--		3.07		4685	N	N
109	1	2	A	S	675	453	4	16.75	22	240	500	1927	3,066	614,280	-- --	--	44.2		24°-26° A P I					
	2	2	A	S	1,125	755	8	14.75	21	257	975	1929	2,159	959,500	-- --	--	58.8		190° SU @ 100°F		NO			
	3	2	A	S	1,125	755	8	14.75	21	257	975	1931[c]	65P	328,560	-- --	--	66.7		0.5% S ; 0.2% B S & W					
			Plant		2,925	1,963					2,450			1,902,340	208,377	9.13		54.1	Less than 1.0% Ash.	5.60		4300	N	N

TABLE IV - INFORMATION COVERING FUEL OIL (Page 2)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM - SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING - K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	GROSS OUTPUT - K.W. HRS.	FUEL OIL USED - GALLONS	GROSS K.W. HRS. PER GALLON OF FUEL OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	NATURE OF FUEL OIL USED - (SEE NOTES)	AVERAGE COST OF FUEL OIL-CENTS PER GALLON	IS FUEL CENTRIFUGED?	PLANT ALTITUDE - FEET ABOVE SEA LEVEL	PURPOSE FOR WHICH JACKET WATER HEAT IS UTILIZED. (SEE NOTES)	PURPOSE FOR WHICH EXHAUST HEAT IS UTILIZED (SEE NOTES)
7	1	4	M	-	360	242	6	11.5	16	277	375	1925	3,009	512,400	---	--	57.7		36°-40° A P I					
	2	4	M	-	360	242	6	11.5	16	277	375	1925	3,255	430,100	---	--	54.6		44.4° SU @ 100°F					
	3	4	M	-	360	242	6	11.5	16	277	375	1925	3,413	471,400	---	--	57.1		0.23% S ; 0.012% Ash.		YES			
	4	4	M	-	750	504	6	16	22	240	750	1927	1,667	481,200	---	--	57.3		0.016% CC					
	5	4	M	-	930	624	6	18	22	257	750	1930	4,101	1,487,300	---	--	58.1		1.39% BS & W					
	Plant				2,760	1,854					2,625			3,382,400	276,590	12.22		57.3		3.43		1230	F	N
732	1	4	A	-	250	168	4	13.75	17.5	257	200	1915	0	-- --	---	--	--		--		--			
	2	4	A	-	250	168	4	13.75	17.5	257	200	1916	1,825	-- --	---	--	--		--		--			
	3	4	A	-	520	349	4	19	24.5	200	500	1921	3,650	-- --	---	--	--		--		--			
	4	4	A	-	750	504	4	23	32	150	625	1923	4,745	-- --	---	--	--		--		--			
	5	4	A	-	750	504	4	23	32	150	625	1925	5,110	-- --	---	--	--	--	--		--			
	Plant				2,520	1,693					2,150			3,212,915	345,682	9.35		49.1		2.92		850	N	N
978	1	2	M	S	840	564	6	16	20	257	700	1931	--	-- --	---	---	--		--					
	2	2	M	S	840	564	6	16	20	257	700	1931	--	-- --	---	---	--		--					
	3	2	M	S	840	564	6	16	20	257	700	1931	--	-- --	---	---	--		--		NO			
	Plant				2,520	1,692					2,100			2,303,700	210,058	10.96	--	--		3.33		--	N	N
41	1	4	A	-	520	349	3	22	29.5	164	--	1922	4,023	888,530	92,910	9.56	63.3		20.4° A P I					
	2	4	A	-	700	470	4	22	29.5	164	--	1923	3,765	1,071,500	108,080	9.92	60.5		47° S U @ 100°F		NO			
	3	4	A	-	1,150	772	6	23	29.5	164	--	1928	3,862	2,182,700	201,062	10.85	73.2		1.3% S ; 0.15% CC					
	Plant				2,370	1,591								4,142,730	401,992	10.30		67.3		4.57		20	N	B
723	1	2	M	DC	300[d]	202	6	14	17	257	250	1925	460	74,100	---	--	79.8		32°-33° A P I					
	2	2	M	DC	300[d]	202	6	14	17	257	250	1925	440	66,100	---		74.3							
	3	2	M	S	720	484	6	16	20	257	600	1927	1,382	488,100	---		73.0				NO			
	4	2	M	S	1,050	705	7	16	20	257	910	1931	1,988	1,021,600	---		72.9							
	Plant				2,370	1,593					2,010			1,649,900	156,225	10.56		73.3		4.21		1206	N	N
79	1	4	A	-	225	151	3	16	24	164	198	1911	940	45,004	10,262	4.39	31.7		18°-20° A P I-Summer					
	2	4	A	-	225	151	3	16	24	164	198	1912	77	3,696	798	4.63	31.8		28°-30° " Winter					
	3	4	A	-	600	403	6	16.25	23	225	512	1929	4,881	1,342,100	110,074	12.19	68.2		64° SU @ 100°F		NO			
	4	4	A	-	520	349	4	19	24.5	200	438	1923	2,805	688,300	61,394	11.21	70.3		0.404% S ; 1.12% CC					
	5	4	A	-	600	403	6	16.25	23	225	512	1925	4,861	1,299,500	109,337	11.88	66.3		Trace Ash & BS & W					
	Plant				2,170	1,457					1,858			3,378,600	291,865	11.57		66.8		2.62		370	N	N
5	1	2	M	S	720	484	6	16	20	257	--	1926	--	-- --	---	--	--		--		--			
	2	2	M	S	720	484	6	16	20	257	--	1927	--	-- --	---	--	--		--		--			
	3	2	M	S	720	484	6	16	20	257	--	1927	--	-- --	---	--	--		--		--			
	Plant				2,160	1,452								1,918,770	207,547	9.25		--		3.33		--	-	-
80	1	4	A	-	365	245	4	16.25	21	225	--	1921	2,105	-- --	---	--	--		27° A P I					
	2	4	A	-	520	349	4	19	24.5	200	--	1921	2,105	-- --	---	--	--		0.65% S		NO			
	3	4	A	-	600	403	6	16.25	23	225	--	1929	3,272	-- --	---	--	--							
	4	4	M	-	665	447	4	21	30	180	--	1926	4,625	-- --	---	--	--		Partly by Tank Wagon					
	Plant				2,150	1,444								2,587,100	255,781	10.12		55.9		2.41		1134	N	B
6	1	2	M	DC	300[d]	202	6	14	17	257	--	1926	--	-- --	---	--	--				--			
	2	4	A	-	600	403	6	17	25	200	--	1926	--	-- --	---	--	--				--			
	3	4	A	-	600	403	6	17	25	200	--	1927	--	-- --	---	--	--				--			
	4	4	A	-	600	403	6	17	25	200	--	1928	--	-- --	---	--	--				--			
	Plant				2,100	1,411								1,651,300	138,444	11.92		--		4.12		- -	-	-
68	1	2	M	DC	240	161	4	14	17	257	300	1925	2,635	231,300	27,770	8.33	54.5		28°-32° A P I					
	2	2	M	DC	360	242	6	14	17	257	300	1925	3,125	442,600	49,790	8.89	58.5				NO			
	3	2	M	S	560	376	4	16	20	257	470	1923	4,128	1,174,400	117,990	9.95	70.5							
	4	2	M	S	900	604	6	16	20	257	774	1930	3,834	1,637,400	131,960	12.41	70.7							
	Plant				2,060	1,383								3,485,700	327,510	10.64		67.5		3.37		332	N	N
728	1	4	A	-	600	403	6	16.25	23	225	512	1926	4,100	1,274,700	---	--	77.1		24.6°A P I ; 0.76% S					
	2	4	A	-	600	403	6	16.25	23	225	512	1926	3,900	1,216,800	---	--	77.4		110" SU @ 100°F		NO			
	3	2	M	S	840	564	6	16	20	257	700	1929	3,800	1,109,830	---	--	51.9		0.17% CC;0.08% Ash.					
	Plant				2,040	1,370					1,724			3,601,330	333,650	10.78		67.1		Trace BS & W	5.20		8	N	N
93	1	2	M	S	700	470	5	16	20	257	--	1928	6,115	1,478,600	---	--	51.5		--					
	2	2	M	S	700	470	5	16	20	257	--	1928	6,002	1,457,100	---	--	51.7		--		NO			
	3	2	M	S	560	376	4	16	20	257	--	1928	3,406	716,100	---	--	55.9		--					
	Plant				1,960	1,316								3,651,800	410,083	8.91		52.3		2.79		15	N	N
215	1	2	M	S	980	658	7	16	20	257	845	1930	1,566	917,000	---	--	89.0		33.5° A P I		NO			
	2	2	M	S	980	658	7	16	20	257	845	1930	1,141	684,100	---	--	91.2		0.2% S ; 0.3% Ash					
	Plant				1,960	1,316					1,690			1,601,100	136,454	11.73		89.9		3.97		2150	N	N
837	1	4	A	-	750	504	4	23	32	150	625	1922	7,637	1,933,280	217,674	8.98	50.2		By Tank Wagon					
	2	4	A	-	450	302	6	16	24	164	413	1918	492											
	3	4	A	-	225	151	3	16	24	164	200	1904	1,024								NO			
	4	4	A	-	225	151	3	16	24	164	200	1904	1,101	330,954	48,024	6.89	56.3							
	5	4	A		225	151	3	16	24	164	187	1906	774											
	Plant				1,875	1,259					1,625			2,264,234	265,698	8.53		51.1		2.72		5	N	F
997	1	4	A	-	600	403	6	16.25	23	225	475	1929	7,889	---	---	--	--		25.°0 A P I					
	2	4	A	-	600	403	6	16.25	23	225	475	1929	7,976	---	---	--	--		1.1% S ; 0.041% Ash		NO			
	3	4	A	-	600	403	6	16.25	23	225	475	1929	7,286	---	---	--	--		Water-Trace					
	Plant				1,800	1,209					1,425			7,210,320	624,168	11.54		75.5		0.15% Sediment	5.94		7130	F	B
212	1	4	A	-	875	588	8	17.5	24	200	671	1929	245	105,500	9,659	10.92	73.3		35°A P I ; 0.31% S		NO			
	2	4	A	-	875	588	8	17.5	24	200	671	1929	316	123,900	11,631	10.66	66.7		Tank Wagon 3 months out					
	Plant				1,750	1,176					1,342			229,400	21,290	10.78		69.6		of 12	4.43		585	N	N
722	1	2	M	DC	200[d]	134	4	14	17	257	170	1923	282	27,870	---	--	73.7		32°-33° A P I					
	2	2	M	DC	300[d]	202	6	14	17	257	250	1923	392	62,760	---	--	79.3				NO			
	3	2	M	S	840	564	6	16	20	257	700	1929	1,830	731,400	---	--	70.8							
	4	4	A	-	365	245	4	16.25	21	225	330	1916	729	144,900	---	--	81.1							
	Plant				1,705	1,145					1,450			996,930	89,832	10.76		72.8		4.26		1285	N	N
735	1	2	M	S	840	564	6	16	20	257	700	1928	3,256	1,517,600	---	--	82.6		34°-36° A P I		NO			
	2	2	M	S	840	564	6	16	20	257	700	1928	3,256	1,341,800	---	--	73.1							
	Plant				1,680	1,128					1,400			2,859,400	333,503	8.57		77.8		3.33		--	N	N

TABLE IV - INFORMATION COVERING FUEL OIL (Page 3)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM - SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	RATED B.H.P.	EQUIVALENT K.W.-90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED - R.P.M.	GENERATING RATING K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	GROSS OUTPUT - K.W. HRS.	FUEL OIL USED - GALLONS	GROSS K.W. HRS. PER GALLON OF FUEL OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	NATURE OF FUEL OIL USED - (SEE NOTES)	AVERAGE COST OF FUEL OIL - CENTS PER GALLON	IS FUEL CENTRIFUGED?	PLANT ALTITUDE - FEET ABOVE SEA LEVEL	PURPOSE FOR WHICH JACKET WATER HEAT IS UTILIZED - (SEE NOTES)	PURPOSE FOR WHICH EXHAUST HEAT IS UTILIZED (SEE NOTES)
18	1 2	4	A	-	450	302	5	15.25	16	275	--	1922	--	---	---	--			--					
	2 4	A	-		600	403	6	17	25	200	--	1926	--	---	---	--			--					
	3 4	A	-		600	403	6	17	25	200	--	1928	--	---	---	--			--					
	Plant				1,650	1,108					---			2,502,301	239,705	10.44	--	--		3.94		--	--	-
8	1 4	4	A	-	400	269	4	16.5	24	200	--	1921	4,958	592,430	---	--	44.9		28°-30° A P I					
	2 4	A	-		560	376	3	22.5	22.5	225	--	1922	335	55,070	---	--	43.7				NO			
	3 4	A	-		600	403	6	17	25	200	--	1926	4,430	1,249,160	---	--	70.0							
	Plant				1,560	1,048					--			1,902,660	195,285	9.75		58.7		2.81		1160	B	B
10	1 4	4	A	-	750	504	4	22	29.5	166.7	625	1924	3,717	1,253,000	118,012	10.61	66.9		27° A P I;0.8% S					
	2 2	2	A	S	750	504	4	17	27	176.5	625	1925	4,881	1,772,700	159,301	11.12	72.0		40° SU @ 100°F		NO			
	Plant				1,500	1,008					1,250			3,025,700	277,313	10.89		69.9	1.5%cc;0.004% Ash	2.45		21	N	L
132	1 4	4	A	-	375	252	3	17.5	24.5	225	319	1929	4,479	709,600	64,699	10.96	62.9		34° A P I ; 0.5% S					
	2 4	4	A	-	500	336	4	17.5	24.5	225	432	1929	3,164	788,100	67,528	11.66	74.1		0.5% Ash.		NO			
	3 4	4	A	-	625	420	5	17.5	24.5	225	538	1929	1,351	407,600	35,576	11.45	71.8		1.0% BS & W					
	Plant				1,500	1,008					1,289			1,905,300	167,803	11.35		69.1	Trace CC	3.63		1100	N	B
289	1 4	4	A	-	250	168	4	13.75	17.5	257	210	1920	5,991	704,190	60,636	11.61	70.0		26°-28° A P I					
	2 4	4	A	-	250	168	4	13.75	17.5	257	210	1920	5,594	651,560	5u,429	11.53	69.3				NO			
	3 4	4	A	-	365	245	4	16.75	21	245	300	1915	3,175	527,890	45,115	11.69	67.8							
	4 4	4	A	-	600	403	6	16.25	23	225	512	1927	2,367	740,100	60,877	12.15	77.6							
	Plant				1,465	984					1,232			2,623,740	223,057	11.75		71.3		4.15		1670	F	N
865	1 4	4	A	-	225	151	3	16	24	164	160	1906[e]	--	4,970	---	--	--		14°-20°A P I					
	2 4	4	A	-	225	151	3	16	24	164	187	1911[e]	--	5,920	---	--	--		200°-750° SU @ 100°F		NO			
	3 4	4	A	-	500	336	4	18.875	28.375	164	475	1921	--	197,970	---	--	--		1.8% S					
	4 4	4	A	-	500	336	4	18.875	28.375	164	475	1917	--	171,510	---	--	--							
	Plant				1,450	974					1,297			380,370	36,297	10.47		--		1.53		2400	N	L
70	1 2	M	S		560	376	4	16	20	257	470	1928	3,451	844,865	---	--	65.1		32°-36° A P I					
	2 2	M	S		560	376	4	16	20	257	470	1928	4,129	925,650	---	--	59.6				NO			
	3 2	M	CC		300[d]	202	6	14	17	257	250	1920	2,200	272,585	---	--	61.3							
	Plant				1,420	954					1,190			2,043,100	214,950	9.51		62.0		2.83		1200	N	N
154	1 4	4	A	-	600	403	6	17	24	200	---	1927	--	-----	---	--	--		--		--			
	2 4	4	A	-	400	269	4	17	24	200	---	1926	--	-----	---	--	--		--		--			
	3 4	4	A	-	400	269	4	17	24	200	---	1925	--	-----	--	--	--		--		--			
	Plant				1,400	941					---			1,641,605	132,026	12.42		--		3.59		--	-	-
163	1 4	M		-	400	269	4	17.75	22	225	343	1924	5,306	1,054,400	89,015	11.83	73.9		36.1° A P I ; 0.263% S					
	2 4	M		-	400	269	4	17.75	22	225	343	1927	6,521	1,349,400	119,675	11.26	76.9		40°SU @ 100°F		NO			
	3 4	M		-	600	403	6	17.75	22	225	625	1929	5,514	1,601,600	136,072	11.76	77.2		0.034% CC					
	Plant				1,400	941					1,311			4,005,500	344,762	11.62		76.3	0.002% Ash-Trace by Pipe Line BS & W	1.92		1020	N	N
11	1 4	4	A	-	750	504	4	22	29.5	166.7	625	1924	233	103,940	9,445	11.00	88.5		27° A P I ; 0.5% S					
	2 4	4	A	-	600	403	6	16.5	24	200	513	1926	3,165	1,232,600	107,573	11.46	96.7		40° SUS 100°F;1.5% CC		NO			
	Plant				1,350	907					1,138			1,336,540	117,018	11.42		95.9	0.004% Ash	1.80		130	N	L
725	1 4	4	A	-	600	403	6	17	25	200	500	1926	---	---	---	--	--		32°-35°API					
	2 4	4	A	-	400	269	4	17	25	200	337	1925	---	---	---	--	--		0.3% S					
	3 2	M	CC		200[d]	134	4	14	17	257	170	1924	---	---	---	--	--				NO			
	4 2	M	CC		110[d]	74	2	14	17.5	257	90	1923	---	---	---	--	--							
	Plant				1,310	880					1,097			---	249,535	--		--		3.35		-	-	-
13	1 4	4	A	-	275	185	4	13.5	19.5	257	219	1923	6,509	743,100	71,644	10.36	61.7		32°-36°API					
	2 4	4	A	-	200	134	3	13.5	19.5	257	156	1923	3,402	298,700	28,795	10.36	65.5		42° SUS 100°F		NO			
	3 4	4	A	-	750	504	4	23	29.5	164	625	1925	1,574	388,200	45,974	8.45	49.0		0.6%S ; 0.02% CC					
	Plant				1,225	823					1,000			1,430,000	146,413	9.77		58.3	Trace Ash & BS & W	5.41		79	N	N
17	1 4	4	A	-	600	403	6	17	25	200	---	1926	---	---	---	--	--		--		--			
	2 4	4	A	-	600	403	6	17	25	200	---	1926	---	---	---	--	--		--		--			
	Plant				1,200	806					---			2,258,800	212,601	10.62		--		3.97		-	-	-
94	1 4	4	A	-	600	403	6	17	24	200	---	1927	1,437	374,330	---	--	64.6		--					
	2 4	4	A	-	600	403	6	17	24	200	---	1927	1,374	401,080	---	--	72.4		--		NO			
	Plant				1,200	806					---			775,410	64,561	12.01		68.4		2.26		1000	N	N
333	1 4	4	A	-	1,200	806	6	20	24	300	1,000	1929	950	637,000	60,674	10.48	83.3	83.3	38°API	5.40	NO	4000	N	N
451	1 2	M	S		600	403	4	16	20	257	512	1931[f]	---	469,230	---	--	--		30°-32°API					
	2 2	M	S		600	403	4	16	20	257	512	1931[f]	---	466,030	---	--	--		36° SUS 100°F		NO			
	Plant				1,200	806					1,024			935,260	89,072	10.48		--	Less than 0.5%S	3.25		--	N	N
980	1 4	4	A	-	450	302	4	17	24.5	225	375	1927	---	744,300	---	--	--		--					
	2 4	4	A	-	750	504	6	17.5	24.5	225	625	1929	---	1,415,500	---	--	--		--		NO			
	Plant				1,200	806					1,000			2,159,800	204,023	10.58		--		5.32		15	N	N
616	1 4	4	A	-	300	202	3	16.5	24	200	250	1923	---	681,440	57,780	11.78	--		22°API					
	2 2	M	CC		100[d]	67	2	14	17	257	75	1920	---	10,000	1,000	10.00	--		Less than 0.5%S		NO			
	3 4	4	A	-	750	504	4	23	32	150	657	1925	---	1,078,710	96,882	11.10	--							
	Plant				1,150	773					982			1,768,150	155,662	11.35		--		4.50		20	N	P
95	1 2	M	S		560	376	4	16	20	257	---	1929	2,468	477,100	---	--	51.4		--					
	2 2	M	S		560	376	4	16	20	257	---	1929	2,334	459,550	---	--	52.4		--		NO			
	Plant				1,120	752					---			936,650	107,459	8.72	--	51.9		2.36		746	N	N
96	1 4	M		-	600	403	6	17	24	225	---	1925	1,220	230,100	---	--	46.8		--					
	2 4	M		-	300	202	3	17	24	225	---	1923	6,984	751,600	---	--	53.2		--		NO			
	3 2	M	CC		200[d]	134	4	14	17	257	---	1921	1,041	56,200	---	--	40.3		--					
	Plant				1,100	739					---			1,037,900	104,222	9.95		50.9		2.74		605	N	N
721	1 2	M	CC		360	242	6	14	17	257	300	1925	---	---	---	--	--		28°-32°API					
	2 2	M	CC		360	242	6	14	17	257	300	1925	---	---	---	--	--		1.5% S		NO			
	3 2	M	CC		360	242	6	14	17	257	300	1926	---	---	---	--	--							
	Plant				1,080	726					900			777,700	138,500	5.61		--		3.22		2500	N	N
78	1 4	M		-	400	269	4	17	24	225	300	1928	6,134	560,474	78,440	7.14	34.0		25.2°API ; 2.43%S					
	2 4	M		-	600	443	6	17	24	240	500	1930	6,016	1,445,360	142,585	10.16	54.3		46° SUS 100°F		NO			
	Plant				1,060	712					800			2,005,834	221,025	9.08		46.5	0.007% Ash. 0.07% BS & W	5.07		173	N	N

TABLE IV - INFORMATION COVERING FUEL OIL (Page 4)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM - SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED - R.P.M.	GENERATING RATING K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	GROSS OUTPUT - K.W. HRS.	FUEL OIL USED - GALLONS	GROSS K.W. HRS. PER GALLON OF FUEL OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	NATURE OF FUEL OIL USED - (SEE NOTES)	AVERAGE COST OF FUEL OIL - CENTS PER GALLON	IS FUEL CENTRIFUGED?	PLANT ALTITUDE - FEET ABOVE SEA LEVEL	PURPOSE FOR WHICH JACKET WATER HEAT IS UTILIZED (SEE NOTES)	PURPOSE FOR WHICH EXHAUST HEAT IS UTILIZED (SEE NOTES)
83	1 4 A	-	600	403	6	16.25	23	225	520	1929	4,790	1,156,900	103,990	11.11	59.8		30° API ; 0.5% S					
	2 4 A	-	400	269	4	16.25	23	225	340	1927	5,042	632,500	59,620	10.60	46.6		50° SU @ 1000°F		NO			
	Plant		1,000	672					860			1,788,400	163,610	10.92		54.4	Trace Ash & BS & W	6.00		12	N	N
98	1 2 M	CC	360	242	6	14	17	257	--	1926	3,328	526,800	--	--	65.4		Tank Wagon					
	2 2 M	CC	360	242	6	14	17	257	--	1929	3,329	523,500	--	--	65.0		--		NO			
	3 2 M	CC	180	121	3	14	17	257	--	1926	5,180	421,900	--	--	67.3		--					
	4 2 M	CC	100[d]	67	2	14	17	257	--	--	0	0	--	--	--		--					
	Plant		1,000	672								1,472,200	190,442	7.73		65.8		3.44		2931	N	N
738	1 4 A	-	1,000	672	6	22	32	150	720	1918[e]	2,249	1,151,600	125,965	9.14	76.3	76.3	22°-26° A P I	3.00	NO	300	N	N
801	1 2 A	S	1,000	672	4	20.75	26	180	937	1927	8,390	4,778,300	456,514	10.46	84.9	84.9	500° SU @ 100°F 1.0% S	3.31	YES	558	N	N
1061	1 2 M	S	980	658	7	16	20	257	975	1930	3,822	2,048,030	178,296	11.48	81.4	81.4	28°-36° API;0.92% S 44" SUS @ 100°F;0.02%CC	3.93	NO	1200	N	N
863	1 4 A	-	300	202	3	17	24	200	250	1925	4,806	753,800	70,697	10.65	78.1		22°-24°API ; 0.75% S					
	2 4 A	-	675	453	6	17	24	225	625	1927	5,484	2,100,700	177,612	11.82	84.5		40" SUS 100°F ; 1.0% CC		NO			
	Plant		975	655					875			2,854,500	248,309	11.49		82.7	1.0% Ash ; 0.5% BS & W	3.50		932	F	S
49	1 4 A	-	485	326	4	18	24	200	420	1924	4,156	689,270	70,301	9.80	50.9		18°-22° API		NO			
	2 4 A	-	485	326	4	18	24	200	420	1924	4,216	814,170	82,969	9.82	51.9							
	Plant		970	652					840			1,503,440	153,270	9.81		51.5		3.97		20	N	N
53	1 2 M	CC	900[d]	134	4	14	17	257	170	1923	2,581	240,560	---	--	69.6		23.5°API		NO			
	2 2 M	CC	300[d]	202	6	14	17	257	250	1923	4,867	706,180	---	--	72.0		50" SUS 100°F					
	3 4 A	-	400	269	4	16.25	23	225	338	1925	7,927	1,506,880	---	--	70.7							
	Plant		900	605					758			2,453,620	271,633	9.04		71.0		2.65		1650	N	N
260	1 2 A	S	900	605	6	14.75	21	257	750	1928	263	80,593	8,660	9.31	50.6	50.6	24°-28° API	5.00	YES	5000	N	N
74	1 2 M	CC	225	151	3	15.25	16	277	187	1926	378	33,088[h]	3,850	8.59	56.0		Tank Wagon					
	2 4 M	-	260	175	4	14.5	18	277	219	1928	5,768	721,602[h]	56,117	12.85	71.5		32°-36° API		NO			
	3 4 M	-	400	269	6	14.5	18	277	344	1928	2,794	439,802[h]	42,054	10.44	58.4		30" SUS 100°F					
	Plant		885	595					750			1,193,492[h]	102,021	11.69		65.7	Less than 0.75% S	3.59		475	N	N
189	1 4 M	-	550	369	5	17	24	240	470	1929	8,606	993,706	112,216	8.85	31.3		28°-30° API					
	2 4 M	-	200[b]	134	2	17	24	200	170	1928	154	13,645	1,705	8.00	66.1				NO			
	3 4 M	-	100[b]	67	1	17	24	200	90	1915	0	0	0	--	--							
	Plant		850	570					730			1,007,351	113,921	8.84		31.5		3.68		1850	N	N
153	1 2 M	S	840	564	6	16	20	257	700	1929	--	594,940	56,540	10.52	--	--	--	3.41	-	---	-	-
494	1 2 M	S	840	564	6	16	20	257	700	1930	2,164	937,565	79,820	11.75	76.8	76.8	28°-30° API	3.56	NO	1100	N	N
619	1 2 M	S	840	564	6	16	20	257	700	1929	2,090	873,880	71,586	12.19	74.1	74.1	28°-30° API	3.72	NO	1100	N	N
105a	1 2 M	S	840	564	6	16	20	257	700	1929	4,822	2,073,460	183,691	11.27	76.2	76.2	28°-36° API; 0.92% S 44" SUS@100°F 0;02% CC	3.84	NO	1200	N	N
731	1 4 A	-	825	554	4	25	32	184	750	1926	1,825	618,602	60,119	10.28	61.2	61.2	30°-32° API less than 0.5% S	4.84	NO	865	N	N
648	1 4 M	-	400	269	4	17	24	225	333	1928	2,555	553,000	49,500	11.16	80.5		28°-28° API		NO			
	2 4 M	-	400	269	4	17	24	225	333	1928	3,006	655,893	58,510	11.21	81.2							
	Plant		800	538					666			1,208,893	108,010	11.19		80.9		3.53		1100	N	N
100	1 2 M	CC	300[d]	202	6	14	17	257	--	1923	115	8,970	---	--	38.6	--						
	2 2 M	CC	240	161	4	14	17	257	--	1924	2,688	271,180	---	--	62.6	--			NO			
	3 2 M	CC	200[d]	134	4	14	17	257	--	1924	2,646	255,440	---	--	72.0	--						
	Plant		740	497								535,530	59,412	9.01		86.0		2.79		1286	N	N
112	1 2 M	CC	200[d]	134	4	14	17	257	170	1925	1,798	180,200	20,120	8.96	74.6		34° API					
	2 2 M	CC	240	161	4	14	17	257	200	1927	6,930	777,900	63,872	12.17	69.7				NO			
	3 2 M	CC	300[d]	202	6	14	17	257	250	1925	2,293	336,400	37,262	9.03	73.3							
	Plant		740	497					620			1,294,500	121,254	10.67		71.3		3.42		845	N	N
67	1 2 M	CC	360	242	6	14	17	257	250	1925	1,860	369,790	35,505	10.41	82.2		26° API; 0.25% S					
	2 2 M	CC	360	242	6	14	17	257	250	1925	2,622	522,010	49,600	10.52	82.3		65" SUS 100°F		NO			
	Plant		720	484					500			891,800	85,105	10.47		82.3	0.037% CC	4.58		860	N	N
170	1 2 M	CC	360	242	6	14	17	257	300	1929	2,700	---	---	---	--							
	2 2 M	CC	360	242	6	14	17	257	300	1929	2,830	---	---	---	--				NO			
	Plant		720	484					600			863,979	91,234	9.47		64.3	34°API ; 0.4% S 42"-46" SUS@ 100° F Trace BS & W Tank Truck	6.20		50	N	N
527	1 2 M	CC	360	242	6	14	17	257	300	1930	5,000	450,000	--	--	37.2		28°-32° API; less than 0.5% S		NO			
	2 2 M	CC	360	242	6	14	17	257	300	1930	5,602	612,000	--	--	45.2		and 0.5% BS & W; 40"-90" SU @ 100°F	2.83		750	N	N
	Plant		720	484					600			1,062,000	110,470	9.63		41.4						
718	1 2 M	CC	360	242	6	14	17	257	300	1926	4,593	448,950	---	--	40.4		32°-36° API; 0.7% S		NO			
	2 2 M	CC	360	242	6	14	17	257	300	1926	4,467	312,750	---	--	28.9		36" SUS 100°F ; 0.01% CC					
	Plant		720	484					600			761,700	93,240	8.17		34.9	0.001% Ash;Trace BS & W	3.45		1200	N	N
19	1 2 M	CC	100[d]	67	2	14	17	257	--	--	--	---	---	--	--		--		-			
	2 2 M	CC	240	161	4	12.5	13.25	325	--	1927	--	---	---	--	--		--		-			
	3 2 M	CC	360	242	4	15.25	18	275	--	1927	--	---	---	--	--		--		-			
	Plant		700	470								747,510	97,972	7.63		--		3.73		---	-	-
368	1 4 A	-	400	269	4	16.25	23	225	340	1925	6,250	1,375,600	110,940	12.39	81.8		26°- 28° API; 0.648% S		NO			
	2 4 A	-	300	202	3	16.25	23	225	250	1929	4,750	921,500	70,170	12.09	96.0		0.026% CC; Trace BS & W					
	Plant		700	471					590			2,297,100	187,110	12.28		87.2	54" SU @ 100°F	5.00		1400	N	N
61	1 4 M	-	300	202	3	17.75	22	225	275	1926	7,759	1,083,600	---	--	69.5		26°API ; 0.25% S		NO			
	2 4 M	-	300	202	3	17.75	22	225	275	1928	3,541	476,100	---	--	66.9		45" SU @ 70°F					
	Plant		600	404					550			1,559,700	127,975	12.18		68.4	0.62% CC	4.80		620	T	N
106	1 4 A	-	600	403	6	17	25	200	--	1929	---	1,549,330	140,875	10.99	--	--	--	3.97	-	---	-	-
1094	1 4 A	-	600	403	6	16.25	23	225	513	1928	2,213	700,574	60,614	11.54	78.5	78.5	28°-30° API; 0.63% S 44" SF @ 77°F; 0.48% CC; 0.03% BS & W Trace Ash.	3.84	NO	1150	N	N
1096	1 4 A	-	600	403	6	16.25	23	225	513	1928	2,570	766,386	61,915	12.37	74.0	74.0	28°-30° API; 0.53% S; 3.65" SF @ 77°F ;0.03% BS&W Trace Ash.	3.88	NO	1150	N	N
103	1 2 M	CC	240	161	4	14	17	257	--	1927	5,497	505,150	---	--	57.1		--					
	2 2 M	CC	200[d]	134	4	14	17	257	--	1920[i]	3,731	325,500	---	--	65.1		--		NO			
	3 2 M	CC	150[d]	101	3	14	17	257	--	1923	4,353	232,080	---	--	52.8		--					
	Plant		590	398								1,062,730	142,180	7.47		58.3		3.31		3100	N	N

TABLE IV - INFORMATION COVERING FUEL OIL (Page 5)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM-SEE NOTES	SCAVENGING SYSTEM-SEE NOTES	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING - K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	GROSS OUTPUT - K.W. HRS.	FUEL OIL USED - GALLONS	GROSS K.W. HRS. PER GALLON OF FUEL OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	NATURE OF FUEL OIL USED (SEE NOTES)	AVERAGE COST OF FUEL OIL-CENTS PER GALLON	IS FUEL CENTRIFUGED?	PLANT ALTITUDE - FEET ABOVE SEA LEVEL	PURPOSE FOR WHICH JACKET WATER HEAT IS UTILIZED (SEE NOTES)	PURPOSE FOR WHICH EXHAUST HEAT IS UTILIZED (SEE NOTES)
862	1 2 M	2	CC	75[d]	50	2	12	15	300	60	1920	89	---	---	---	---		34°-36° API					
	2 2 M	2	CC	75[d]	50	2	12	15	300	60	--	242	---	---	---	---							
	3 2 M	2	CC	180	121	3	14	17	257	150	1925	4,823	---	---	---	---				NO			
	4 4 A	-		258	173	3	16.5	24.5	180	225	1918[j]	4,847	---	---	---	---							
	Plant			588	394					495			676,320	74,683	9.05		46.9		3.27		221	N	N
102	1 2 M	2	CC	240	161	4	14	17	257	--	1928	3,778	285,570	--	--	46.9		--					
	2 2 M	2	CC	240	161	4	14	17	257	--	1928	3,517	282,220	--	--	49.9		--		NO			
	3 2 M	2	CC	100[d]	67	2	14	17	257	--	1928[j]	2,158	93,940	--	--	65.0		--					
	Plant			580	389								661,730	89,158	7.42		50.1		2.49		565	N	N
101	1 2 M	2	CC	240	161	4	14	17	257	--	1926	5,024	435,515	--	--	53.8		--					
	2 2 M	2	CC	150[d]	101	3	14	17	257	--	1919	1,600	86,430	--	--	53.4		--		NO			
	3 2 M	2	CC	150[d]	101	3	14	17	257	--	1923	3,933	239,680	--	--	60.3		--					
	Plant			540	363								761,615	93,807	8.12		55.7		2.34		992	N	N
246	1 2 M	2	CC	100[d]	67	2	14	17	257	60	1918	1,990	- -	6,766	--	--		36°-40°API					
	2 2 M	2	CC	200[d]	134	4	14	17	257	170	1922	4,720	- -	29,189	--	--				NO			
	3 2 M	2	CC	240	161	4	14	17	257	200	1929	2,200	- -	23,276	--	--							
	Plant			540	362					430			427,085	59,231	7.21		38.0		6.29		5000	B	N
540	1 2 M	2	CC	360	242	6	14	17	257	300	1929	--	- -	- -	--	--		--					
	2 2 M	2	CC	180	121	3	14	17	257	125	1926	--	- -	- -	--	--		--		NO			
	Plant			540	363					425			750,000	88,730	8.45		--		3.30		725	-	-
772	1 4 A	-		250	168	4	13.75	17.5	275	250	1916	889	111,231	- -	--	74.5		28°-30° API					
	2 4 A	-		250	168	4	13.75	17.5	275	263	1916	977	122,300	- -	--	74.5				NO			
	Plant			500	336					513			233,531	21,940	10.63		74.5		3.72		1100	N	N
501	1 2 M	2	CC	120	81	2	14	17	257	90	1925	4,896	128,038	20,649	6.20	32.3		370-40° API					
	2 2 M	2	CC	120	81	4	14	17	257	90	1925	4,829	128,975	21,152	6.09	33.3				NO			
	3 2 M	2	CC	240	161	4	14	17	257	200	1929	2,625	216,700	21,904	9.90	51.3							
	Plant			480	323					380			473,713	63,705	7.43		39.3		3.98		1106	N	N
615	1 2 M	2	CC	100[d]	67	2	14	17	257	75	1922	4,130	--	---	--	--		--					
	2 2 M	2	CC	100[d]	67	2	14	17	257	75	1922	490	--	---	--	--		--		NO			
	3 2 M	2	CC	240	161	4	14	17	257	219	1930	4,995	--	---	--	--		--					
	Plant			440	295					369				51,390	--	--	---		6.00		140	N	N
931	1 2 A	S		440	295	4	15	20	225	--	1922	3,706	814,915	75,185	10.82	74.6	74.6	29.8° API ; 0.8% S 39" SU @ 100°F; ash nil	4.60	NO	23	N	N
335	1 4 A	-		250	168	4	13.75	17.5	257	219	1924	6,310	362,040	36,204	10.00	34.2		0.05% CC; Trace BS & W 32°-36° API					
	2 4 A	-		180	121	4	11.75	15	300	150	1919	2,450	141,960	15,430	9.20	47.9		0.5 % S		NO			
	Plant			430	289					369			504,000	51,634	9.76		37.3		3.82		867	N	N
24	1 4 M	-		200	134	3	14.5	18	277	163	1925	4,531	287,300	--	--	47.3		32°-36° API					
	2 4 M	-		200	134	3	14.5	18	277	163	1925	4,531	285,400	--	--	47.0				NO			
	Plant			400	268					326			572,700	67,530	8.55		47.1		2.82		760	N	N
194	1 4 M	-		400	269	4	17	24	225	--	1928	4,115	384,250	42,900	8.96	34.7	34.7	20°-22°API;Tank Wagon	3.26	NO	1000	N	N
247	1 2 M	2	CC	50[d]	34	1	14	17	257	40	1915	320	--	--	--	--							
	2 2 M	2	CC	100[d]	67	2	14	17	257	75	1920	3,650	--	--	--	--		29°-34° API		NO			
	3 2 M	2	CC	240	161	4	14	17	257	200	1928	5,115	---	--	--	--							
	Plant			390	262					315			481,150	62,647	7.68		44.6		4.62		5000	N	N
105	1 2 M	2	CC	150[d]	101	3	14	17	257	--	1922	4,580	255,060	---	--	55.1		--					
	2 2 M	2	CC	120	81	2	14	17	257	--	1926	2,668	74,960	---	--	34.7		--		NO			
	3 2 M	2	CC	100[d]	67	2	14	17	257	--	1923	2,670	75,560	---	--	42.2		--					
	Plant			370	249								405,580	55,728	7.28		47.4		2.67		1580	N	N
152	1 2 M	2	CC	360	242	6	14	17	257	--	1925	--	54,600	5,996	9.10	--	--	--	4.75	-	--	-	-
161	1 2 M	2	CC	360	242	6	14	17	257	300	1929	4,335	1,001,000	81,910	12.22	95.4	95.4	30.8° API; 0.535% S 44° SU @ 100° F ; 0.05% CC ; 0.05% BS & W	5.33	-	450	N	N
1091	1 2 M	2	CC	180	121	3	14	17	257	135	1929	4,380	191,300	--	--	36.1							
	2 2 M	2	CC	180	121	3	14	17	257	135	1930	4,380	191,500	--	--	36.2		38°API		NO			
	Plant			360	242					270			382,800	53,100	7.21		36.2		5.33		5000	N	N
688	1 2 M	2	CC	50[d]	34	1	14	17	257	-	1923	---	--	--	--	--		36°API					
	2 2 M	2	CC	120	81	2	14	17	257	-	1926	---	--	--	--	--				NO			
	3 2 M	2	CC	180	121	3	14	17	257	-	1927	---	--	--	--	--							
	Plant			350	236								--	54,480			--		5.00		870	N	N
365	1 4 A	-		165	111	4	11.75	15	277	110	1917	--	262,700	27,799	9.45	--		--					
	2 4 A	-		180	121	4	11.75	15	300	120	1922	--	290,800	30,051	9.69	--		--		NO			
	Plant			345	232					230			553,500	57,850	9.57			--	3.85		--	N	N
774	1 4 A	-		165	111	4	11.75	15	277	135	1917	972	73,600	8,860	8.31	68.3		28°-30° API					
	2 4 A	-		180	121	4	11.75	15	300	135	1920	525	39,647	4,846	8.18	62.4		0.2-0.9% S		NO			
	Plant			345	232					270			113,247	13,706	8.27		66.2	Trace BS & W	3.79		1100	N	N
265	1 2 M	2	CC	60	40	1	14	17	257	48	1927	2,555	---	--	--	--		38°API					
	2 2 M	2	CC	120	81	2	14	17	257	90	1928	6,905	---	--	--	--				NO			
	3 2 M	2	CC	140	94	2	14	17	300	111	1931[k]	400	---	--	--	--							
	Plant			320	215					249			---	32,745	--		--		7.79		940	N	N
27	1 2 M	2	CC	75[d]	50	2	12	15	300	--	1926	--	---	--	--	--		--			-		
	2 2 M	2	CC	240	161	4	12.5	13.25	325	--	1926	--	---	--	--	--		--			-		
	Plant			315	211								229,080	39,944	5.73		--		4.06		--	-	-
382	1 4 M	-		150	101	4	9.5	14	300	--	1927	3,279	67,900	--	--	20.5		36°-40° API ; 0.1% S		NO			
	2 4 M	-		150	101	4	9.5	14	300	--	1927	1,135	23,400	--	--	20.4		40°-50° SUG 100°F Trace CC					
	Plant			300	202								91,300	9,418	9.69		20.5	Ash nil BS & W not over 1.0%	3.35		875	BP	N

TABLE IV - INFORMATION COVERING FUEL OIL (Page 6)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM - SEE NOTES	SCAVENGING SYSTEM - SEE NOTES	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	NUMBER OF CYLINDERS	CYLINDER BORE - INCHES	STROKE - INCHES	RATED R.P.M.	GENERATOR RATING K.V.A.	YEAR ENGINE STARTED TO WORK	ENGINE HOURS OPERATED IN REPORTED PERIOD	GROSS OUTPUT - K.W. HRS.	FUEL OIL USED - GALLONS	GROSS K.W. HRS. PER GALLON OF FUEL OIL	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	NATURE OF FUEL OIL USED (SEE NOTES)	AVERAGE COST OF FUEL OIL - CENTS PER GALLON	IS FUEL CENTRIFUGED?	PLANT ALTITUDE - FEET ABOVE SEA LEVEL	PURPOSE FOR WHICH JACKET WATER HEAT IS UTILIZED (SEE NOTES)	PURPOSE FOR WHICH EXHAUST HEAT IS UTILIZED (SEE NOTES)
1129	1	2 M	CC	b 75[d]	50	2	10.5	14	257	72	--	3,337	52,515	---	--	31.5		34°-36°API Summer					
	2	2 M	CC	b 75[d]	50	2	10.5	14	257	72	--	75	1,035	---	--	27.6		38°-40° " Winter		NO			
	3	2 M	CC	150[d]	101	3	14	17	257	125	1922	5,402	159,250	---	--	29.2							
	Plant			300	201					269			212,800	37,444	5.68		29.7		4.02		4000	N	N
156	1	2 M	CC	150[d]	101	3	14	17	257	125	1929[l]	---	---	---	--	--		28°-32°API		NO			
	2	2 M	CC	120	81	2	14	17	257	90	1929	---	---	---	--	--		1.5% S					
	Plant			270	182					215			199,500	41,500	4.81		--		3.18		3000	N	N
1097	1	2 M	CC	150[d]	101	3	14	17	257	125	1924	---	193,776	22,008	8.80	--		30°API ; 0.5% S		-			
	2	2 M	CC	120	81	2	14	17	257	90	1928	---	104,488	13,031	8.02	--		0.5% CC ; 0.03% Ash		-			
	Plant			270	182					215			298,264	35,039	8.52		--	Trace BS & W	5.50		1500	N	B
169	1	4 M	-	240	161	6	10.75	13.75	327	200	1928	2,793	191,340	20,684	9.25	42.5	42.5	34°API; 0.315 % S less than 40" SU @ 100°F 0.1% BS & W	5.56	NO	20	B	N
516	1	2 M	CC	120	81	2	14	17	257	94	1926	--	---	---	--	--							
	2	2 M	CC	120	81	2	14	17	257	94	1930	--	---	---	--	--		32°-36°API		NO			
	Plant			240	162					188			---	31,000	--		--		4.30		--	N	N
220	1	2 M	CC	60	40	1	14	17	257	48	1930	--	---	---	--	--		28°-32°API					
	2	2 M	CC	120	81	2	14	17	257	90	1930	--	---	---	--	--		1.5% S		NO			
	Plant			180	121					138			44,000	20,190	2.18		--		3.46		3500	N	N
733	1	2 M	CC	100	67	2	12.5	13.25	327	60	1922	41	1,860	349	5.33	67.7		Tank Wagon					
	2	2 M	CC	75[d]	50	2	12	15	300	60	1914	52	1,500	274	5.47	57.7				NO			
	Plant			175	117					120			3,360	623	5.39		62.5		5.78		850	N	N
646	1	2 M	CC	120	81	2	14	17	257	90	1927	1,789	23,340	5,009	4.66	16.1	16.1	32°-34°API	7.00	NO	600	N	N
201	1	2 M	CC	50[d]	34	2	10.5	12	375	63	1924	--	---	---	--	--		32°-34°API					
	2	2 M	CC	50[d]	34	1	14	17	257	40	1927	--	---	---	--	--		1.5% S		NO			
	Plant			100	68					103			71,640	19,700	3.64		--	Tank wagon	7.45		1000	N	N

NOTES:

INJECTION SYSTEM

A- Air
M- Mechanical

SCAVENGING SYSTEM (2 Stroke Cycle Only)

S- Separate (i.e., Pump or blower)

CC-Crank Case

HEAT UTILIZATION

N- not utilized

B- To heat building

F- To heat Fuel Oil

L- To heat Lubricating Oil (before purification)

BF-To heat Boiler Feed Water

T- Thawing Ice Cans

NATURE OF FUEL

API- American Petrolium Institute

"S.U.- Seconds Saybolt Universal
"S.F.- " " Furol

S- Sulphur

CC- Carbon by Conradson Method

BS & W- Bottom Sediment & Water

Asph.-Asphaltic Residue

LETTERED NOTES

a- No deduction made for motor-driven Scavenging Blower

b- Horizontal Engine

c- November 5th

d- Semi-Diesel Engine

e- Date of original installation; date installed in this plant unknown

f- May 18th

g- Date of original installation; installed in this plant in 1924

h- Increments over net output estimated by operator

i- Date of original installation; installed in this plant in 1930

j- Date of original installation; installed in this plant in 1928

k- November 9th

l- Date installed in this plant; date of original installation not known

TABLE V - INFORMATION COVERING MAINTENANCE AND OPERATING LABOR (Page 1)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM-SEE NOTES	SCAVENGING SYSTEM-SEE NOTES	NUMBER OF CYLINDERS	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	YEAR ENGINE STARTED TO WORK	PEAK LOAD DURING REPORTED PERIOD - K.W.	B.MEP. AT RATED B.H.P. - LBS. PER SQ. IN.	B.MEP. AT PEAK LOAD (90% GENERATING EFFICIENCY)	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	PISTON COOLING-SEE NOTES	ARE AIR FILTERS USED?	AVERAGE LINER - WEAR PER 1000 HRS. OPERATION - INS.	OPERATING PERIOD OVER WHICH LINER WEAR WAS MEASURED - HRS.	TYPE COOLING SYSTEM (SEE TEXT)	COST OF REGULAR ENGINE UPKEEP - TOTAL DOLLARS MATERIAL - EXTRA LABOR	COST OF REPAIRS FOR ENGINE ACCIDENTS - TOTAL DOLLARS - MATERIAL - EXTRA LABOR	TOTAL ENGINE REPAIRS IN DOLLARS PER RATED B.H.P. PER YEAR	NUMBER OF ENFORCED SHUTDOWNS	TOTAL DURATION OF ENFORCED SHUTDOWNS - HOURS	NUMBER OF SHIFTS PER DAY	NUMBER OF HOURS PER SHIFT	NUMBER OF ATTENDANTS PER SHIFT
43	1	4	A	-	3	500	336	1915	300	78.4	70.0	69.3		W	No											
	2	4	A	-	3	500	336	1916	300	78.4	70.0	68.4		W	No				381-891	0	1.27					
	3	4	A	-	4	520	349	1919	310	74.0	65.7	76.6		W	No											
	4	4	A	-	4	520	349	1920	320	74.0	67.8	76.8		W	No				361-715	0	1.03					
	5	4	A	-	6	1,000	671	1922	650	73.9	71.6	63.7		W	No											
	6	4	A	-	6	1,000	671	1925	650	73.9	71.6	67.3		W	No				876-1,775	0	1.33					
	7	2	A	S	6	3,750	2,517	1928	2,700	67.2	72.1	66.8		W	Yes	.0022	7,651									
	8	2	A	S	6	3,750	2,517	1929	2,800	67.2	74.8	80.5		W	Yes	.0014	8,156		7,767-8,071	0	1.84					
	Plant					11,540	7,746		7,100				73.2					B-E								
46	1	2	A	S	5	1,250	840	1922	---	62.5	--	72.0		W	No	--	---									
	2	2	A	S	5	1,250	840	1922	---	62.5	--	70.3		W	No	--	---									
	3	2	A	S	4	2,250	1,510	1922	---	66.2	--	78.1		W	No	--	---									
	4	2	A	S	4	2,250	1,510	1922	---	66.2	--	79.8		W	No	--	---		-13,768-		2.03					
	5	2	A	S	4	2,250	1,510	1930	---	66.2	--	81.4		W	No	.0046	4,122									
	Plant					9,250	6,210		5,400				78.6					B-E								
168	1	2	A	S	5	1,250	840	1918	---	62.5	--	76.5		W	No	--	---									
	2	2	A	S	5	1,250	840	1918	---	62.5	--	75.5		W	No	--	---									
	3	2	A	S	5	1,250	840	1919	---	62.5	--	78.5		W	No	--	---		-20,862-		2.45					
	4	2	A	S	5	1,250	840	1916	---	62.5	--	77.5		W	No	--	---									
	5	2	A	S	5	1,250	840	1918	---	62.5	--	77.3		W	No	--	---									
	6	2	A	S	4	2,250	1,510	1929	---	66.2	--	97.1		W	No	.0075	6,859									
	Plant					8,500	5,710		5,000				83.5					B-E								
82	1	4	A	-	4	360	242	1921	---	72.7	--	63.9		A	No	.0039	963		264-1,238	0	4.17	0	---			
	2	2	A	S	4	750	504	1923	---	67.3	--	57.5		W	Yes	.0040	1,410		587-2,184	436-870	5.04	1	24			
	3	2	A	S	4	750	504	1924	---	67.3	--	58.3		W	Yes	.0040	909		205-957	0	1.55	0	---			
	4	2	A	S	6	1,150	772	1927	---	68.8	--	68.5		W	Yes	.0040	3,077		528-2,487	0	2.62	0	---			
	5	2	A	S	6	1,150	772	1927	---	68.8	--	67.2		W	Yes	.0040	3,489		442-2,100	0	2.21	0	---			
	6	2	A	S	8	1,500	1,007	1929	---	67.3	--	72.2		W	Yes	.0040	6,466		909-4,286	0	3.46	2	1/3	1	8	4
	Plant					5,660	3,801		3,080				69.0					B-E							2	8	2
91	1	4	M	-	6	1,125	755	1927	---	72.3	--	58.0		A	Yes	--	---		---	---	--	--	---			
	2	4	M	-	6	1,125	755	1928	---	72.3	--	58.4		A	Yes	--	---		---	---	--	--	---			
	3	4	M	-	6	600	403	1927	---	64.6	--	57.6		A	Yes	--	---		---	---	--	--	---			
	4	4	M	-	6	600	403	1927	---	64.6	--	57.0		A	Yes	--	---		---	---	--	--	---			
	5	4	A	-	6	1,800	1,208	1930	---	72.9	--	66.8		O	Yes	--	---		---	---	--	--	---			
	Plant					5,250	3,524		1,950[a]				59.3					D							3	8	1
73	1	2	A	S	5	1,250	840	1923	---	62.5	--	64.0		W	Yes	--	---					0	---			
	2	2	A	S	5	1,250	840	1924	---	62.5	--	66.9		W	Yes	--	---					0	---			
	3	2	A	S	5	1,250	840	1928	---	62.5	--	69.8		W	Yes	--	---		1,800-0	0	0.36	0	---			
	4	2	A	S	5	1,250	840	1928	---	62.5	--	64.0		W	Yes	--	---					0	---			
	Plant					5,000	3,360		2,400				66.3					C D E							1 2	8 8	4 2
92	1	2	A	S	5	1,250	840	1928	---	62.5	--	65.2		W	Yes	--	---		---	---	--	--	---			
	2	2	A	S	5	1,250	840	1928	---	62.5	--	71.1		W	Yes	--	---		---	---	--	--	---			
	3	2	A	S	8	2,400	1,612	1930	---	63.5	--	61.2		W	Yes	--	---		---	---	--	--	---			
	Plant					4,900	3,292		2,100[a]				63.5					D							3	8	1
129	1	2	A	S	5	1,250	840	1918	---	62.5	--	67.1		W	No	--	---									
	2	2	A	S	5	1,250	840	1921	---	62.5	--	69.2		W	No	.0041	6,685		-2,666-		2.13	--	---			
	3	2	A	S	4	2,250	1,510	1930	---	66.2	--	85.6		W	No	.0043	6,724		-3,036-		2.43	--	---			
	Plant					4,750	3,190		3,100				77.1					B-E			-4,857-		2.16	--	---	1	8	3
																								2	8	2
52	1	4	A	-	4	830	557	1928	585	75.3	79.1	69.8		W	No	--	---			0		0	---			
	2	4	A	-	6	1,250	840	1928	885	75.7	79.8	80.1		W	No	--	---			0		0	---			
	3	4	A	-	6	1,250	840	1928	900	75.7	81.1	80.2		W	No	.0032	13,170		368-3,766[b]	0	0.90	0	---			
	4	4	A	-	6	1,250	840	1930	880	75.7	79.3	78.3		W	No	--	---			0		0	---			
	Plant					4,580	3,077		2,900				78.6					A							2	8	2
45	1	4	A	-	6	600	403	1923	350	77.2	67.0	45.2		A	Yes	.0012	843			0		0	---			
	2	4	A	-	6	600	403	1923	350	77.2	67.0	47.4		A	Yes	.0013	777			0		0	---			
	3	4	A	-	4	750	504	1923	400	74.5	59.1	60.5		W	Yes	.0023	440		873-1,879	0	1.67	0	---			
	4	4	A	-	6	1,150	772	1926	800	76.1	78.9	68.7		W	Yes	.0006	5,326			714-3,587[c]		0	---			
	5	4	A	-	6	1,150	772	1926	800	76.1	78.9	68.3		W	Yes	.0006	5,394			8-39[c]		0	---	3	8	2
	Plant					4,250	2,854		1,500				66.5					B							and 1-relief man		
54	1	4	A	-	6	600	403		---	69.8	--	--		A	No	--	---									
	2	4	A	-	6	600	403		---	69.8	--	--		A	No	--	---									
	3	4	A	-	6	600	403	1925	---	69.8	--	--		A	No	--	---									
	4	4	A	-	6	600	403	-26	---	69.8	--	--		A	No	--	---									
	5	4	A	-	6	600	403		---	69.8	--	--		A	No	--	---									
	6	4	A	-	6	600	403		---	69.8	--	--		A	No	--	---									
	7	4	A	-	6	600	403		---	69.8	--	--		A	No	--	---									
	Plant					4,200	2,821		2,275[d]				95.6					B-E							2	8	2
864	1	4	A	-	4	500	336	1915	---	76.0	--	--		A	No	--	---									
	2	4	A	-	4	500	336	1915	---	76.0	--	--		A	No	--	---									
	3	4	A	-	4	500	336	1915	---	76.0	--	--		A	No	--	---									
	4	2	A	S	5	1,250	840	1928	---	62.5	--	--		W	No	--	---									
	5	2	A	S	5	1,250	840	1928	---	62.5	--	--		W	No	--	---									
	Plant					4,000	2,688		2,310				--					B-E							2	8	3
130	1	4	A	-	6	600	403	1924	470	73.8	86.1	68.2		A	No	--	---		3,822-233[e]	0	6.74	0	---			
	2	4	A	-	6	600	403	1926	540	72.7	97.4	65.1		A	No	--	---		1,312-158	0	2.45	0	---			
	3	4	A	-	6	1,200	806	1929	1,000	70.1	87.0	65.9		O	Yes	--	---		530-57	22-1,749	1.97	1	2,160			
	4	4	A	-	6	1,200	806	1930	820	70.1	71.5	68.0		O	Yes	--	---		60-218	0	0.23	0	---			
	Plant					3,600	2,418		1,200				67.0					A							3	8	1
2	1	4	A	-	6	1,150	772	1925	---	74.3	--	97.7		W	No	.0044	8,018			0		20	31	1	8	5
	2	4	A	-	6	1,150	772	1925	---	74.3	--	97.8		W	No	.0045	7,597		14,032-6,516	0	5.95	21	54	2	8	2
	3	4	A	-	6	1,150	772	1925	---	74.3	--	98.0		W	No	.0044	7,978			0		16	22	also 1-relief operator		
	Plant					3,450	2,316		2,355				97.8					A							1- " oiler		

TABLE V - INFORMATION COVERING MAINTENANCE AND OPERATING LABOR (Page 2)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM-SEE NOTES	SCAVENGING SYSTEM-SEE NOTES	NUMBER OF CYLINDERS	RATED B.H.P.	EQUIVALENT K.W. - 80% GENERATING EFFICIENCY	YEAR ENGINE STARTED TO WORK	PEAK LOAD DURING REPORTED PERIOD - K.W.	B. MEP. AT RATED B.H.P. LBS. PER SQ. IN.	B. MEP. AT PEAK LOAD (80% GENERATING EFFICIENCY)	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	PISTON COOLING - SEE NOTES	ARE AIR FILTERS USED?	AVERAGE LINER - WEAR PER 1000 HRS. OPERATION - INS.	OPERATING PERIOD OVER WHICH LINER WEAR WAS MEASURED HRS.	TYPE COOLING SYSTEM (SEE TEXT)	COST OF REGULAR ENGINE LABOR & TOTAL DOLLARS MATERIAL - EXTRA LABOR	COST OF REPAIRS FOR ENGINE ACCIDENTS - TOTAL DOLLARS MATERIAL EXTRA LABOR	TOTAL ENGINE REPAIRS IN DOLLARS PER RATED B.H.P. PER YEAR	NUMBER OF ENFORCED SHUTDOWNS	TOTAL DURATION OF ENFORCED SHUTDOWNS - HOURS	NUMBER OF SHIFTS PER DAY	NUMBER OF HOURS PER SHIFT	NUMBER OF ATTENDANTS PER SHIFT
164	1 4 M	-		6	840	564	1926	550[d]	72.4	70.6	79.9			0	NO	.0014	6,099									
	2 4 M	-		6	840	564	1926	571[d]	72.4	73.3	75.3			0	NO	.0042	6,207									
	3 4 M	-		6	840	564	1926	579[d]	72.4	74.4	69.9			0	NO	.0027	6,149									
	4 4 M	-		6	840	564	192b	567[d]	72.4	78.9	81.0			0	NO	.0011	5,465									
	Plant				3,360	2,256		2,225[d]					76.3					A	--18, 760--	4.99		3	8	3
720	1 4 M	-		8	380[f]	255	1922	--	70.1					A	NO											
	2 4 M	-		8	300	202	192b	--	64.7					A	NO											
	3 2 M	S		6	840	564	1928	--	53.6					0	YES				--159--	0.05						
	4 2 M	S		6	720	484	1927	--	46.0					0	YES											
	5 2 M	S		6	840	564	1928	--	53.6					0	YES											
	Plant				3,080	2,059		1,100										B-E				3	8	2
109	1 2 A	S		4	675	453	1927	--	57.4	--	44.2			0	NO							2	15			
	2 2 A	S		8	1,125	755	1929	--	60.4	--	58.8			0	YES				1,292-192	0.81	3	0				
	3 2 A	S		8	1,125	755	1931[f]	--	60.4	--	68.7			0	YES							0				
	Plant				2,925	1,963		1,300					54.1					B				2	9	2	2½
7	1 4 M	-		6	360	242	1925	--	103.2	--	57.7			A	YES							0				
	2 4 M	-		6	360	242	1925	--	103.2	--	54.0			A	YES							0				
	3 4 M	-		6	360	242	1925	--	103.2	--	57.1			A	YES				981-1,575	983-392	1.43	0				
	4 4 M	-		6	750	504	1927	--	93.2	--	57.3			A	YES							0				
	5 4 M	-		6	930	624	1930	--	85.2	--	58.1			A	YES							1	120			
	Plant				2,760	1,854		879					57.3					D				3	8	2	
732	1 4 A	-		4	250	168	1912	--	74.1					A	YES											
	2 4 A	-		4	250	168	1916	--	74.1					A	YES											
	3 4 A	-		4	520	349	1921	--	74.0					W	YES											
	4 4 A	-		4	750	504	1923	--	74.5					W	YES				--3,492--	1.39						
	5 4 A	-		4	750	504	1925	--	74.5					W	YES											
	Plant				2,520	1,693		1,470					49.1					B				3	8	2	
978	1 2 M	S		6	840	564	1931	--	53.6					0	YES											
	2 2 M	S		6	840	564	1931	--	53.6					0	YES											
	3 2 M	S		6	840	564	1931	--	53.6					0	YES											
	Plant				2,520	1,692		1,750					--					D-E				-	-	-	
41	1 4 A	-		3	520	349	1922	400	74.6	85.5	63.3			W	NO											
	2 4 A	-		4	700	470	1923	500	75.3	80.1	60.5			W	NO											
	3 4 A	-		6	1,150	772	1928	900	75.5	88.0	73.2			W	NO				--965--	0.41						
	Plant				2,370	1,591		1,800					67.3					A				2	8	3	
723	1 2 M	CC		6	300[1]	202	1925	190	29.4	27.7	79.8			A	NO											
	2 2 M	CC		6	300[1]	202	192b	190	29.4	27.7	74.3			A	NO											
	3 2 M	S		6	720	484	1927	470	46.0	44.7	73.0			0	YES				--1,345--	0.57						
	4 2 M	S		7	1,050	705	1931	700	57.5	57.1	72.9			0	YES											
	Plant				2,370	1,593							73.3					C				1	8	2½	
79	1 4 A	-		3	225	151	1911	110	75.0	54.6	31.7			A	NO	-	-					2	8	1		
	2 4 A	-		3	225	151	1912	110	75.0	54.6	31.8			A	NO	-	-									
	3 4 A	-		6	600	403	1929	410	73.8	75.1	98.8			A	NO	.0002	4,881		625-0	0.29						
	4 4 A	-		4	520	349	1923	350	74.0	74.7	70.3			W	NO	.0002	2,805									
	5 4 A	-		6	600	403	192b	410	73.8	75.1	86.3			A	NO	.0002	4,861									
	Plant				2,170	1,457							66.8					B				2	8	5	
5	1 2 M	S		6	720	484	1926	--	46.0	--	--			0	YES											
	2 2 M	S		6	720	484	1927	--	46.0	--	--			0	YES											
	3 2 M	S		6	720	484	1927	--	46.0	--	--			0	YES											
	Plant				2,160	1,452		2,200					--					-				-	-	-	
80	1 4 A	-		4	365	245	1921	151	73.8	45.5	--			A	NO	.0010	2,105									
	2 4 A	-		4	520	349	1921	266	74.0	56.4	--			W	NO	.0009	2,105		--1,191--	0.55						
	3 4 A	-		6	600	403	1929	306	73.8	56.1	--			A	NO	.0012	3,272									
	4 4 M	-		4	665	447	1926	308	70.4	46.5	--			A	YES	.0038	4,695									
	Plant				2,150	1,444		740					55.8					B-E				2	8	1	3[k]
6	1 2 M	CC		6	300[1]	202	192b	--	29.4	--	--			A												
	2 4 A	-		6	600	403	1926	--	69.8	--	--			A												
	3 4 A	-		6	600	403	1927	--	69.8	--	--			A												
	4 4 A	-		6	600	403	1928	--	68.8	--	--			A												
	Plant				2,100	1,411		1,600					--					-				-	-	-	
68	1 2 M	CC		4	240	161	1925	120	35.3	26.3	54.5			A	NO											
	2 2 M	CC		6	360	242	1925	220	35.3	32.1	58.5			A	NO				--1,044--	0.51						
	3 2 M	S		4	560	376	1928	340	53.6	54.2	70.5			0	YES											
	4 2 M	S		6	900	604	1930	600	57.5	57.1	70.7			0	YES											
	Plant				2,060	1,383		670					67.5					B-E				3	8	1	
728	1 4 A	-		6	600	403	1926	420	73.8	76.9	77.1			A	NO											
	2 4 A	-		6	600	403	1926	425	73.8	77.8	77.4			A	NO				200-140	700-252	0.63	0				
	3 2 M	S		6	840	564	1929	575	53.6	54.6	51.8			0	NO							0				
	Plant				2,040	1,370							67.1					D				3	8	2	
93	1 2 M	S		5	700	470	1928	--	53.6	--	51.5			0	YES											
	2 2 M	S		5	700	470	1928	--	53.6	--	51.7			0	YES											
	3 2 M	S		4	560	376	1928	--	53.6	--	55.9			0	YES											
	Plant				1,960	1,316		1,400[2]					52.3					D				3	8	1	
215	1 2 M	S		7	980	658	1930	700	53.6	57.0	89.0			0	YES											
	2 2 M	S		7	980	658	1930	800	53.6	65.2	91.2			0	YES				439-96	0 229-61	0.42					
	Plant				1,960	1,316		1,400[2]					89.9					B				3	8	1	
837	1 4 A	-		4	750	504	1922	--	74.5	--	50.2			W	NO											
	2 4 A	-		6	450	302	1918	--	75.0	--				A	NO											
	3 4 A	-		3	225	151	1904	--	75.0	--	56.3			A	NO				3725 -- 3325	3.76						
	4 4 A	-		3	225	151	1904	--	75.0	--				A	NO											
	5 4 A	-		3	225	151	1906	--	75.0	--				A	NO											
	Plant				1,875	1,259		720					51.1					A				3	8	2	
997	1 4 A	-		6	600	403	1929	350	73.8	64.1	--			A	YES											
	2 4 A	-		6	600	403	1929	--	73.8	--	--			A	YES											
	3 4 A	-		6	600	403	1929	--	73.8	--	--			A	YES											
	Plant				1,800	1,209		1,050					75.5					D-E				3	8	1½	
212	1 4 A	-		8	875	588	1929	510	75.1	65.1	73.3			A	NO											
	2 4 A	-		8	875	588	1929	580	75.1	74.1	66.7			A	NO				-- 20 --	0.01		0				
	Plant				1,750	1,176		720					69.6					A				-	-	1[h]	
722	1 2 M	CC		4	300[1]	134	1923	130	29.4	28.5	73.7			A	NO											
	2 2 M	CC		6	300[1]	202	1923	200	29.4	29.1	79.3			A	NO											
	3 2 M	S		6	840	564	1929	580	53.6	53.2	70.8			0	YES				--111--	0.06						
	4 4 A	-		4	365	245	1916	220	73.8	66.3	81.1			A	NO											
	Plant				1,705	1,145		1,090[2]					72.8					C				1	8	2	2[k]

TABLE V - INFORMATION COVERING MAINTENANCE AND OPERATING LABOR (Page 3)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM-SEE NOTES	SCAVENGING SYSTEM-SEE NOTES	NUMBER OF CYLINDERS	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	YEAR ENGINE STARTED TO WORK	PEAK LOAD DURING REPORTED PERIOD - K.W.	B.MEP. AT RATED B.H.P.- LBS. PER SQ. IN.	B.MEP. AT PEAK LOAD (90% GENERATING EFFICIENCY)	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	PISTON COOLING-SEE NOTES	ARE AIR FILTERS USED?	AVERAGE LINER - WEAR PER 1000 HRS. OPERATION - INS.	OPERATING PERIOD OVER WHICH LINER WEAR WAS MEASURED - HRS.	TYPE COOLING SYSTEM (SEE TEXT)	COST OF REGULAR ENGINE UPKEEP - TOTAL DOLLARS - MATERIAL - EXTRA LABOR	COST OF REPAIRS FOR ENGINE ACCIDENTS - TOTAL DOLLARS - MATERIAL - EXTRA LABOR	TOTAL ENGINE REPAIRS IN DOLLARS PER RATED B.H.P. PER YEAR	NUMBER OF UNFORCED SHUT-DOWNS	TOTAL DURATION OF ENFORCED SHUTDOWNS - HOURS	NUMBER OF SHIFTS PER DAY	NUMBER OF HOURS PER SHIFT	NUMBER OF ATTENDANTS PER SHIFT
735	1	2	M	S	6	840	564	1928	550	53.6	52.3	82.6		O	YES	.0013	12,416		--14 38--		{0.86}	-	-			
	2	2	M	S	6	840	564	1928	550	53.6	52.3	73.1		O	YES	.0010	15,692					-	-			
	Plant					1,680	1,128		1,100				77.8					D-E						3	10	2h
18	1	2	M	CC	5	450	302	1922	--	44.3				A					--	--	--	-	-			
	2	4	A	-	6	600	403	1926	--	69.8				A					--	--	--	-	-			
	3	4	A	-	6	600	403	1928	--	69.8	--	--		A					--	--	--	-	-			
	Plant					1,650	1,108		1,100				--					--					-	--	--
8	1	4	A	-	4	400	269	1921	270	77.2	77.5	44.9		A	NO	--	--		--	--	--	0	-			
	2	4	A	-	3	560	376	1922	290	73.4	56.6	43.7		O	NO	--	--		--	--	--	0	-			
	3	4	A	-	6	600	403	1926	450	69.8	78.0	70.0		A	YES	--	--		--	--	--	0	-			
	Plant					1,560	1,048		650				58.7					B-E						3	8	2
10	1	4	A	-	4	750	504	1924	440	79.5	69.4	86.9		W	NO	.0015	--		283-1,237		0	2.03	2	8	1	8	4
	2	2	A	S	4	750	504	1925	440	68.5	59.9	72.0		W	YES	.0097			397-603		0	1.33	3	8	2	8	2
	Plant					1,500	1,008		440				69.9					A						also 1 relief operator			oiler
132	1	4	A	-	3	375	252	1929	250	74.7	74.1	62.9		A	YES	--	--		0		0	0	0			
	2	4	A	-	4	500	336	1929	340	74.7	75.6	74.1		A	YES	--	--		0		0	0	0			
	3	4	A	-	5	625	420	1929	430	74.7	76.5	71.8		A	YES	--	--		0		0	0	0			
	Plant					1,500	1,008		565[a]				69.1					D						3	10	1
																								1	8	1
289	1	4	A	-	4	250	168	1920	--	74.1	--	70.0		A	YES	--	--		---		---	--	--			
	2	4	A	-	4	250	168	1920	--	74.1	--	69.3		A	YES	--	--		---		---	--	--			
	3	4	A	-	4	365	245	1915	--	73.8	--	67.8		A	YES	--	--		---		---	--	--			
	4	4	A	-	6	600	403	1927	--	73.8	--	77.6		A	YES	--	--		---		---	--	--			
	Plant					1,465	984		780				71.3					A						1	8	3
																								2	8	2
865	1	4	A	-	3	225	151	1908[1]	--	75.1	--	--		A	NO	--	--		---		---	--	--			
	2	4	A	-	3	225	151	1911[1]	--	75.0	--	--		A	NO	--	--		---		---	--	--			
	3	4	A	-	4	500	336	1921	--	76.0	--	--		A	NO	--	--		---		---	--	--			
	4	4	A	-	4	500	336	1917	--	76.0	--	--		A	NO	--	--		---		---	--	--			
	Plant					1,450	974		830				--					B-E						1	8	3
																								2	8	1
70	1	2	M	S	4	560	376	1928	380	53.6	54.2	65.1		O	YES	--	--				0		0			
	2	2	M	S	4	560	376	1928	400	53.6	57.0	59.6		O	YES	--	--		{--1,579-}		0	{1.11}	0			
	3	2	M	CC	6	300[1]	202	1920	190	29.4	27.7	61.3		A	NO	--	--				0		0			
	Plant					1,420	954		560				82.0					A						3	8	1
154	1	4	A	-	6	600	403	1927	--	72.7	--	--		A	-				---		---	--	-			
	2	4	A	-	4	400	269	1926	--	72.7	--	--		A	-				---		---	--	-			
	3	4	A	-	4	400	269	1925	--	72.7	--	--		A	-				---		---	--	-			
	Plant					1,400	941		640				--					--					-	--	--
163	1	4	M	-	4	400	269	1924	275	64.7	66.1	73.9		A	NO	--	--		---		---	--	-			
	2	4	M	-	4	400	269	1927	275	64.7	66.1	76.9		A	NO	--	--		---		---	--	-			
	3	4	M	-	6	600	403	1929	500	64.7	80.3	77.2		A	NO	--	--		---		---	--	-			
	Plant					1,400	941		1,050				76.3					A						3	8	1
11	1	4	A	-	4	750	504	1924	500	79.5	78.9	88.5		W	NO	.0057	--		222-1,322		0	2.06	0	--	1	8	3
	2	4	A	-	6	600	403	1926	400	77.1	76.5	96.7		A	NO	.0023	--		978-1,491		0	4.11	2	4½	1	8	2
	Plant					1,350	907		500				95.9					A						also 1 relief oiler			
725	1	4	A	-	6	600	403	1926	--	69.8	--	--		A	NO	--	--						0			
	2	4	A	-	4	400	269	1925	--	69.8	--	--		A	NO	--	--		{502-0}		{50-0}	{0.42}	0			
	3	2	M	CC	4	200[1]	134	1924	--	29.4	--	--		A	NO	--	--						0			
	4	2	M	CC	2	110[1]	74	1923	--	31.5	--	--		A	NO	--	--						0			
	Plant					1,310	880		630				--					D-E						3	8	1
13	1	4	A	-	4	275	185	1923	200	75.8	81.9	61.7		A	NO	.0011	6,509		225-0		0	0.82	0				
	2	4	A	-	3	200	134	1923	135	73.5	74.0	65.5		A	NO	.0012	3,402		100-0		0	0.52	0				
	3	4	A	-	4	750	504	1925	425	73.8	62.4	49.0		W	NO	.0032	1,574		70-0		0	0.09	0				
	Plant					1,225	823		425				58.3					A						3	8	1
17	1	4	A	-	6	600	403	1926	--	69.8	--	--		A	-	--	--		---		---	--	-			
	2	4	A	-	6	600	403	1926	--	69.8	--	--		A	-	--	--		---		---	--	-			
	Plant					1,200	806		950				--					--					-	--	--
94	1	4	A	-	6	600	403	1927	--	72.7	--	64.6		A	YES	---	--		---		---	--	-			
	2	4	A	-	6	600	403	1927	--	72.7	--	72.4		A	YES	---	--		---		---	--	-			
	Plant					1,200	806		950[a]				68.4					D						3	10	1
333	1	4	A	-	6	1,280	806	1929	800	70.1	69.7	83.3	83.3	O	NO	---	--	A	{ --15 00-- }			{1.25}	-	--	2	10	1
																								1	4	1
451	1	2	M	S	4	600	403	1931[m]	440	57.5	62.8	--		O	YES	---	--		{ --1 04-- }			{0.09}	{1 (20)}			
	2	2	M	S	4	600	403	1931[m]	440	57.5	62.8	--		O	YES	---	--									
	Plant					1,200	806		480				--					B-E						3	8	1
980	1	4	A	-	4	450	302	1927	--	71.2	--	--		A	NO	---	--		165-13		736-53	2.16	-	--			
	2	4	A	-	6	750	504	1929	--	74.7	--	--		A	NO	---	--		197-25		0	0.30	-	--	1	8	3
	Plant					1,200	806		640				--					A						2	8	2
616	1	4	A	-	3	300	202	1923	200	77.2	76.4	--		A	NO	--	--				0		-			
	2	2	M	CC	2	100[1]	67	1920	50	29.4	21.9	--		A	NO	--	--		{3,413-572}		0	{3.47}	-	{1}			
	3	4	A	-	4	750	504	1925	520	74.5	76.9	--		W	NO	--	--				0		-		1	8	2
	Plant					1,150	773		520				--					B						2	8	1
95	1	2	M	S	4	560	376	1929	--	53.6	--	51.4		O	YES	--	--		---		---	--	-			
	2	2	M	S	4	560	376	1929	--	53.6	--	52.4		O	YES	--	--		---		---	--	-			
	Plant					1,120	752		500[a]				51.9					B						3	10	1
96	1	4	M	-	6	600	403	1925	--	64.6	--	46.8		A	NO	--	--		---		---	--	-			
	2	4	M	-	3	300	202	1923	--	64.6	--	53.2		A	NO	--	--		---		---	--	-			
	3	2	M	CC	4	200[1]	134	1921	--	29.4	--	40.3		A	NO	--	--		---		---	--	-			
	Plant					1,100	739		600[a]				50.9					A						3	10	1
721	1	2	M	CC	6	360	242	1925	--	35.3	--	--		A	YES	--	--				0		-			
	2	2	M	CC	6	360	242	1925	--	35.3	--	--		A	YES	--	--		{-1460-}		0	{4.88}	-			
	3	2	M	CC	6	360	242	1926	--	35.3	--	--		A	YES	--	--				3,401[o]-409[q]		-			
	Plant					1,080	726		300				---					B-E						3	8	1

TABLE V - INFORMATION COVERING MAINTENANCE AND OPERATING LABOR (Page 4)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM-SEE NOTES	SCAVENGING SYSTEM-SEE NOTES	NUMBER OF CYLINDERS	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	YEAR ENGINE STARTED TO WORK	PEAK LOAD DURING REPORTED PERIOD - K.W.	B.MEP. AT RATED B.H.P. - LBS. PER SQ. IN.	B.MEP. AT PEAK LOAD (90% GENERATING EFFICIENCY)	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	PISTON COOLING - SEE NOTES	ARE AIR FILTERS USED?	AVERAGE LINER - WEAR PER 1000 HRS. OPERATION - INS.	OPERATING PERIOD OVER WHICH LINER WEAR WAS MEASURED - HRS.	TYPE COOLING SYSTEM (SEE TEXT)	COST OF REGULAR ENGINE UPKEEP - TOTAL DOLLARS - MATERIAL - EXTRA LABOR	COST OF REPAIRS FOR ENGINE ACCIDENTS - TOTAL DOLLARS - MATERIAL - EXTRA LABOR	TOTAL ENGINE REPAIRS IN DOLLARS PER RATED B.H.P. PER YEAR	NUMBER OF ENFORCED SHUT-DOWNS	TOTAL DURATION OF ENFORCED SHUTDOWNS - HOURS	NUMBER OF SHIFTS PER DAY	NUMBER OF HOURS PER SHIFT	NUMBER OF ATTENDANTS PER SHIFTS
78	1 | 4 | M	--	4	400	269	1928	280	64.6	67.2	34.0		A	YES	.0057	30,270		5-73	0	0.20	0	--	2	12	1
	2 | 4 | M	--	6	660	443	1930	475	66.7	-	54.3		A	YES	--	--		149-57	0	0.31	0	--	2	12	1
	Plant					1,060	712		730[p]		71.5		40.5										2	8	1
83	1 | 4 | A	--	6	600	403	1929	420	73.8	76.9	59.8		A	YES	.0004	4,790									
	2 | 4 | A	--	4	400	269	1927	300	73.8	82.3	40.6		A	YES	.0006	5,042		{200-1598}	0	{1.80}	0	--			
	Plant					1,000	672		610				54.4									B	3	8	2
98	1 | 2 | M	CC	6	360	242	1926	--	35.3	--	65.4		A	YES				---	---	--	-	--			
	2 | 2 | M	CC	6	360	242	1929	--	35.3	--	65.0		A	YES				---	---	--	-	--			
	3 | 2 | M	CC	3	180	121	1926	--	35.3	--	67.3		A	YES				---	---	--	-	--			
	4 | 2 | M	CC	2	100[1]	67	--	--	29.4	--	0		A	NO				---	---	--	-	--			
	Plant					1,000	672		475				65.6									D	3	10	1
738	1 | 4 | A	--	6	1,000	672	1918[s]	600	72.4	64.8	76.3	76.3	0	NO	--	--	-	{ -1,397- }		{1.40}	-	--	3	10	1
801	1 | 2 | A	S	4	1,000	672	1927	700	62.5	65.2	84.9	84.9	W	YES	.0060	8,390	A	{ -1,289- }		{1.29}	5	--	3	8	2[h]
1061	1 | 2 | M	S	7	980	658	1930	700	53.6	57.0	81.4	81.4	0	NO	--	--	D	-1,239	0	1.27	-	--	2	8	1
863	1 | 4 | A	-	3	300	201	1925	150	72.7	54.3	78.1		A	YES	--	--		0	0	0	0	--			
	2 | 4 | A	-	6	675	453	1927	400	72.7	64.2	84.5		A	YES	--	--		1,400-120	0	2.25	0	--			
	Plant					975	655		550				82.7			D-E							3	8	1
49	1 | 4 | A	-	4	485	326	1924	280	78.6	67.5	50.9		A	NO	.0060	4,156									
	2 | 4 | A	-	4	485	326	1924	290	78.6	69.9	51.9		A	NO	.0052	4,810		{ -1,850- }		{1.91}	-	--			
	Plant					970	652		390				51.8			A							2	12	2
53	1 | 2 | M	CC	4	200[1]	134	1923	130[a]	29.4	28.5	69.6		A	NO	.0065	2,911		{-663-b		{3.32	-	--			
	2 | 2 | M	CC	6	300[1]	202	1923	190[a]	29.4	27.7	72.0		A	NO	.0058	4,838		-1,313-b		4.38}	-	--			
	3 | 4 | A	-	4	400	269	1925	260[a]	73.8	71.3	70.7		A	NO	.0018	7,045		-4,956-b-n		12.39n}	-	--			
	Plant					900	605		530[a]				71.0			B-E							3	10	1
260	1 | 2 | A	S	6	900	605	1928	615	64.4	65.6	50.6	50.6	0	NO	--	--	A	0	0	0	-	--	1[r]	1-18	1-2
74	1 | 2 | M	CC	3	225	151	1924	150	36.7	36.5	58.0		0	NO	--	--		0	0	0	0	--			
	2 | 4 | M	-	4	280	175	1928	200	62.5	71.4	71.5		A	YES	--	--			0		0	--			
	3 | 4 | M	-	6	400	269	1928	325	64.1	77.5	58.4		A	YES	--	--		{60-21}	0	{0.12}	0	--			
	Plant					885	595		475				65.7			B							3	8	1
189	1 | 4 | M	-	5	550	369	1929	300	66.7	54.2	31.3		A	YES	--	--		---	---	--	0	--			
	2 | 4 | M	-	2	200[1]	134	1922	125	72.6	67.8	66.1		A	NO	--	--		---	---	--	0	--			
	3 | 4 | M	-	1	100[1]	67	1915	0	72.6	--	0		A	NO	--	--		---	---	--	0	--			
	Plant					850	570		300				31.5			D-E							3	8	1
153	1 | 2 | M	S	6	840	564	1929	560	53.6	53.2	--	--	0	-	--	--	-	---	---	--	-	--	-	--	-
494	1 | 2 | M	S	6	840	564	1930	500	53.6	47.5	70.8	70.8	0	YES	--	--	B-E	405-150	0	0.66	3	72	1	10	1
619	1 | 2 | M	S	6	840	564	1929	550	53.6	52.3	74.1	74.1	0	YES	--	--	B-E	100-100	1,008-200	1.68	2	288	1	10	1
1056	1 | 2 | M	S	6	840	564	1929	570	53.6	54.2	76.2	76.2	0	YES[5]	--	--	D	-1,266-	0	1.51	30	233	2	8	1-2
731	1 | 4 | A	-	4	825	554	1926	600	74.9	81.1	61.2	61.2	W	YES	--	--	D	350-0	0	0.42	0	--	3	9	1
648	1 | 4 | M	-	4	400	269	1928	225	64.6	54.0	80.5		A	YES	--	--		11-10	0	0.05	0	--			
	2 | 4 | M	-	4	400	269	1928	225	64.6	54.0	81.2		A	YES	--	--		0	0	0	0	--			
	Plant					800	538		430				80.9			B-E							2	10	1
100	1 | 2 | M	CC	6	300[1]	202	1923	-	29.4	--	38.6		A	YES	--	--		---	---	--	-	--			
	2 | 2 | M	CC	4	240	161	1924	-	35.3	--	62.6		A	YES	--	--		---	---	--	-	--			
	3 | 2 | M	CC	4	200[1]	134	1924	-	29.4	--	72.0		A	YES	--	--		---	---	--	-	--			
	Plant					740	497		320[t]				66.0			B							3	10	1
112	1 | 2 | M	CC	4	200[1]	134	1925	130	29.4	28.5	74.8		A	NO	--	--									
	2 | 2 | M	CC	4	240	161	1927	150	35.3	32.9	69.7		A	NO	--	--		{ -420- }		{0.57}	-	--			
	3 | 2 | M	CC	6	300[1]	202	1925	190	29.4	27.8	73.7		A	NO	--	--					-	--	1	4	1
	Plant					740	497		425				71.3			D							2	10	1
67	1 | 2 | M	CC	6	360	242	1925	210	35.3	30.6	82.2		A	NO	--	--			0						
	2 | 2 | M	CC	6	360	242	1925	210	35.3	30.6	82.3		A	NO	--	--		{539-396}		{1.30}	5	--	1	8	2[h]
	Plant					720	484		210				82.3			A				7	--	1	8	1[h]
170	1 | 2 | M	CC	6	360	242	1929	-	35.3	--	--		A	NO	.0029	4,650			67-46		3	7			
	2 | 2 | M	CC	6	360	242	1929	-	35.3	--	--		A	NO	.0021	4,740		{171-1,901[t]	483-235	{3.06}	4	23			
	Plant					720	484		440				64.8			B-E							1	10	2[h]
527	1 | 2 | M	CC	6	360	242	1930	230	35.3	33.6	37.2		A	YES	--	--			0		2	1/5			
	2 | 2 | M	CC	6	360	242	1930	230	35.3	33.6	45.2		A	YES	--	--		{117-0}	0	{0.16}	0	--			
	Plant					720	484		380				41.4			D							3	8	1
718	1 | 2 | M	CC	6	360	242	1926	225	35.3	32.8	40.4		A	NO	--	--			0						
	2 | 2 | M	CC	6	360	242	1926	210	35.3	30.6	38.9		A	NO	--	--		{-4,341}	0	{6.03}	2	1/15			
	Plant					720	484		225				34.9			B							2	12	1[h]
19	1 | 2 | M	CC	2	100[1]	67	--	--	29.4	--	--		A	-	--	--		---	---	--	-	--			
	2 | 2 | M	CC	4	240	161	1927	--	45.0	--	--		A	-	--	--		---	---	--	-	--			
	3 | 2 | M	CC	4	360	242	1927	--	44.3	--	--		-	-	--	--		---	---	--	-	--			
	Plant					700	470		245		--					-							-	--	-
368	1 | 4 | A	--	4	400	269	1925	270	73.8	74.1	81.8		A	NO	.0018	6,250		513-0	975-0	3.72	2	12			
	2 | 4 | A	--	3	300	202	1929	200	73.8	73.1	96.0		A	NO	--	--		210-0	325-0	1.78	1	6	1	11	1
	Plant					700	471		470				87.2			B						1	12	1
61	1 | 4 | M	--	3	300	202	1928	230	64.7	74.1	69.5		A	NO	--	--		20-84	0	0.35	0	--			
	2 | 4 | M	--	3	300	202	1928	230	64.7	74.1	66.9		A	NO	--	--		498-110	647-309	5.22	3	467			
	Plant					600	404		460				68.4			B							3	9	1[u]
106	1 | 4 | A	--	6	600	403	1929	630	69.8	109.0	--	--	A	-	--	--	-	---	---	--	-	--	-	--	-

TABLE V - INFORMATION COVERING MAINTENANCE AND OPERATING LABOR (Page 5)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM-SEE NOTES	SCAVENGING SYSTEM-SEE NOTES	NUMBER OF CYLINDERS	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	YEAR ENGINE STARTED TO WORK	PEAK LOAD DURING REPORTED PERIOD - K.W.	B.MEP. AT RATED B.H.P. - LBS. PER SQ. IN.	B.MEP. AT PEAK LOAD (90% GENERATING EFFICIENCY)	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	PISTON COOLING-SEE NOTES	ARE AIR FILTERS USED?	AVERAGE LINER - WEAR PER 1000 HRS OPERATION - INS.	OPERATING PERIOD OVER WHICH LINER WEAR WAS MEASURED - HRS.	TYPE COOLING SYSTEM (SEE TEXT)	COST OF REGULAR ENGINE UP-KEEP - TOTAL DOLLARS - MATERIAL - EXTRA LABOR	COST OF REPAIRS FOR ENGINE ACCIDENTS - TOTAL DOLLARS - MATERIAL - EXTRA LABOR	TOTAL ENGINE REPAIRS IN DOLLARS PER RATED B.H.P. PER YEAR	NUMBER OF ENFORCED SHUT-DOWNS	TOTAL DURATION OF ENFORCED SHUTDOWNS - HOURS	NUMBER OF SHIFTS PER DAY	NUMBER OF HOURS PER SHIFT	NUMBER OF ATTENDANTS PER SHIFT
1094	1 4	A	-	6	600	403	1928	410	73.8	75.1	78.5	78.5	A	No	--	---	D	816-408	0	2.04	1	72	2	10	1[h]
1096	1 4	A	-	6	600	403	1928	420	73.8	76.9	74.0	74.0	A	No	--	---	D	250-100	0	0.58	2	48	2	10	1
103	1 2	M	CC	4	240	161	1927	-	35.3	--	57.1		A	No	--	---		---	---	--	-	--			
	2 2	M	CC	4	200[1]	134	1920[V]	-	29.4	--	65.1		A	No	--	---		---	---	--	-	--			
	3 2	M	CC	3	150[1]	101	1923	-	29.4	--	52.8		A	No	--	---		---	---	--	-	--			
	Plant					396		350				58.3					D						3	10	1
862	1 2	M	CC	2	75[1]	50	1920	-	29.1	--	--		A	No	--	---									
	2 2	M	CC	2	75[1]	50	--	-	29.1	--	--		A	No	--	---									
	3 2	M	CC	3	180	121	1925	-	35.3	--	--		A	No	--	---									
	4 4	A	-	3	258	173	1918[W]	-	72.2	--	--		A	No	--	---		-795-		1.35	-	--			
	Plant				588	394		300				46.9					B						3	8	1
102	1 2	M	CC	4	240	161	1928	-	35.3	--	46.9		A	Yes	--	---		---	---	--	-	--			
	2 2	M	CC	4	240	161	1928	-	35.3	--	49.9		A	Yes	--	---		---	---	--	-	--			
	3 2	M	CC	2	100[1]	67	1922[V]	-	29.4	--	65.0		A	Yes	--	---		---	---	--	-	--			
	Plant				580	389		290[A]				50.1					D						3	10	1
101	1 2	M	CC	4	240	161	1926	-	35.3	--	53.8		A	Yes				---	---	--	-	--			
	2 2	M	CC	3	150[1]	101	1919	-	29.4	--	53.4		A	No	--	---		---	---	--	-	--			
	3 2	M	CC	3	150[1]	101	1923	-	29.4	--	60.3		A	No	--	---		---	---	--	-	--			
	Plant				540	363		180[A]				55.7					B						3	10	1
246	1 2	M	CC	2	100[1]	67	1918	45	29.4	19.8	--		A	No	--	---		0	0	0	0	--			
	2 2	M	CC	4	200[1]	134	1922	100	29.4	21.9	--		A	No	--	---		142-0	0	0.71	0	--			
	3 2	M	CC	4	240	161	1929	120	35.3	26.3	--		A	No	--	---		0	0	0	0	--	2	10	1
	Plant				540	362		145				38.0					B						1	4	1
540	1 2	M	CC	6	360	242	1929	240	35.3	35.0	--		A	Yes	--	---									
	2 2	M	CC	3	180	121	1926	125	35.3	36.5	--		A	Yes	--	---		-270-		0.50	0	--			
	Plant				540	363		260				--					B						3	8	1
772	1 4	A	-	4	250	168	1916	140	74.1	61.7	74.5		A	Yes	--	---		135-270	0	1.62	0	--			
	2 4	A	-	4	250	168	1916	140	74.1	61.7	74.5		A	Yes	--	---		140-270	0	1.64	0	--			
	Plant				500	336		280				74.5					B-E						2	10	1[h]
501	1 2	M	CC	2	120	81	1925	55	35.3	24.0	32.3		A	Yes	--	---			0		0	--			
	2 2	M	CC	2	120	81	1925	50	35.3	21.8	33.3		A	Yes	--	---		138-145	0	0.59	1	1/12	1	10	1
	3 2	M	CC	4	240	161	1929	145	35.3	31.8	51.3		A	Yes	--	---			0		0	--	1	9	1
	Plant				480	323		145				39.3					B-E						1	5	2
615	1 2	M	CC	2	100[1]	67	1922	65	29.4	28.5	--		A	No	--	---		100-0	0	1.00	0	--			
	2 2	M	CC	2	100[1]	67	1922	60	29.4	26.3	--		A	No	--	---		0	0	0	0	--			
	3 2	M	CC	4	240	161	1930	125	35.3	27.4	--		A	No	--	---		0	0	0	0	--	1	12	1
	Plant				440	295		125				--					A						1	12	2
931	1 2	A	S	4	440	295	1922	-	54.8	--	74.6	74.6	W	No	.0017	39,675	B	43-0	58-0	0.23	0	--	3	8	1
335	1 4	A	-	4	250	168	1924	147	74.1	64.8	34.2		A	No	--	---			0		0	--			
	2 4	A	-	4	180	121	1919	120	73.0	72.4	47.9		A	No	--	---		195-0	0	0.45	0		1	4	1
	Plant				430	289		147				37.3					A						2	10	1
24	1 4	M	-	3	200	134	1925	120	64.1	57.4	47.3		A	No	--	---		22-10	0	0.16	0	--			
	2 4	M	-	3	200	134	1925	120	64.1	57.4	47.0		A	No	--	---		103-12	0	0.58	0	--			
	Plant				400	268		150				47.1					B						3	8	1
194	1 4	M	-	4	400	269	1928	188	72.7	50.8	34.7	34.7	A	Yes	--	---	D	0	0	0	0	--	1	11-12	2[h]
247	1 2	M	CC	1	50[1]	34	1915	-	29.4	--	--		A	No	--	---			0		0	--			
	2 2	M	CC	2	100[1]	67	1920	57	29.4	25.0	--		A	No	.0002	3,650		193-0	0	0.50	0	--			
	3 2	M	CC	4	240	161	1928	168	35.3	36.9	--		A	No	.0002	12,300			0		0	--			
	Plant				390	262		219				44.6					A						3	8	1
105	1 2	M	CC	3	150[1]	101	1922	-	29.4	--	55.1		A	Yes	--	---		---	---	--	-	--			
	2 2	M	CC	2	120	81	1926	-	35.3	--	34.7		A	Yes	--	---		---	---	--	-	--			
	3 2	M	CC	2	100[1]	67	1923	-	29.4	--	42.2		A	Yes	--	---		---	---	--	-	--			
	Plant				370	249		140				47.4					B						3	10	1
152	1 2	M	CC	6	360	242	1925	150	35.3	21.9	--	--	A	-	--	---	-	---	---	--	-	--	-	--	-
161	1 2	M	CC	6	360	242	1929	255	35.3	37.2	95.4	95.4	A	Yes	--	---	B	97-0	0	0.27	-	--	2	12	1
1091	1 2	M	CC	3	180	121	1929	120	35.3	35.0	36.1		A	No	--	---		---	---	--	-	--			
	2 2	M	CC	3	180	121	1930	120	35.3	35.0	36.2		A	No	--	---		---	---	--	-	--			
	Plant				360	242		150				36.2					B						3	8	1
688	1 2	M	CC	1	50[1]	34	1923	25	29.4	21.6	--		A	No	--	---			0		0	--			
	2 2	M	CC	2	120	81	1926	80	35.3	34.9	--		A	No	--	---		50-25	0	0.21	0	--			
	3 2	M	CC	3	180	121	1927	125	35.3	36.5	--		A	No	--	---			0		0	--			
	Plant				350	236		150				--					D						3	8	1
365	1 4	A	-	4	165	111	1917	120	72.6	78.5	--		A	No	--	---		---	---	--	0	--			
	2 4	A	-	4	180	121	1922	120	73.1	72.5	--		A	No	--	---		---	---	--	0	--			
	Plant				345	232		175				--					B						3	8	1
774	1 4	A	-	4	165	111	1917	90	72.6	58.8	68.3		A	No	--	---					0	--			
	2 4	A	-	4	180	121	1920	90	73.1	54.3	62.4		A	No	--	---		535-134		1.94	1	336			
	Plant				345	232		180				66.2					B-E						1	10	1[h]
265	1 2	M	CC	1	60	40	1927	30	35.3	26.5	--		A	No	--	---			0		0	--			
	2 2	M	CC	2	120	81	1928	65	35.3	28.3	--		A	No	--	---		75-25	0	0.31	0	--			
	3 2	M	CC	2	140	94	1931[S]	96	35.3	36.1	--		A	No	--	---			0		0	--			
	Plant				320	215		110				--					C-E						2	12	1
27	1 2	M	CC	2	75[1]	50	1926	-	29.1	--	--		A	-	--	---		---	---	--	-	--			
	2 2	M	CC	4	240	161	1926	-	45.0	--	--		A	-	--	---		---	---	--	-	--			
	Plant				315	211		85				--					-						-	--	-
382	1 4	M	-	4	150	101	1927	-	99.7	--	20.8		A	No	-	---		0	0	0	0	--			
	2 4	M	-	4	150	101	1927	-	99.7	--	20.4		A	No	--	---		0	0	0	0	--	1	10	2[h]
	Plant				300	202		-				20.5					A						1	13	2[h]

TABLE V - INFORMATION COVERING MAINTENANCE AND OPERATING LABOR (Page 6)

PLANT NUMBER	ENGINE DESIGNATION	ENGINE CYCLE	INJECTION SYSTEM-SEE NOTES	SCAVENGING SYSTEM-SEE NOTES	NUMBER OF CYLINDERS	RATED B.H.P.	EQUIVALENT K.W. - 90% GENERATING EFFICIENCY	YEAR ENGINE STARTED TO WORK	PEAK LOAD DURING REPORTED PERIOD - K. W. / B.MEP. AT RATED B.H.P.- LBS. PER SQ. IN.	B.MEP. AT PEAK LOAD (90% GENERATOR EFFICIENCY)	RUNNING ENGINE CAPACITY FACTOR (SEE TEXT)	RUNNING PLANT CAPACITY FACTOR (SEE TEXT)	PISTON COOLING-SEE NOTES	ARE AIR FILTERS USED?	AVERAGE LINER - WEAR PER 1000 HRS. OPERATION - INS.	OPERATING PERIOD OVER WHICH LINER WEAR WAS MEASURED - HOURS.	TYPE COOLING SYSTEM (SEE TEXT)	COST OF REGULAR ENGINE UPKEEP - TOTAL DOLLARS - MATERIAL - EXTRA LABOR	COST OF REPAIRS FOR ENGINE ACCIDENTS - TOTAL DOLLARS - MATERIAL - EXTRA LABOR	TOTAL ENGINE REPAIRS IN DOLLARS PER RATED B.H.P. PER YEAR	NUMBER OF ENFORCED SHUT-DOWNS	TOTAL DURATION OF ENFORCED SHUTDOWNS - HRS.	NUMBER OF SHIFTS PER DAY	NUMBER OF HOURS PER SHIFT	NUMBER OF ATTENDANTS PER SHIFT
1129	1	2	M	CC	2	f75[i]	50	--	-- 47.6	--	31.5		A	NO	---	--		962-120	0	3.64	-	--			
	1	2	M	CC	2	f75[i]	50	--	-- 47.6	--	27.6		A	NO	---	--			0		-	--			
	3	2	M	CC	3	150[i]	101	1922	-- 29.4	--	29.2		A	NO	-- -	--			0		-	--	2	10	1
	Plant					300	201		80			29.7					B						1	4	1
156	1	2	M	CC	3	150[i]	101	1929[x]	-- 29.4	--	--		A	-	---	--			0	1.23	-	--			
	2	2	M	CC	2	120	81	1929	-- 35.3	--	--		A	-	---	--		--333--	0		-	--			
	Plant					270	182		78			--					B						2	12	1
1097	1	2	M	CC	3	150[i]	101	1924	110 29.4	32.0	--		A	YES	---	--		50 - 28	0	0.52	0	--			
	2	2	M	CC	2	120	81	1928	80 35.3	34.9	--		A	YES	---	--		20 - 23	0	0.36	0	--			
	Plant					270	182		110			--					A						1-2	10	1[h]
169	1	4	M	--	6	240	161	1928	160[y] 77.6	77.2	42.5	42.5	A	YES	.0004	3,055	B	109-289	144-267	3.37	2	32¼	1	8½	1[h]
515	1	2	M	CC	2	120	81	1926	-- 35.3	--	--		A	NO	---	--		0	0	0	0	--			
	2	2	M	CC	2	120	81	1930	-- 35.3	--	--		A	NO	---	--		0	0	0	0	--			
	Plant					240	162		100			--					B-E						1	10	1[z]
222	1	2	M	CC	1	60	40	1930	-- 35.3	--	--		A	NO	---	--					-	--			
	2	2	M	CC	2	120	81	1930	-- 35.3	--	--		A	NO	---	--		--201--		1.12	-	--			
	Plant					180	121		15			--					B						2	12	1[h]
733	1	2	M	CC	2	100	67	1922	63 37.2	35.0	67.7		A	NO	---	--		0	0	0	1	3/4			
	2	2	M	CC	2	75[i]	50	1914	54 29.1	31.4	57.7		A	NO	---	--		0	0	0	-		no regular		1[h]
	Plant					175	117		115			62.5					A								
646	1	2	M	CC	2	120	81	1927	50 35.3	21.8	16.1	16.1	A	NO	---	--	B-E	0	0	0	-	4	2	12	1[h]
201	1	2	M	CC	1	50[i]	34	1924	-- 50.8	--	--		A	NO	---	--					-	--			
	2	2	M	CC	1	50[i]	34	1927	-- 29.4	--	--		A	NO	---	--		--906--		9.06	-	--			
	Plant					100	68		22			--					B						2	12	1

NOTES:

INJECTION SYSTEM

A - Air

M - Mechanical

SCAVENGING SYSTEM (For 2-Stroke Cycle only)

S - Separate (i.e., Pump or Blower)

CC - Crank Case

PISTON COOLING

A - Air

O - Oil

W - Water

LETTERED NOTES

a - 60 Minute peak

b - Total maintenance labor; none included in attendance

c - Further expense on account of Sabotage in 1928

d - 24 average peak

e - 7 year overhaul for #1

f - Horizontal engine

g - November

h - attendant gives only part time to engine operation

i - Semi-diesel engine

j - Part of attendance is included in repair cost

k - Extra man part time only

l - Date of first installation; date installed in this plant unknown

m - May

n - Includes some new liners and pistons

o - Crank shaft failure

p - 35 Minute peak

r - Occasional

s - Installed in this plant in 1924

t - On overtime

u - Attendants have other duties - full charge is made against diesel plant, however

v - Installed in this plant in 1924

w - Installed in this plant in 1928

x - Date installed in this plant; date of original installation unknown

y - Estimated by operator

z - Runs part time unattended.

TABLE VI - INFORMATION ON INVESTED COSTS (Page 1)

PLANT NUMBER	NUMBER OF ENGINES	TOTAL INSTALLED K.W.	YEAR IN WHICH INVESTMENT WAS MADE	COST OF LAND		COST OF BUILDINGS		COST OF EQUIPMENT		OTHER COSTS		TOTAL COST OF PLANT		PLANT NUMBER
				COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	
43	8	7,746	Total	8,882	1	185,188	24	1,329,038	172	200,590	26	1,723,698	223	43
82	6	3,801		15,210	4	---	--	----	--	---	--	941,172	248	82
73	4	3,360	1923	---		---		95,000						73
			1924	---		---		91,000						
			1927	---		---		159,000						
			Total	---	-	---	--	345,000	103	---	--	----	--	
52	4	3,077	1915	10,584[a]		---		----						52
			1928	---		166,167		334,487						
			1929	---		1,899		15,265						
			1930	---		3,032		110,431						
			1931	12,633		2,747		486						
			Total	23,217	8	173,845	57	460,669	150	1,128	--	658,859	215	
45	5	2,854	1922	25,000		46,396		176,895						45
			1926					215,714						
			Total	25,000	9	46,396	16	392,609	137	45,623	16	509,628	178	
130	4	2,418	1923	500[a]		8,486[a]		---						130
			1924	10		8,672		94,948						
			1925	1,993		3,657		---						
			1926	284		3,407		62,336						
			1927	--		138		1,532						
			1928	--		11		1,383						
			1929	--		2,576		63,674						
			1930	--		104		56,525						
			Total	2,787	1	27,051	11	240,398	99	---	--	270,236	111	
109	3	1,963	1927	257		11,318		50,455						109
			1929	--		6,387		71,272						
			1931	--		9,358		61,671						
			Total	257	-	27,063	14	183,398	94	---	--	210,718	108	
7	5	1,854	Total	3,000	2	10,434	6	155,665	84	3,925	2	173,024	94	7
41	3	1,591	1912	5,785		10,309		54,357						41
			1922	---		20,000		---						
			1923	---		---		60,000						
			1928	---		20,000		100,000						
			Total	5,785	4	50,309	32	214,357	135	3,023	2	273,474	173	
215	2	1,316	1930	1,250		10,800		126,910						215
			1931	---		----		3,094						
			Total	1,250	1	10,800	8	130,004	99	---		142,054	108	
132	3	1,008	1929	----	-	39,262	39	102,046	101	21,500	21	----	--	132
13	3	823	1900	2,000		---		----						13
			1923	---		6,000		56,500						
			1925	---		8,000		65,000						
			Total	2,000	2	14,000	17	121,500	148	----	--	137,500	167	

TABLE VI - INFORMATION ON INVESTED COSTS (Page 2)

PLANT NUMBER	NUMBER OF ENGINES	TOTAL INSTALLED K.W.	YEAR IN WHICH INVESTMENT WAS MADE	COST OF LAND		COST OF BUILDINGS		COST OF EQUIPMENT		OTHER COSTS		TOTAL COST OF PLANT		PLANT NUMBER
				COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	
451	2	806	1931	---	-	12,750	16	71,138	88	13,482	17	---	--	451
616	3	773	Total	5,000	6	56,000	73	128,900	167	Included in equipment	--	189,900	246	616
801	1	672	1927	---	-	----	--	112,620	168	----	--	---	--	801
1061	1	658	1930	1,046		20,308		85,059						1061
			1931	---		----		638						
			Total	1,046	2	20,308	31	85,697	130	144	--	107,195	163	
49	2	652	1924	2,000[b]	3	3,500[b]	5	121,670	187	---	--	---	--	49
260	1	605	Total	---	-	9,087	15	60,064	99	853	1	---	--	260
74	3	595		1,000		---		---						74
			1924	---		5,000		34,000						
			1928	---		11,000		60,000						
			1929	1,430		----		---						
			Total	2,430	4	16,000	27	94,000	158	---	-	112,430	189	
494	1	564	1929	5,120		---		---						494
			1930	---		18,172		66,973						
			Total	5,120	9	18,172	32	66,973	119	4,226	7	94,491	167	
619	1	564	1929	---	-	19,404	35	73,191	130	4,782	8	---	--	619
1056	1	564	1929	1,623		32,597		107,427						1056
			1931	---		---		1,384						
			Total	1,623	3	32,597	58	108,811	193	5,129	9	148,160	263	
67	2	484	1887	100		---		----						67
			1925	---		9,000		49,815						
			Total	100	-	9,000	19	49,815	103	---	-	58,915	122	
527	2	484	1929	3,500		5,920		{82,015}						527
			1930	---		---								
			Total	3,500	7	5,920	12	82,015	169	---	-	91,435	188	
501	3	323	1924	1,000		---		----						501
			1925	---		6,000		24,000						
			1929	---		2,000		24,500						
			Total	1,000	3	8,000	25	48,500	150	2,000	6	59,500	184	
335	2	289	--	Gift		5,750		---						335
			1915	---		1,200		---						
			1919	---		---		28,735						
			1924	---		---		31,525						
			Total	0	-	6,950	24	60,260	208	3,061	11	70,271	243	
194	1	269	1928	---		---		41,000	152	---		---		194
247	3	262	Total	10	-	7,000	27	36,000	137	500	2	43,510	166	247

All fuel-oil and lubricating-oil costs include costs for the handling of oil from the cars to the tanks.

It is noteworthy that fuel-oil costs reported for plants Nos. 73 and 54 are considerably below anything previously reported—namely, 0.95 cent and 1.03 cents per gallon, respectively.

Cooling Water. The various water-cooling systems are indicated by the following symbols:

System A, raw water going to waste after one pass
System B, raw water recirculated after passing over cooling tower or spray pond
System C, soft water continuously recirculated, cooled by raw water going to waste after one pass through heat exchanger
System D, soft water continuously recirculated, cooled by raw water also recirculated after cooling by cooling tower or spray pond
System E, any of the foregoing systems with engine circulating water treated (added as a suffix).

TABLE VI - INFORMATION ON INVESTED COSTS (Page 3)

PLANT NUMBER	NUMBER OF ENGINES	TOTAL INSTALLED K.W.	YEAR IN WHICH INVESTMENT WAS MADE	COST OF LAND		COST OF BUILDINGS		COST OF EQUIPMENT		OTHER COSTS		TOTAL COST OF PLANT		PLANT NUMBER
				COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	COST IN DOLLARS	COST IN DOLLARS PER EQUIV. K.W.	
161	1	242	1929	---	-	---	--	34,000	140	---	--	---	---	161
365	2	232	1917	1,000		---		17,000						365
			1922	---				22,094						
			1929	---				6,887						
			Total	1,000	4			45,981	198	---	--	---	---	
265	3	215	1927	500		3,416		27,338						265
			1931	---		---		14,900						
			Total	500	2	3,416	16	42,238	196	---	--	46,154	214	
382	2	202	1927	---	-	2,402	12	28,466	141	---	--	---	---	382
222	2	121	1930	175	1	2,930	24	25,861	214	---	--	28,966	239	222

NOTES:

a - Appraisal Value as of year Indicated

b - Estimated by Operator

Investment Costs. In this report the investment cost data collected for 1930, together with what new information has been obtained for 1931, are set forth in Table 6. Owing to the difficulty in obtaining reliable information on expenditures over past years and to the fact that such data may not reflect current costs, the committee has decided to omit this feature of the investigation in the future. The questionnaire for 1932 will therefore not contain a question on investment costs.

Enforced Shutdowns. No change is made in this report over previous issues in respect to enforced shutdowns. For the next investigation, however, the committee purposes to ask also for the number of hours each unit was unavailable for service on account of regular maintenance. In the past no definition of "enforced shutdowns" has been laid down; the committee now proposes to define an enforced shutdown as a stoppage caused by actual or imminent engine trouble. A prearranged shutdown for maintenance work is not considered an enforced shutdown. The duration of an enforced shutdown is the time required to correct the trouble that caused the shutdown.

Peak Loads. The peak loads presented in the report are the highest average loads sustained for 15 min, unless otherwise stated in Table 5.

Liner Wear. Liner wear figures presented in Table 5 are in each case the average of the wear in inches in all the cylinders of the engine, the wear in each cylinder being found by averaging two measurements—one taken parallel to the crankshaft and the other at 90 deg to the crankshaft, both taken at the point of greatest wear, which in most cases will be at the top of the piston ring travel.

The committee realizes that liner wear is a complicated subject worthy of a separate investigation, and therefore feels that it should not go into this question more intensively, because it wishes to maintain its concentration on its main purpose. Therefore, it will omit investigation of liner wear from the next year's report rather than slight the subject.

Supplies and Miscellaneous. Supplies in the meaning of this report are items used in the power-generating plant that are consumed in the operating process—namely, such items as waste, packing, wipers, gage glasses, gaskets, bolts, screws, nails, dynamo and motor brushes, cans for containing rags and waste, transformer oil, and hand oil cans. The term "miscellaneous" as used in this report refers to such items as expenditures for lighting, heating, and cleaning systems, fire-protection systems, janitor's supplies, ice water, meals and carfares, stationery, telephone and toilet service, care of streets, yards, and sidings.

Type of Load. The terms used for type of load are defined as follows:

Complete power: The plant is run regularly alone when needed, without assistance from any base-load or peak-load service
Base load: The plant is run at substantially full load whenever its capacity can be used, usually supplemented by a peak-load service. When full or nearly full capacity cannot be used, the plant is shut down
Peak load: The plant is run only when the load exceeds the capacity of the regular source of power
Standby: The plant is run only when the regular source of power is interrupted.

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Progress in Oil and Gas Power

DURING the past year the spotlight in the oil-engine situation shifted to the fuel. The modern high-speed engines were found to be rather particular about their fuel and showed their dislike of certain fuels by knocking. This fact was made the subject of some searching investigations by several British and American oil companies. It was discovered that the suitability of a Diesel oil is largely determined by a previously unknown fuel characteristic, the ignitibility. The oil which ignites readily after a short ignition lag is the desirable fuel for Diesel engines. The customary lists of Diesel fuel-oil specifications never contained such an item as ignitibility; therefore they are now considered inadequate. The A.S.M.E. Committee for Diesel Fuel-Oil Specifications, in cooperation with S.A.E. and A.S.T.M. committees, is giving special attention to this phase of the fuel problem. Projected research is destined to find the best ways to determine the ignitibility of an oil and to express it in comparable units. The insight gained by recent investigations already has enabled several oil companies to bring distinctly superior Diesel oils on the market. Their high price prevents their more general use as yet, but further developments are impending.

The high-speed engines are not the only ones concerned with the fuel program. An important factor in retarding development of the large marine Diesel engine here is the fuel-price situation, which in the United States shows a differential of over 100 per cent in the prices of boiler fuel and Diesel fuel in favor of the former.

The development in the future of the large Diesel engine, stationary and marine, in this country appears to be contingent on active development of methods of utilizing boiler oil for Diesel engines; this is now in progress.

The field of the oil engine is now so diversified and includes so many types that each progresses more or less along its own line. General tendencies in design that could be noted have been the almost complete displacement of the air injection by mechanical injection, further increase of speed and reduction of weight for given piston displacement, a slight shift in trend in favor of the single-acting, two-stroke cycle, the preference for enclosed engines, the increased use of alloy steels, and growing dependence on specialty manufacturers for such parts as injection pumps and nozzles.

Stationary Engines

The five 7000-hp engines for the world's largest Diesel power plant have been completed by the Hooven, Owens, Rentschler Company; they have all undergone shop tests and will be erected in Vernon, Calif., during the winter. The engines are of the double-acting, two-cycle Hamilton-M.A.N. type, having eight cylinders 24 × 36 in., running at 167 rpm.

The world's largest Diesel engine unit of 22,500 bhp has been installed by Burmeister & Wain in the Copenhagen municipal electric plant to supply energy for the peak load. (See Fig. 1.) It is of the double-acting, two-cycle type, having eight 33-in. by 59-in. cylinders, uniflow scavenging, with piston-type exhaust valves in the heads, two rotary blowers driven from auxiliary shaft, and a hydraulic oscillation damper. The guaranties call for a fuel consumption of 0.55 lb per kwhr at 15,000 kw and 0.53 lb per kwhr at 12,500 kw. Another peak-load engine of 6900 hp was installed by M.A.N. in Minden, Germany. Several engines of 3000 to 4000 hp were installed for public utilities by the Busch-Sulzer Company in this country. The qualifications of the Diesel engine for peak-load service are more and more appreciated, and the attitude of the electric utilities has undergone a perceptible change in favor of the Diesel engine in the last two years. Two factors contributed to this change: First, the initial cost of Diesels went down considerably recently. The overall cost of a complete Diesel-generating station of moderate capacity, including land and building, may now safely be estimated at about $85 per installed kilowatt, as compared with $150 only a few years ago. This reduction, which is greater than the general drop of price in machinery during the same period, can be explained by the technological advances made in the design and construction of the large Diesel engines. The adoption of solid injection, the use of trunk pistons of progressively larger diameters, and the stepping up of the rotative speed are mainly responsible for the decreased cost per kilowatt. The 3000-hp installation of Busch-Sulzer Company in the Sterling plant of the Public Service Company of Colorado, running at

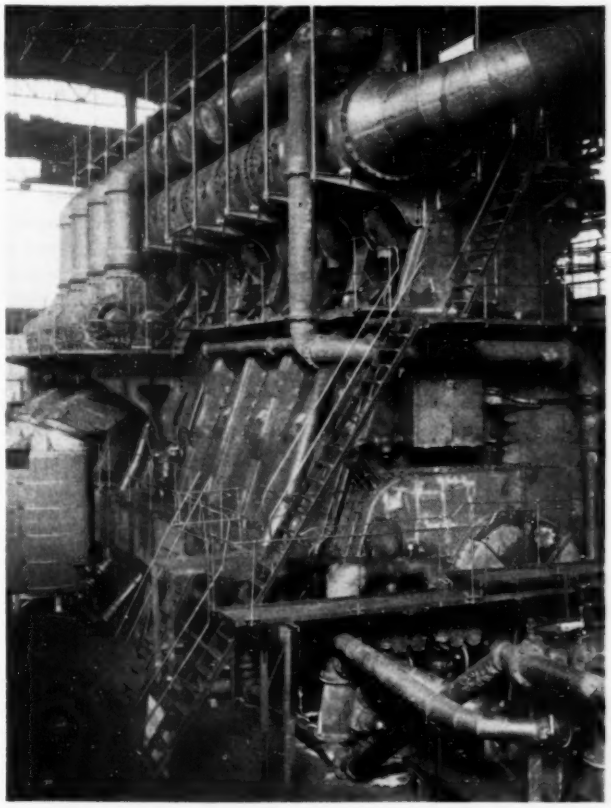


Fig. 1 World's Largest Diesel Engine, 22,500 Hp on a Single Shaft

(Being installed by Burmeister & Wain in the municipal electric plant in Copenhagen for peak-load service.)

Presented at the session of the Oil and Gas Power Division, Wednesday afternoon, December 7, 1932, during the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

The report was prepared by Dr. P. H. Schweitzer, Associate Professor of Engineering Research, School of Engineering, The Pennsylvania State College, State College, Pa.

Personnel of the Executive Committee of the Oil and Gas Power Division for 1932: Louis R. Ford, Chairman; Edgar J. Kates, Secretary; Harte Cooke, L. M. Goldsmith, Lawrence B. Jackson, and Ralph Miller. Associates: L. H. Morrison and Arthur M. Tode.

240 rpm, and the four-unit plant, totaling 2760 hp, for the Commander Milling Company of Minneapolis by Fairbanks-Morse Company, running at 277 rpm, are cited as examples. The second factor is the increased amount of reliable information now available on the operating costs of Diesel engines. The A.S.M.E. Committee on Oil Engine Power Costs has collected and recently published an immense amount of carefully analyzed data, filling a long-felt need of the power producers. The study of that report is recommended to everybody interested in Diesel-power costs.

Marine Oil Engines[1]

The world-wide business depression reduced shipbuilding to the lowest level reached in half a century. Progress in marine Diesel engineering, as well as in all other phases of marine engineering, has been retarded, but this condition has also had the effect of emphasizing the inherent economic advantage of the Diesel engine. Between July, 1931, and July, 1932, the gross tonnage of all steamships decreased by 960,473, while the oil-engined ships increased in tonnage by 602,064. Motorships have been able to continue trading with reasonable earnings in a market that has necessitated laying up some 12,000,000 gross tons of steamers.

The outstanding marine Diesel installations of the year are found in the German naval vessel *Bremse*, the British passenger liner *Georgic*, and the Italian passenger liner *Neptunia*, each representing a different type of Diesel-engine installation. The *Bremse* is equipped with 26,000 bhp in the form of eight two-cycle, double-acting M.A.N. eight-cylinder engines, operating at 600 rpm, and connected to two propeller shafts through Vulcan gearing. The *Georgic* (27,000 gross tons) is driven by two B. & W. four-cycle, double-acting 10-cylinder Diesel engines, each developing 10,000 bhp and direct-connected to the propeller shafts. The *Neptunia* (20,000 gross tons) is fitted with four Sulzer two-cycle, single-acting Diesels direct-connected to four propeller shafts, developing a total of 18,000 bhp.

The success of the light-weight German naval engines is attested by the fact that a second and third sister ship of the cruiser *Deutschland* are being laid down, fitted with similar machinery, which consists of eight 7100-hp M.A.N. double-acting, two-cycle engines, with return-flow scavenging and rotary valves in the exhaust ports, each engine having nine 16.5 by 22.8-in. cylinders, totaling 56,000 hp, and the auxiliaries consisting of eight 250-kw Diesel-driven dynamos. The engines for these dynamos, built by Linke-Hoffman, are six-cylinder, single-acting, four-cycle, cross-head, solid-injection units running at 1000 rpm. They develop at that speed 375 to 400 bhp, and the weight per bhp, without the dynamos, is about the same as that of the main engines, i.e., 17.6 lb. The *Deutschland* machinery stood the acceptance tests excellently, the observed full-load fuel consumption being 0.385 lb per bhp-hr for the main engines, including scavenging, and 0.395 lb per bhp-hr for the auxiliaries.

Development of the marine Diesel engine along lines calculated to indicate the most suitable types of engines for various services has been carried out by Japan more systematically than in any other country. Japanese government departments concerned with the shipping industry have placed in service different types of Diesel-engine vessels specially designed to develop the possibilities and increase the national knowledge of marine Diesel engineering.

The United States has continued to lag behind the rest of the world in the development of large marine Diesel engines, but has made a creditable showing with small engines. A recent Department of Commerce report shows that in the past 10 years the number of Diesel-driven vessels increased from 95 to 3458.

[1] Largely from report by Louis R. Ford.

In regard to construction, the welded-steel frame constitutes the most noteworthy innovation that has originated in the last year or so from numerous sources. The result obtained in the way of weight reduction is little short of miraculous. The engines of the *Bremse* weigh only 14 lb per bhp, while both M.A.N. and Sulzer are developing engines of 2000 bhp per cylinder at 265 rpm and weighing around 22 lb per bhp.[2] Although such engines are not yet built for commercial purposes, the fact that engines are constructed of the power mentioned, with about one-tenth of the weight that a few years ago was called normal, opens immense possibilities.

A somewhat similar development approved by commercial practice is the "hydrogen brazing" of engine parts, primarily cylinder heads, introduced by the German A.E.G. Instead of a casting, several simple forgings are made and united in a very effective manner by the new method.

Torsional vibrations, old well-known trouble makers for long multicylinder engines, are now well in hand. Vibration dampers of the hydraulic types are finding extended use. Supercharging is gaining, and also the exhaust-gas utilization, which permits very attractive economy on board. It may be mentioned that in the contract for the engines of the motorship *Berganger*, a vessel now approaching completion in Denmark, it is stipulated that the fuel consumption shall not exceed 0.35 lb per bhp-hr.

Automotive Diesel Engines

In appearance and construction the giant Diesels have little in common with the Diesel engines used in trucks and buses, of which now about 3500 are in operation in Germany alone. The General Omnibus Company in London is gradually replacing its gasoline engines with Diesels. In America at least four makes of trucks are now offered with Diesel-engine drive, and during the last year quite a number of Diesel-equipped American tractors have been shipped to many parts of the world. The saving in fuel costs claimed with Diesel-engined vehicles is as high as 70 to 80 per cent, while in spite of views expressed to the contrary, in acceleration and flexibility the Diesel truck excels the gasoline truck of equal power, as has been evidenced by Professor Langer's systematic tests and the favorable experiences of American fleet operators with Cummins Diesel-equipped trucks, one of which holds a non-stop record of 13,500 miles (December, 1931). The majority of the automotive Diesel engines are around 80 hp, but now demand is being expressed for more powerful engines. The top speed of 2000 rpm is today no more exceptional, and the engine weights approximate those of gasoline engines. The automotive industry begins to take the Diesel very seriously, and experimentation is going on behind closed doors in many plants.

While in large Diesels the direct-injection and undivided combustion chamber is sovereign, in automotive engines there is a fight among the various systems. At this time the two-chamber principle seems to be ahead of its rival, as is exemplified by such recently developed successful types as the new Mercedes-Benz, Linke-Hoffmann, Oberhaensli, Dorman-Ricardo, and Ganz-Jendrassik engines.

The open combustion chamber seems to hold its best in opposed-piston engines of the Junkers type and its various modifications, like the German Michel engine, which has a common combustion space for a Y-shaped cylinder and was subjected to successful tests recently. Distinct from this is the crankless Michell engine, being developed in this country, which also has opposed pistons.

The economy and performance of the automotive Diesel is no more questioned. Where it fights its battle is on the plane of knock, smoke, odor, and ease of starting. All of these are

[2] Several companies are now prepared to build 40,000-hp engines.

more or less related to the problem of ignition lag, and here the cooperation of the engine designer with the pump specialist and the fuel technologist is bound to give the answer.

Aircraft Oil Engines

During the last year several new types have been reported as having been developed or as being developed, but little progress is noticed in the actual use of the Diesel engine in the air. We know of a 720-hp Junkers engine in line service between Berlin and Amsterdam, having 120 hours in the air behind.

A notable development is the Guiberson radial air-cooled Diesel, somewhat along the Packard line, while several other American aero-engine manufacturers experiment with spark-ignition, solid-injection engines.

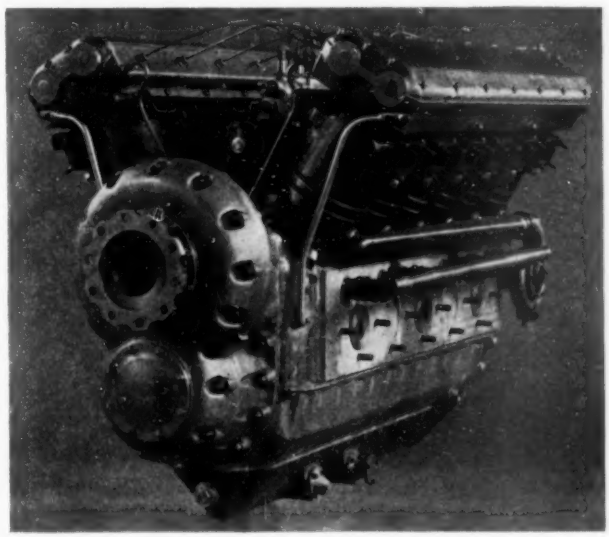


Fig. 2 Daimler-Benz 750-Hp Aviation Diesel Engine

Abroad, the Daimler-Benz Company is ready to announce the completion of a 750-hp aviation engine. It is a water-cooled 12-cylinder V-engine, 6.5 in. bore, 8.3 in. stroke, with two overhead camshafts and two inlet and exhaust valves for each cylinder. Split roller bearings are used for the crankshaft and Bosch pumps for the injection of the fuel. The whole engine weighs 2100 lb. It develops 700 hp at 1675 rpm and 750 hp at 1720 rpm crank speed, which is reduced with spur gears by 1:1.5:1.2, so that the maximum propeller speed is under 1000 rpm. The observed fuel consumption is 0.366 lb per bhp-hr at 1508 rpm and 518 hp and 0.42 lb per bhp-hr at 1753 rpm and 792 hp, which is the peak output with slightly visible smoke. (See Fig. 2.)

Very unlike to this is a 1000-hp, 7-cylinder airship engine on the test floor of the M.A.N. Company. While the Daimler-Benz engine follows gasoline-engine lines, making liberal use of the light alloys, the M.A.N. development, a supercharged double-acting, two-cycle job, is almost exclusively out of steel, with through anchor bolts along the line of their welded battleship engines.

According to unauthorized information, the engine that ran over 70 hours already develops a peak output of 1200 hp at 1000 rpm, and its total weight is below 4.4 lb per bhp. It is expected that it will add much to the fire safety of the new German zeppelins.

Development of aviation Diesel engines is also reported from England, France, and Italy, but if any outstanding success was attained, they have succeeded in keeping it secret. Yet there is little doubt that injection engines will displace carburetor engines in the air before long.

Locomotive Engines

It is exceptionally unfortunate that just when railroads had convinced themselves that the Diesel locomotive is the only logical switching locomotive, they do not have the money to buy them. As an exception 37 Diesel locomotives have been installed for operation in the freight yards on the west side of New York City by the New York Central. In spite of lack of orders, several engine builders are at work developing locomotives, and some of them are known to be successful, technically speaking.

While main-line Diesel locomotives still seem to be some distance off, a very promising application is that to the railcar. In the last years more and more branch-line passenger traffic has been lost to the railroads. To ease this situation the railroads frequently have gone into the bus-operating business themselves. Recently it became apparent that the railroads need not go off the rail to meet bus competition, if they apply the light-weight, high-speed principle to their equipment. What can be done is shown impressively by the Budd-Michelin 40-passenger railcar, one of which was put into regular service recently by the Pennsylvania Railroad in Philadelphia. Its weight is 14,000 lb, instead of the 175,000 lb usual for rail coaches of similar size of or 90,000-lb, the weight of the average railcar in branch-line operation. It is a one-man-operated unit, instead of having the crew of four required for steam equipment. The unusual weight reduction is attained by the use of pneumatic tires, a thin welded-steel-sheet construction, and a light 90-hp Diesel engine of the Junkers opposed-piston type. The operating expense of such a railcar, including wages, fuel, repairs, supplies, interest, and depreciation, is calculated as 20.5 cents per train-mile, compared with 26 cents for motorbus, 55 cents for the orthodox railcar, and $1.02 for steam trains. Of course, any good automotive Diesel engine is fit for railcar service, which fact is apt to furnish incentive for such development.

Research

Mention has already been made of the recent Diesel fuel research. Investigations by Le Mesurier and Stansfield, of the Anglo-Persian Oil Company, and Boerlage and Broeze, of the Royal Dutch Shell Company, increased our knowledge of fuel and combustion considerably. Neumann's theoretical work and Pope and Murdock's experimental work also have contributed to it. The point is now reached that fuels can be evaluated on a firm basis and steps can be taken to control their ignition in the engine. "Dope" for Diesel fuels is an imminent possibility.

National Advisory Committee for Aeronautics

The National Advisory Committee for Aeronautics attacked a number of problems during the year. A most outstanding result of their high-speed photographic research relates to fuel vaporization. Contrary to prevalent opinion, Rothrock found that the vaporization of the fuel is extremely fast and practically complete before ignition occurs in the engine. Pressure waves and injection phenomena received much attention as also did combustion analysis. The engine research of the committee was divided between the quiescent and the turbulent engine, both being investigated methodically.

Pennsylvania State College

Pennsylvania State College completed its investigation on spray dispersion. The information made available, complemented by Holfelder's data, answers most questions, theoretical and practical, in regard to the dispersion of oil sprays. Liquid turbulence was found to be the primary and air friction the contributory cause for spray dispersion.

The Massachusetts Institute of Technology obtained interesting results with a fuel-injection spark-ignition engine.

In the Dresden Technische Hochschule, Schmidt conducted a valuable investigation on the temperature fluctuations at various points of the combustion space. Haenlein and Holfelder obtained interesting data on fuel sprays. In Hannover, Neumann and Klüsener investigated the charging and scavenging process. Injection pumps and nozzles received the attention of several investigators in England and Germany.

Gas Engines

Two-cycle gas engines again made their appearance, but of much improved fuel economy. M.A.N. is applying its reverse scavenging principle to the gas engine, obtaining four-cycle economy with much reduced first costs.

Clark Brothers Company developed the "injection gas engine," in which the gas is introduced by a pump after the exhaust ports are covered by the piston, and thermal efficiencies exceeding those of four-cycle engines are obtained.

Business Outlook

The present economic crisis did not spare the oil and gas power industry, although it probably did not hit it more than other machinery industries. Even if the immediate future is rather uncertain, the distant future is distinctly bright for the survivors. The triumph of the Diesel engine may duplicate that of the electric motor, as the field is beginning to be ready for it. The Diesel Engine Manufacturers Association elected Mr. H. Birchard Taylor for president, and a broadening of its activities is being contemplated.

OGP-55-3

Improving Operation and Overall Economy of Diesel-Engine-Driven Generating Stations

These Conditions May Be Attained Through the Use of Selected Boiler Fuel Oils and More Intelligent Supervision

By H. W. STALNAKER,[1] OSCEOLA, IOWA

The majority of Diesel plants are operating far below par, and if real economical operation is to be obtained, better supervision is necessary. The larger plants can afford to retain engineers capable of producing results. Small plants could engage supervising engineering companies on a percentage of the net savings. If boiler fuel oil could be proved to operate more economically, it should be used.



WHEN we look back over the 1930–1931 operating-cost records of even the most modern Diesel-engine-driven generating stations, as compiled by the A.S.M.E. Diesel Cost Committee, there can be but little doubt of the vital necessity for cheaper Diesel fuel oils and more efficient operating methods. Indeed, unless some such changes are made in the very near future, the Diesel-engine manufacturers are going to find some of the most fertile fields closed to the use of Diesel engines. It is quite obvious that prospective customers of Diesel engines must be shown authentic operating-cost records of plants in their locality to substantiate the estimated operating costs as compiled by the manufacturers' sales organizations. In the Eastern part of the United States, where the so-called good Diesel fuel oil costs 5.5 to 6 cents per gallon and where the on-the-switchboard cost per kwhr is well above 1 cent, can these authentic records be obtained? The three largest and most modern Diesel plants in the East are all municipal owned. Their generating costs are 8.20 mills, 10.29 mills, and 13.86 mills. None of these generating costs are in line with what it costs to generate 1 kwhr in the modern central-station steam plant; neither are they in line with what it costs to generate 1 kwhr in a modern Diesel plant where the cost of fuel oil is more reasonable. Such generating costs, when fixed charges and distribution costs are added, are prohibitive. For the latter reason many of the larger electrical utility companies, particularly in the East, refuse even to consider the operation of Diesel engines except for standby service.

This paper is intended to show that Diesel-plant operation can be improved and that the present more-or-less closed fields for Diesel-engine installation can be opened up. While the paper contains many positive statements of a highly controversial nature, they reflect actual experience of the author while in the employ of one of the large electrical utility companies and while using the heaviest of boiler fuel oils as Diesel fuel.

The two chief reasons for high generating costs in Diesel-driven generating stations are the high cost of so-called good Diesel fuel and the employment of unskilled supervisors, the latter usually having been recommended by Diesel-engine builders. To unskilled supervision can also be attributed low thermal efficiency, high maintenance costs, and high fixed charges—the latter on account of the necessity for installing standby generating capacity to take care of service in case of mechanical failures.

In the Eastern part of the United States Diesel fuel oil that will meet the specifications of the Diesel-engine manufacturers costs about 40 cents per 1,000,000 Btu, while the best qualities of steam coal cost but 8 cents per 1,000,000 Btu. In other words, the Diesel operator must pay about five times as much for fuel as the steam plant pays. Then consider what the difference in thermal efficiency is between steam and Diesel plants.

A few years ago about 30,000 Btu was required to produce 1 kwhr in the central-station steam plants, and the Diesel plant of even small size required but 12,000 Btu for the same work. However, today the modern central-station steam plant of even medium size requires but 15,000 Btu, and in some cases even less. On the other hand, the Diesel plant of even the largest size remains at 12,000 Btu per kwhr in even the most efficiently operated plants. In other words, while the efficiency of the steam plant has been increased by more than 100 per cent, the Diesel has remained at a standstill. In view of these facts it begins to look as though it will be but a short time until the steam plant will have surpassed the Diesel in thermal efficiency, just as it already in many cases has surpassed it in overall economy. Such being the case, it is quite obvious that a centrally located steam plant of, say, 25,000 kwhr capacity can supply electrical energy to two or more small cities far cheaper than it can be produced in Diesel-driven municipal-owned plants operating at the present high generating cost per kwhr.

Even in view of these startling facts the Diesel manufacturers are very reluctant to turn to the cheaper boiler fuels as a possible means of bettering conditions. They still seem to think that they can still further reduce overall generating costs by lower initial cost per horsepower and lower labor costs by practically automatic operation. Again referring to fuel oil, the Diesel manufacturers are even unable to agree on any one standard specification, but rather each Diesel manufacturer has his own specification and insists that his engines be operated on fuel of such characteristics, otherwise they wash their hands of any

[1] Consulting Diesel Engineer. H. W. Stalnaker was born in Charleston, W. Va. He served apprenticeship as machinist, and took home-study courses in mechanical and chemical engineering. During the World War he was assistant superintendent, Poison Gas Division, Rollin Chemical Corporation, and was sent to England to join the staff of the Hedworth Barium Company, Newcastle-on-Tyne, as plant-betterment engineer. Returning to the United States in 1921, he joined the Laurie Coal Company in West Virginia as chief engineer. In 1923 he was appointed chief engineer and superintendent of power of a string of Diesel properties operated by the Haytien Corporation of America, Port au Prince, Haiti. While in the West Indies he also acted as consulting engineer for several smaller companies. He returned to the United States in 1930, and has since been associated with the Gulf and Great Lakes Pipe Line Companies.

Presented at the Fifth National Meeting of the Oil and Gas Power Division of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, State College, Pa., June 8–11, 1932.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

troubles that may develop. Such conditions of course go a long way toward keeping up the prices of the light Diesel fuel oils. The oil companies complain that such specifications compel them to store large quantities of many different-quality fuel oils and that this, added to the refinery costs, is what keeps up the prices. Whether this is true is of course open to argument, but nevertheless it is a good alibi, and the Diesel manufacturers would better conditions quite a little if they would get together and decide on one specification for light-duty medium to high-speed Diesels and still another for the heavy-duty slow-speed engines. So far but one manufacturer of Diesel engines has shown any inclination toward very broad specifications. His specification is, "Our engines will operate on any commercial fuel oil or crude produced in the United States or Mexico." It is very clear that such broad specifications if adopted by the rest of the Diesel industry would go a long way toward lowering the cost per Btu. In other words, the manufacturers of Diesel engines have built an engine and expect the oil companies to produce a fuel to operate it with. Why not the other way around? Build an engine with an injection system that will operate efficiently on the fuel oils that the oil companies at present produce. However, there are quite a few slow-speed heavy-duty engines that will handle the heavier fuel oils with but few changes, all of a minor nature. However, as I have previously mentioned, these same manufacturers are slow to take advantage of the cheaper fuel oils.

Take the residual fuel oils from the low-pressure stills: These residues are all pumped to the same storage tanks, and the resulting mixture is marketed as Bunker C. This fuel, which is almost a drug on the market, due to the ever-increasing use of pulverized coal in the large steam stations, can be purchased for about one-half of what the lighter fuel oils sell for. Bunker C, when properly prepared by passing it through a centrifuge and when heated to give the desired viscosity, makes a very suitable fuel for slow-speed heavy-duty air- or mechanical-injection Diesel engines. That such engines will operate efficiently, with 100 per cent reliability, and with very low overall generating costs, can probably best be shown by a comparison of plants operating on such fuel oils with plants operating on the best qualities of so-called good Diesel fuel oil.

Five of the plants used for comparison are taken from the yearly A.S.M.E. Cost Committee Report for 1930–1931, and two plant records are from plants operated by a large electric utility company in the West Indies. The author was connected with the latter for quite a few years. These records are given in Table 1.

TABLE 1

A.S.M.E. plant	Kwhr generated	Kwhr per gal fuel	Generating cost, mills	Fixed charges, mills	Total cost, mills	Maintenance hp per year, dollars	Fuel cost per gal, cents	Load factor, per cent
Municipal-owned plants operating on best quality 24 to 36 B Diesel fuel oils								
45	7,000,000	10.25	8.20	7.50	15.70	1.65	5.19	69.50
52	7,000,000	10.75	10.30	9.10	19.40	0.89	5.70	75.20
82	8,000,000	11.13	13.86	11.75	25.61	2.74	5.70	65.00
Municipal-owned plant operating on 11 to 14 B boiler fuel oil, 18,150 Btu per lb								
73	8,500,000	10.26	4.59	7.50	11.69	0.17	1.30	64.00
Privately owned utility plants operating on 12 to 20 B boiler fuel oil								
92	10,500,000	9.66	4.08	5.82	9.90	0.54	2.09	65.00
1*	5,500,000	11.88	7.66	5.60	13.26	1.50	6.00	72.65
2*	530,000	11.00	7.29	6.22	13.51	0.17	5.19	74.50

* Operated by a large electrical utility company in the West Indies. Maintenance charge in plant 1 is estimated at $1.50 per hp per year over a five-year period. Actual costs are well within this limit. Analysis of fuel oil as supplied to plants in the West Indies by the Shell Oil Company is: Baumé gravity, at 80 F, 15.1; flash point, Cleveland, 270 F; fire point, 310 F; water and sediment, A.S.T.M. D-96-30, 0.70 per cent; viscosity, Saybolt Universal at 100 F, 6054 sec; ash, 0.10 per cent; carbon residue, Conradson, A.S.T.M. D-189-30, 10.88 per cent sulphur, 2.04 per cent.

The engines represented in the comparison in Table 1 are: McIntosh-Seymour, De La Vergne, Nordberg, and Busch-Sulzer; speed range, 125 rpm to 360 rpm. However, with the 360-rpm engine the piston speed is 1200 fpm, and the operation on boiler fuel oil is very unsatisfactory; the liner and piston-ring wear is more than double that of the slower speed engines. From all indications the maintenance costs per horsepower per year will be in excess of $3 over a five-year period (in other words, liners will have to be replaced or rebored every $2^1/_2$ years).

Plants 52 and 82 are each at the present time installing additional 3000-hp units, although their annual plant-load factors are but 34 per cent. What is going to happen to the fixed charges in these plants when this installation work is finished? (Plant 82 has been finished.)

As to reliability, the only shutdown was one of two hours' duration in plant 52, which is one of the light-fuel plants. None of the plants operating on boiler fuel oil reported any shutdowns. It would seem from this that 100 per cent reliability can be expected, even when operating on the heaviest of boiler fuel oils.

From the foregoing comparison it looks as though the heavy boiler fuel oil plants have the best of it when it comes to maintenance. However, the author does not wish to be quoted as saying that maintenance with boiler fuel will be less than with light Diesel fuel oils. Rather, the opposite will be the case. When operating on light fuel oil, engines of the type used in the plants taken for comparison should be maintained for $1 to $1.25 per hp per year, depending of course on operating conditions and type of supervision. On the other hand, the same engines operated on boiler fuel oil will show a maintenance charge of very close to $1.50 per hp per year, and then only when the operating force does all maintenance work and a capable Diesel man is employed as supervisor.

Of course in either case much depends on what the machine-shop facilities are. There are many very large savings that can be made through having adequate machine-shop facilities; 500 per cent on fuel-pump plungers, valve, and valve-cage repairs, bushings, and many other small parts is not uncommon, while savings on bearings, regrooving of piston-ring slots, making of special tools, etc. will be but slightly less. A lathe, drill press, tool-post grinder, shaper, and portable grinder will cost but little when the many advantages and savings are taken into consideration.

Continuing along the same lines, of the larger plants operating in the East, the A.S.M.E. Cost Committee plant 45 is by far the most efficiently operated of any shown in the 1930–1931 report, and it also has the lowest generating cost. However, even in view of this being the most efficient plant, there are still some very attractive savings that can be made in two ways; one through increased thermal efficiency (report shows plant to be far below manufacturer's guarantee) and another through the use of cheap boiler fuel oil. The author would like to point out that the actual kwhr generated is not the true load on the engine shaft. Motor-generator sets are used for excitation, and when this is deducted, the actual effective load will be more like 7,000,000 kwhr, and this still leaves some 91,500 kwhr for power and lights used in production. Still another thing to point out is that this plant uses a 35 B gas oil which weighs but 7.065 lb per gal. Another thing is that these engines are of the four-cycle air-injection type and operate at a slow speed. Engines of such type will not as a rule operate satisfactorily and give their guaranteed thermal efficiency on such light fuel oils. The injection period of time is much longer than with engines operating at higher speeds, and the loss of high-pressure air during injection is quite large. Fuel pumps also give lots of trouble due to the very light fuel cutting the plungers, and the packing must be kept very tight to prevent leakage. However, on the other hand, if an engineer with sufficient Diesel experience is employed as supervisor, he can, by use of an indicator, set the

injection valve so that good thermal efficiency can be obtained. But how many plants employ such a type of engineer? In any case it is necessary to watch the fuel-valve settings very closely, as a very small change in setting will often make as much as 10 per cent difference in fuel consumption. Employment of inexperienced engineers can usually be traced to the Diesel manufacturer. Quoting one of the manufacturers of high-grade engines, "A Diesel plant is so simple that a week or two of training will make a good attendant of any man with average intelligence and mechanically inclined." They fail, however, to say anything about supervision. They also go on to say that plants can be operated on 24-hour basis with but two men. Such conditions are largely responsible for high maintenance and generating costs.

Analysis of power costs in plant 45 is given in Table 2. Column 1 shows actual costs; 2, possible costs with expert supervision; 3, possible generating costs with expert supervision and using boiler fuel oil. Maintenance costs in the estimated part of the analysis are based on $1 per hp per year with light fuel oil and $1.50 per hp per year when using boiler fuel. Estimated efficiencies are based on 0.43 lb of 18,500 Btu fuel per hp-hr at 75 per cent running plant-load factor (manufacturer's guarantee). A 100 per cent perfect combustion cannot of course be obtained with boiler fuel oil (using manufacturer's guarantee for comparison); due allowance has been made. A different distribution of production supplies and miscellaneous expense has also been made in the estimated analysis, so that it will be more in accordance with "Classifications of accounts for electrical utilities." (See Table 2.)

TABLE 2 PLANT 45, A.S.M.E. DIESEL COST COMMITTEE REPORT, 1930–1931

	(1) Actual cost	(2) If efficiently operated	(3) Possible with heavy boiler fuel oil
Kwhr generated	7,492,325		
Kwhr used in station:			
Lights and power	91,500		
Motor-generator sets	492,325		
Actual load on engine, kw	7,000,000	7,000,000	7,000,000
Actual load on engine, hp	10,290,000	10,290,000	10,290,000
Fuel oil used, gal	676,475	620,567	579,480
Fuel oil used, lb	4,779,296	4,384,305	4,659,598
Actual kw per gal fuel	10.348	11.280	12.080
Lb fuel per kw	0.682	0.626	0.665
Lb fuel per hp	0.459	0.425	0.452
Lubricating oil used, gal	1,245	1,245	1,245
Kw per gal lubricating oil	5,540	5,540	5,540
Btu per kwhr	13,056	11,965	12,290
Thermal efficiency, per cent	26.150	28.540	27.780
Running plant-load factor per cent	69.500	69.500	69.500
Weight of fuel per gal, lb	7.065	7.065	8.044
Btu per lb of fuel	19,000+	19,000+	18,500
Btu per gal of fuel	135,000	135,000	144,000
Fuel oil cost, cents per gal	5.190	5.190	3.000
Costs:			
Labor per cent supervision	$12,228.05	$12,228.05	$12,228.05
Fuel oil, including handling	35,760.05	32,207.43	17,384.40
Maintenance of engines	6,977.60	4,250.00	6,375.00
Other plant maintenance	690.85	350.00	350.00
Miscellaneous supplies and expense	69.80	350.00	350.00
Lubricating oil	621.75	621.75	621.75
All other expense	none	100.00	100.00
Total operation and maintenance	$56,348.10	$50,107.23	$37,409.20
Total cost in mills per kw	8.05	7.16	5.34

The savings that can be made in many of the present Diesel-generating stations are very attractive. However, within the next two or three years, or perhaps even much sooner, they will be not only attractive, but a necessity. Taxes on imported crudes will raise the price of fuel. And we all realize that but a short time will see at least a partial application of the Diesel engine in the automotive and airplane industries. In fact, airplane and truck engines are already in commercial production. What is going to happen to the prices of the so-called good Diesel fuel oil which is the type of fuel that must be used in these light high-speed engines? It is not unreasonable to suppose that the fuel consumption of these engines will amount to several hundred million gallons per month. Boiler fuel oil will, however, be even more of a drug on the market than it is now unless a market can be opened up for it.

Before making any attempt to burn boiler fuel oil in Diesel engines—providing the engine is of a type that will handle the boiler fuel oil efficiently—the following provisions should be made:

Of prime importance is the supervision of the plant. A capable Diesel engineer should be employed of a little higher caliber than usually found in the average Diesel plant. He should have at least a high-school education and preferably a technical education. He should have a record of being able to obtain the highest possible efficiencies when operating Diesel engines on the lighter fuel oils. In other words, he should be able to make the engines live up to the manufacturer's guarantee or better. Any engineer that operates a Diesel plant and gets but 11 kwhr per gal of fuel when he should get 12 certainly should not be trusted to operate a plant on boiler fuel. He should also be a first-class machinist and able to do all maintenance and repair work, regardless of its nature. Unless a man with such characteristics is employed, it will usually be found that the heavy-fuel project is a failure even before the first car of heavy fuel oil arrives at the plant.

It is very important that a Diesel engine have plenty of excess air for any type of fuel oil. However, it is far more important that it have a very large excess when it is to operate on boiler fuel oil. Of course, the amount of excess air is to a great extent regulated by the factory air-inlet valve settings. However, in many cases air filters are installed, and often they are too small or perhaps not kept clean. Or perhaps new timing gears have been installed, and the air-inlet valve settings are not the same as the original setting. Or, again, it may be that the original factory setting was not correct. (It is the present practice of many manufacturers to ship engines without running them.) In two-cycle engines the scavenging pressure may have to be increased or the exhaust ports may be fouled with carbon. There are many factors that may have a large effect on the amount of air drawn into the cylinder.

The compression of the engine should be raised to not less than 525 lb per sq in. (by inserting shims under the foot of the connecting rod). Two-piece piston rings should be installed in the third and fourth grooves of the piston to prevent the corrosive exhaust gases (caused by sulphur in boiler fuel) from blowing by into the crankcase. In no case should two-piece piston rings be installed in the top two grooves as they will cause very rapid wear (stepping) of the lands on the piston.

A pyrometer should be installed. (This is very important with solid-injection-type engines.) An indicator must be provided. It is usually unnecessary, however, to provide an indicator motion, since the fuel-valve settings will be much more clearly shown on an out-of-phase or hand-drawn card. However, if the engine does not perform as well or better on the heavy fuel than it did on the lighter fuel oils, then an indicator motion should be installed and phase cards taken to locate the trouble. Sometimes it is necessary to provide burner plates or spray nozzles with slightly larger holes (usually only on mechanical-injection-type engines). It also may be necessary to respace the atomizer disks. Of course, none of this can be done to any set rule, but must be determined from the indicator cards taken after the engine is in operation on the heavy fuel oil.

Next we come to the necessity of cleaning and heating the boiler fuel oil. Nearly all of the commercial fuel oils reach the same viscosity around 200 to 225 F, and by raising the temperature of the boiler fuel, light fuel-oil viscosity can be duplicated. In nearly all of the heavy-duty slow-speed engines the best

operating viscosity is usually between 50 to 75 sec Saybolt, and nearly all of the commercial boiler fuel oils will reach this viscosity well below 200 F. However, even much higher temperatures are well below the flash and carbonization points. The best operating temperature for any engine can be found by actual operation. Separate fuel tanks or accurately calibrated meters must be provided for each engine and the fuel consumption checked each 24 hours. Any change in rate of consumption should immediately be investigated. It is well to remember that the products of poor combustion are very abrasive, and they are even more so with boiler fuel than with the lighter fuel oils, and that poor combustion is responsible for rapid cylinder-liner and piston-ring wear.

A very efficient and very simple type of heater is one that uses a part of the exhaust gases, and it can be made in the plant at small cost. A 15 to 20 ft length of 4-in. black wrought-iron pipe is fitted with a stuffing box at each end bored to take a 2-in. standard-pipe-size copper pipe. One end of the copper pipe is connected to the exhaust header with a gate valve in between, and the other end, after it passes through the 4-in. black pipe (or the jacket of the heater), is continued on outside the building. Two pipe connections are welded to the heater jacket, one on each end. One connection is for the fuel supply, and the other, or outlet, is to be connected to the fuel-injection pump through a Cuno or Hesselman jacketed strainer. Light fuel oil will have to be used for starting and stopping the engine. Not only does the light fuel oil make starting easier, but it clears the cylinders of all corrosive gases that would otherwise remain from combustion of sulphur in the heavy fuel oil. The light-fuel connection should be made in two places, one for normal operation and one for emergency operation. The normal connection is made to the jacket of the heater at the same point as the heavy fuel connection. The emergency connection is to be made up close to the engine injection pump. In normal operation, when the engine is to be shut down, the heat-control valve is closed, and the light fuel oil is turned on and the heavy fuel turned off. The engine is then operated until all of the heavy fuel oil in the heater has been replaced by the lighter fuel. When the engine is again to be started up, the heavy fuel oil is turned on and the heat control valve opened. By the time all of the light fuel oil in the heater is replaced by heavy fuel oil, the temperature of the heavy fuel will be high enough for efficient operation. The other light-fuel-oil connection is to be used only when the engine is to be shut down quickly, as in case of mechanical or other troubles. To prevent gas binding of the fuel pump, a vent must also be provided just as close to the suction side of the fuel pump as possible. This vent is to be run back to the fuel-supply tank up over the top on the outside. The usual operating temperature is 180 to 200 F.

Hot-water-heating systems are also used for heating heavy fuel oil in some installations. However, they are much more expensive, both in operation and in first cost. Also, since the temperature of the hot water is always constant, such a heating system is not very satisfactory where there are large and sudden changes of load; whereas with the exhaust-gas heater, the rise and fall of the load will cause a rise and fall of the exhaust-gas temperature and make the operation of the exhaust-gas heater almost automatic. In either case, all pipes carrying heavy fuel oil should be slightly larger than with light fuel oils, and all pipes carrying hot oil, exhaust gases, or hot water should be insulated. Of particular importance is the insulating of the fuel piping from the engine injection pump to the engine-fuel valve. It is very important that the temperature of the fuel at the fuel valve be but little below the temperature reached in the heater.

The heavy fuel oil should be passed through a centrifuge, although some plants operate with only a heating, settling, and filtering system. The latter system operates very satisfactorily, but the overall maintenance cost will probably be slightly higher. In either case, it is very necessary that the heat required for cleaning the fuel be obtained from the jacket water or exhaust gases, or as in most cases a combination of the two. This is for economical reasons.

It is well to remember that where engines of such type as are capable of operating on boiler fuel are to be operated on boiler fuel oil, it will be necessary for them to operate on such commercial fuel as is available if advantage is to be taken of the present very favorable market prices. Such specifications as are specified must be very broad. A fuel that the author has in mind and that he has used in McIntosh-Seymour, De La Vergne, and M.A.N. engines, and found very satisfactory, is: Baumé gravity, 12 to 16; asphaltum residue, not to exceed 15 per cent; Conradson carbon, not to exceed 8 per cent; sulphur, not to exceed 3 per cent; bottom settlings and water, not to exceed 1 per cent; viscosity and flash-point, flash-point to be above the temperature at which the viscosity reaches 50 sec Saybolt Universal.

The majority of commercial fuel oils in the East will be well within these limits. However, in order that my meaning might be perfectly clear, I would like to say that three distinctly separate fuel-oil specifications should be able to take care of the Diesel-engine industry: 1, a fuel oil for slow-speed to medium-speed heavy-duty full-Diesel-type; 2, a fuel oil for any type engine, with exception of automotive and airplane types; 3, a fuel oil for automotive, airplane, and other engines of like characteristics (the third fuel would probably be very much like the second specification, with the exception that it should be a filtered or centrifuged fuel).

In conclusion, the author would like to suggest this: If real economical operation is to be obtained in Diesel-generating plants, the majority of which are now operating way below par, then better supervision of operation is necessary. The plants of fair size can afford to employ the services of engineers capable of producing the desired results. However, on the other hand, there are many small plants that cannot (even if there were enough good Diesel men to go around). It would seem that this is a very good field for "supervisory engineering companies" to investigate, with a view of profit to themselves, operating on a percentage of the net savings made in each case. Or probably even better still would it be for the manufacturers of Diesel equipment to retain supervisory control over the smaller plants until such time as the operating force can show that they are capable of economical and reliable operation. With "supervisory" control, whether by private engineering firms, consulting engineers, or Diesel manufacturers, there would be no question of what kind of fuel oil to use or if new cylinder liners should be installed or the old ones rebored; such items would take care of themselves. If boiler fuel oil could be shown to operate more economically in the particular locality, then it is certain that boiler fuel would be the one used, just as is the case with large electrical utility companies and pipe-line companies operating long strings of stations. The Diesel-engine manufacturers, with their branch offices in all of the larger cities, should easily be able to operate such a service, and not only better the entire Diesel industry, but show a profit to themselves. Of course, however, it would be necessary for them to employ engineers of a different caliber than the usual factory erection engineers. These should be men of the same caliber as the master mechanics of the large pipe-line companies or men such as the superintendents of Diesel power for the large electrical utility companies; in other words, men that are capable of looking at the overall cost per hp-hr or kwhr, and not just the kind that says, "This cylinder liner shows lots of wear, and you had better change your quality of fuel or lubri-

cating oil and buy several hundred dollars' worth of spare parts to repair the engine."

At any rate, whether or not supervisory control is to be used, it is very obvious that much improvement in operation is necessary and is possible.

Discussion

W. L. Marston.[2] The author's statements that the cost of operating a Diesel plant can be greatly reduced by the use of heavy or boiler fuel oils is well borne out by the Preliminary Report on Oil Engine Power Costs for 1931, which has just been presented. On page 8 of this report, referring to plants 73 and 54, it will be noticed plant 73 reports a fuel cost of 0.93 mill per net kwhr. Plant No. 54 reports a fuel cost of 0.91 mills per net kwhr. These costs are less than half of any other fuel costs shown in the report, with one or two exceptions. It will be further noticed that the total production cost of these two plants is also greatly below that of any other plant. It has been stated that the use of heavy fuel oil increases the cost of maintenance. This is not borne out by the figures reported for plant 73, as its cost of engine repairs is well in line with that of any other plant. From the paper it will be seen that plant 73 uses a fuel oil of 11 to 14 deg A.P.I. Plant 54 uses a 27.2 deg A.P.I. crude oil from the pipe line.

On this same page it is stated that these two plants centrifuge their fuel oil, and with one or two exceptions they are the only plants that are centrifuging their fuel oil. This is significant. The reason is that the heavy fuel oil must be centrifuged first before it can be satisfactorily used in Diesel engines.

It is an interesting fact that on motorships the fuel-oil centrifuge is standard equipment. Practically every motorship using Diesel engines is equipped to centrifuge the fuel. Not only does this permit the ship to use a heavier gravity fuel successfully, but since there are no service stations on the high seas, it helps maintain the engine reliability at the maximum. Stationary Diesel plants likewise will reduce operating costs by using heavier gravity fuel oil. To satisfactorily use the heavier grades of oil, however, it is essential that this oil be thoroughly cleaned by centrifuging.

Edgar J. Kates.[3] Great emphasis is laid in this paper on the importance of using residual fuel oils (so-called boiler fuel oil) in order to reduce operating costs in Diesel-generating stations. However, circumstances alter cases, and the writer feels that exception should be taken to categorical statements favoring the universal use of heavy fuel oils. The question is an economic one, and while under certain operating conditions the use of cheap residual fuels is well justified, under other conditions it is far more desirable to use light fuels.

Heavy oils can be used to advantage in large engines operating at continuous heavy loads; here the cheapness of the fuel more than offsets the cost of preparing it for the engine and the increased cost of maintenance. Thus heavy oils are indicated for ocean-going motorships and for such stationary generating plants as the one described in *Power*, February 2, 1932, where a 2250-kw Diesel plant enjoyed an annual capacity factor of 71 per cent, and boiler oil was 1.95 cents per gallon cheaper than light oil. Here the difference in cost in favor of boiler oil was over $24,000 per year. On the other hand, the A.S.M.E. Oil Engine Power Cost Report contains records of small generating plants with poor load factors, which consume annually less than $2000 worth of 32 B oil or lighter, costing from 5.5 cents per gallon down to 2.52 cents. Such plants are often located where tank-truck deliveries of household furnace oil can be readily obtained, whereas no facilities exist for the delivery of small quantities of boiler oil. Even if boiler oil could be obtained for a trifle less per gallon it is obvious that the small annual saving thus effected would not nearly offset the nuisance of handling, the cost of preheating, settling and cleaning, and the increased cost of supervision and maintenance.

In the writer's opinion the latter part of the paper is the more valuable, giving as it does many excellent practical suggestions for the satisfactory use of residual fuels, which advice is evidently based on the author's own intimate experience.

It is unfortunate that the first part of the paper contains a number of statements that cannot be verified, and that may weaken the force of the author's justified plea for more general consideration of the question of using residual oils for Diesel fuel. For instance, in comparing the relative prices of high-grade Diesel fuel oil and steam coal in the Eastern part of the United States, he states that Diesel fuel costs about 40 cents per million Btu, while steam coal costs only 8 cents per million. These figures show that the author assumed Diesel fuel to cost about $5^1/_2$ cents per gallon and steam coal about $2 per ton. The facts are that in the vicinity of New York the average cost of high-grade Diesel fuel was about $4^3/_4$ cents in 1931 (it is still cheaper now), while good steam coal cost over $4 per ton. Using the corrected prices, Diesel fuel costs about 35 cents per million Btu and steam coal about 16 cents, which is a ratio of only 2.2 to 1, as compared with 5 to 1 as stated by the author.

Furthermore, the author does not give full credit to the modern Diesel engine for its actual thermal efficiency. He says that even in the largest and most efficiently operated Diesel plants 12,000 Btu is required to produce 1 kwhr. Reference to the A.S.M.E. Oil Engine Power Cost Report for 1931 shows 10 plants which did better than 12,000 Btu per kwhr, one of them (No. 164) doing 10,680.

Author's Closure

Mr. Marston points out that the statements relative to lower generating costs prevailing in the plants using heavy residual fuel oils are well borne out in the 1931 Report of the Power Cost Committee. It should also be noted that there are five plants in addition to plant 73 cited by Mr. Marston that are using fuel oil heavier than 20 deg A.P.I., and in each case not only is the fuel cost per kilowatt-hour lower than in plants using light Diesel fuel, but the overall on the switchboard cost is also well below the light fuel plants. In a discussion of this sort thermal efficiencies, maintenance, and other plant expense can be disregarded, since the subject is an economical one. The important issue is, what is the overall cost per net kilowatt-hour generated?

Mr. Marston points out that plant 73 using 11 to 14 deg A.P.I. fuel has maintenance costs in line with other plants and that the statement relative to maintenance being higher in heavy fuel plants is not borne out. The costs for maintenance in plant 73 are for parts alone, and the operating crew does the repair work, as it should. The actual cost of maintenance, however, will be the average cost per kilowatt-hour, or rather horsepower-hour per year—the average after the engines are given their 3 to 5 year overhaul and new cylinder liners have been installed. All Diesel maintenance, whether light fuel or heavy fuel is used, must be kept in cycles if the information is to be of any use, so far as statistics are concerned.

For example, Mr. Lee Schneitter[4] has told of a plant operated in Florida on heavy fuel. It is a plant with two-cycle, air-injection engines and operates under heavy load continuously. New cylinder liners are required every two years. This plant's main-

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[3] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.
[4] Diesel Engineer, Electric Bond and Share Company, New York, N. Y. Assoc-Mem. A.S.M.E.

tenance cycle is but two years. On the other hand, the author has operated four-cycle air-injection engines on heavy fuel where the cycle is five years. The cycle will vary with type of engine and supervision.

Heavy fuel oils for Diesel-engine use must be cleaned in some way, and there is little question that a centrifuge is the most efficient as well as the most economical way of doing it. That clean fuel plays an important part in Diesel-engine cylinder wear was brought out in an article in *Motorship* some time ago. Two sister motorships, one equipped with centrifuges for fuel and the other not so equipped, made trips around the world. At the completion of the trips the one with centrifuges showed but very little cylinder wear, while the other not equipped with centrifuges had to have a new set of cylinder liners. Intelligent supervision is the deciding factor of what kind of fuel should be used and whether or not it should be centrifuged.

Mr. Kates seems to feel that the author is making a plea for the general use of residual fuel oils (so-called boiler fuel). Such is not the intention. The author has pointed out that plants that are using boiler fuel show better economy than plants using light Diesel fuel and that they are showing reliable operation with maintenance costs in line (so far as the A.S.M.E. Power Cost Report shows) with other plants. The author has emphasized the fact that generating costs are, in the majority of Diesel plants, too high and that cheaper fuel oil will help reduce them. He has also emphasized the importance of better plant supervision and that it will not be a question of what kind of fuel to use with good supervision.

It is very true that small Diesel plants would not find any saving in using heavy fuel costing a "trifle" less per gallon than light Diesel fuel, but even small plants would find a saving where the difference is 100 per cent, as is the case along the Atlantic seaboard.

A plant that is generating 2,250,000 kwhr yearly would be considered a small plant, probably two 500 hp engines, and would consume about 200,000 gal of fuel oil at 5 cents per gallon, or a total of $10,000. With the same rate of consumption the same quantity of boiler fuel would cost but $5000. The saving would pay for lots of maintenance and still leave a very attractive profit. Where the cost differential is but slight, of course the heavy fuel oil would lose its attraction.

The author is mistaken in the statement that there is such a large difference in the price of Diesel fuel and of good steam coal. He had in mind certain strategically located steam plants. Such a comparison is of course out of place. He feels, however, that all of the remainder of the first part of the paper is easily verified. Mr. Kates takes exception to the statement that 12,000 Btu is required to produce a kilowatt-hour with Diesel engines. There are of course exceptions to any rule. The plant 164 referred to is one. However, there are nine other plants shown in the Cost Committee's report showing a better performance than 12,000 Btu, but each one has a very large amount of power charged against production, and the latter is a very unfair charge. Motor-driven exciters and scavenging blowers should not be included in the gross generated. When such deductions are made, it will be seen that in each case more than 12,000 Btu are required per kilowatt-hour. Not more than 2 per cent of the total generated load should be charged to production, and if it is necessary, then the Diesel engine loses a lot of its attraction as being a self-contained type of prime mover. Where fuel or lubricating oil is to be heated for centrifuging, it can just as easily be done with exhaust gas heat and at no additional cost. The net kwhr is what one must reckon with.

Referring to increase in supervision costs, due to better supervision being necessary where heavy fuel oil is to be used, most of the plants of even medium size pay from $200 to $300 per month for a plant chief engineer, but they are not always $200 to $300 per month men; the latter is reflected in the operating records of the plants having the highest labor and supervision costs in both the 1930 and the 1931 reports of the Cost Committee. For example, plant 82 spends $25,809.80 for labor and supervision and in addition spends $17,206.60 (most of which is for labor) for maintenance. The on-the-board cost per kwhr is 12.42 mills. On the other hand, plant 73 spends 16,099.09 for labor and supervision. Operating personnel does all of the repair work and spends but 1,825.13, all of which is for parts. The on-the-board cost is 3.52 mills. Plant 82 uses light fuel and plant 73 uses heavy fuel.

DEPOSITORIES FOR A.S.M.E. TRANSACTIONS IN THE UNITED STATES

Alabama

Auburn	Engineering Library, Alabama Polytechnic Institute
Birmingham	Public Library

Arkansas

Fayetteville	Engineering Library, Arkansas University

California

Berkeley	Library, University of California
Long Beach	Public Library
Los Angeles	Public Library
	University of Southern California
Oakland	Oakland City Library
	Teachers' Professional Library
Pasadena	Library, California Institute of Technology
Santa Clara	Library, University of Santa Clara
San Diego	Public Library
San Francisco	Public Library (Civic Center)
	Engineers Club of San Francisco
	Mechanics Institute
Stanford University	Library, Stanford University

Colorado

Boulder	Library, University of Colorado
Denver	Public Library
Fort Collins	Colorado State Agricultural College

Connecticut

Bridgeport	Public Library
Hartford	Public Library
New Haven	Public Library
	Library, Yale University
Waterbury	Silas Bronson Library

Delaware

Newark	University of Delaware
Wilmington	Wilmington Free Institute

District of Columbia

Washington	Scientific Library, United States Patent Office
	Library of Congress
	Bureau of Standards Library
	George Washington University
	Catholic University

Florida

Gainesville	University of Florida
Jacksonville	Free Public Library
Miami	Public Library
Tampa	Public Library

Georgia

Atlanta	Carnegie Public Library
	Georgia School of Technology
Savannah	Public Library

Idaho

Moscow	University of Idaho

Illinois

Chicago	John Crerar Library
	Western Society of Engineers
	Library, Armour Institute of Technology
	Public Library of Chicago
Moline	Public Library
Urbana	University of Illinois

Indiana

Evansville	Public Library
Fort Wayne	Public Library
Indianapolis	Public Library
	Indiana State Library
Notre Dame	Library, University of Notre Dame
Terre Haute	Rose Polytechnic Institute
West Lafayette	Library, Purdue University

Iowa

Ames	Iowa State College
Des Moines	Public Library
Iowa City	State University of Iowa

Kansas

Kansas City	Public Library, Huron Park
Lawrence	Library, University of Kansas
Manhattan	Kansas State Agricultural College
Wichita	Wichita City Library

Kentucky

Lexington	University of Kentucky
Louisville	Speed Scientific School
	University of Louisville

Louisiana

Baton Rouge	Louisiana State University
New Orleans	Howard Memorial Library
	Louisiana Engineering Society
	Public Library

Maine

Orono	University of Maine

Maryland

Annapolis	United States Naval Academy
Baltimore	Johns Hopkins University
	Engineers Club of Baltimore
	Public Library

Massachusetts

Boston	Northeastern University
	Boston Public Library
Cambridge	Harvard University (Engineering Library)
	Massachusetts Institute of Technology
Fall River	Public Library
Lowell	Lowell Textile Institute
Lynn	Free Public Library
New Bedford	Free Public Library
Springfield	Springfield City Library
Tufts College	Tufts College
Worcester	Worcester Polytechnic Institute
	Free Public Library

Michigan

Ann Arbor	University of Michigan
Detroit	Public Library
	Cass Technical High School
	Highland Park Public Library
	University of Detroit
East Lansing	Michigan State College
Flint	Public Library
Grand Rapids	Public Library
Houghton	Michigan College of Mining & Technology
Jackson	Public Library

Minnesota

Duluth	Public Library
Minneapolis	University of Minnesota
	Minneapolis Public Library (Engineering and Circulating Libraries)
St. Paul	James Jerome Hill Reference Library

Mississippi

Columbus	Mississippi Agri. & Mech. College

Missouri

Columbia	University of Missouri
Kansas City	Public Library
Rolla	Missouri School of Mines and Metallurgy
St. Louis	Engineers Club of St. Louis
	Public Library
	Washington University
	Mercantile Library

Montana

Bozeman	Montana State College

Nebraska

Lincoln	University of Nebraska
Omaha	Public Library

Nevada

Reno.....University of Nevada Library

New Hampshire

Durham.....University of New Hampshire

New Jersey

Bayonne.....Free Public Library
Camden.....Free Public Library
Elizabeth.....Free Public Library
Hoboken.....Stevens Institute of Technology
Jersey City.....Free Public Library
Newark.....Newark College of Engineering
Free Public Library
New Brunswick...Rutgers University
Paterson.....Free Public Library
Princeton.....Princeton University
Trenton.....Free Public Library

New York

Albany.....New York State Library
Brooklyn.....Polytechnic Institute
Pratt Institute
Brooklyn Public Library
Buffalo.....The Grosvenor Library
Engineering Society of Buffalo
Buffalo Public Library
Ithaca.....Cornell University
Jamaica, L. I.....Queens Borough Public Library
New York.....Engineering Societies Library
Public Library
College of the City of New York
Cooper Union
Columbia University
N. Y. University Library
Potsdam.....Clarkson College of Technology
Rochester.....Rochester Engineering Society
Schenectady.....Union College
Syracuse.....Syracuse University
Public Library
Troy.....Rensselaer Polytechnic Institute
Utica.....Public Library
Yonkers.....Public Library

North Carolina

Chapel Hill.....University of North Carolina (Engineering Library)
Raleigh.....North Carolina State College

North Dakota

Fargo.....North Dakota State College
Grand Forks.....University of North Dakota

Ohio

Ada.....Ohio Northern University
Akron.....Public Library
University of Akron
Canton.....Public Library
Cincinnati.....University of Cincinnati
Public Library
Engineers Club of Cincinnati
Cleveland.....Public Library
Case School of Applied Science
Cleveland Engineering Society
Columbus.....State of Ohio Library
Public Library
Ohio State University
Dayton.....Engineers Club of Dayton
Toledo.....Public Library
University of Toledo
Youngstown.....Public Library

Oklahoma

Norman.....Oklahoma University
Oklahoma City...Public Library
Stillwater.....Oklahoma Agricultural and Mechanical College
Tulsa.....Public Library

Oregon

Corvallis.....Oregon State Agricultural College
Portland.....Portland Library Association

Pennsylvania

Allentown.....Free Library
Bethlehem.....Lehigh University
Easton.....Public Library
Lafayette College
Erie.....Public Library
Lewisburg.....Bucknell University
Philadelphia.....Engineers Club
Drexel Institute
University of Pennsylvania
Franklin Institute
Pittsburgh.....University of Pittsburgh
Engineers' Society of Western Pennsylvania
Carnegie Institute of Technology
Carnegie Library (Schenley Park)
Carnegie Free Library of Allegheny
Reading.....Public Library
Scranton.....Public Library
State College.....Pennsylvania State College
Swarthmore.....Swarthmore College
Villanova.....Villanova College
Wilkes-Barre.....Public Library

Rhode Island

Kingston.....Rhode Island State College
Providence.....Brown University
Providence Engineering Society
Public Library

South Carolina

Clemson College..Library, Clemson College

Tennessee

Kingsport.....Public Library
Knoxville.....University of Tennessee
Memphis.....Goodwin Institute
Nashville.....Vanderbilt University

Texas

Austin.....University of Texas
College Station...Agricultural & Mechanical College of Texas
Dallas.....Public Library
El Paso.....Public Library
Fort Worth.....Carnegie Public Library
Houston.....Rice Institute
Public Library
Lubbock.....Texas Technological College (School of Engineering)
San Antonio.....Carnegie Library

Utah

Salt Lake City....University of Utah
Public Library

Vermont

Burlington.....University of Vermont

Virginia

Blacksburg.....Virginia Polytechnic Institute
Charlottesville....University of Virginia
Norfolk.....Carnegie Public Library
Richmond.....Virginia State Library

Washington

Pullman.....State College of Washington
Seattle.....Public Library
Engineers Club
University of Washington
Spokane.....Public Library
Tacoma.....Public Library

West Virginia

Morgantown.....West Virginia University

Wisconsin

Madison.....Library, University of Wisconsin
Milwaukee.....Public Library
Board of Industrial Education, Vocational School Library
Marquette University

Wyoming

Laramie.....Wyoming University

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Metallurgical Problems of the Diesel Engine

By R. J. ALLEN,[1] NEW YORK, N. Y.

The metallurgical problems of the Diesel engine consist more in selecting the most suitable material for a given application than in trying to find the solution for a chronic condition. Most of the problems encountered in operation are ones of design, but for economical reasons they often become metallurgical. It is believed that a more careful study of the details of design, with a view to eliminating localized overstress and providing better lubrication, will end most of the troubles. Where such consideration is given, the performance of the Diesel engine is most reliable, and many engines are building up records which rival those of the older reciprocating types of steam engines.



THE metallurgical problems of the Diesel engine are similar to those of other reciprocating types of prime movers, with the exception of the ones having to do with the combustion chamber. Consequently, the knowledge gained during the years in the manufacture and operation of steam and gas engines is largely applicable to the Diesel engine. This is also true of that more recently obtained in the development of the automotive engine.

Selection of Materials

Unfortunately, there is no one metal which will function satisfactorily in all the possible applications in an engine. There is, however, no dearth of types for any particular purpose. Some materials, with slight modification, may serve a number of purposes, such as cast iron when used in the frame, cylinders, heads, liners, and pistons. Others, such as the white metal in the main bearings, are largely single-purpose metals. The metallurgical problem is therefore a matter of selecting the most suitable material, properly conditioned, for each particular application. Naturally the question of cost must be considered.

As the result of intensive effort on the part of the material manufacturers to improve their products, the simple metals and alloys have been augmented by numerous complex alloys. This applies not only to the steels, but to aluminum, cast iron, and the bronzes. Some alloying elements have, over a period of time, proved their worth and are now recognized as essential, while others are still more or less on trial.

While all the numerous alloys offered may serve a useful purpose, it must be recognized that the difference, in many cases, is slight; hence, one is better off by developing a single type to the maximum, rather than half trying every new one brought out. The improvement in some particular characteristic due to a special alloy may be offset by some detrimental feature not shown up until the parts are in service.

The physical characteristics of the more standard alloys are now well established. Charts are available showing the corresponding tensile properties and hardness numerals for many forms of treatment, and the corresponding structural conditions have been recorded by the microscope. The endurance limits are known for normal conditions, at elevated temperatures, and where corrosion influences are present. Impact figures are available for both the single-blow Izod and Charpy and the repeated-blow Stanton types. Further, improvised tests may be made, approximating more nearly the actual working conditions than are obtained in any of the standards available, and additional data may be developed concerning the probable performance of the material when put into service. Hence, there is no reason for stressing materials beyond their safe working limits under any condition of service.

To obtain the most out of a material, it must be suitably processed in all stages of manufacture. In the case of a steel casting, for instance, the conditions of manufacture must be such that the material is free from blowholes, shrinks, and segregation. Then it must be heat treated to relieve strains and to develop a suitable homogeneous structure. Slighting any operation may ruin an otherwise excellent product. Materials should therefore be purchased on the basis of quality and from reputable sources.

When a type of material is selected for use, because of certain desirable characteristics, such facts should form the basis of a specification. Such a specification should be a concise statement of the characteristics and condition of the material which have been found most satisfactory. Its function, like that of a blueprint drawing, is to present to the shop or supplier a clear picture of exactly what is wanted. However, neither is an assurance of quality; the latter can be obtained only by adequate inspection.

Whenever possible, standard specifications of the national societies should be used. A standard material regularly made is likely to be better than a special material infrequently made. Furthermore, there is usually a range of selection in standard materials not available in special varieties. However, if the standard specification does not meet the particular needs, such alterations as seem desirable should be made.

Inspection and Testing of Materials

The inspection of materials is more difficult than that of machined parts. The latter is largely a matter of checking size, the correctness of which can be determined to any degree of accuracy desired. In the case of materials, however, many factors are involved, some of which are quite positive, while others are rather intangible. The evaluation of the latter is largely a matter of personal opinion, and hence the results may vary.

The checking of materials is further handicapped by the destructive nature of most of the tests. Tests such as the X-ray, magnetic, and hardness may be made on finished pieces intended for production, but others, such as the tensile, impact, and fatigue, those most commonly used, must be made on representative pieces, and the uniformity of the product be depended upon in determining the disposition of the lot.

Hardness tests are simple to make and positive in result. They give considerable useful information as to the uniformity

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of the product, and should be used frequently in the inspection of finished materials. The X-ray, although an excellent tool for development purposes and for the checking of special, highly stressed parts, cannot as yet be thought of as regular, routine test equipment. The magnetic test may be used for checking symmetrical sections, but it is not of great value as a general test. Iron filings are extremely helpful in detecting cracks in a hardened piece of steel, if the latter is suitably magnetized.

Acid etching or pickling may of course be used to facilitate the inspection for surface defects of individual pieces without affecting the utility of the product, and incidentally it can also be used to excellent advantage in checking cross-sections of the material for possible segregation.

The testing of large forgings and castings may be facilitated through the use of test prolongations forged or cast integral with each piece. These prolongations may be made sufficiently large for actual physical test purposes, or they may simply provide a means of obtaining a fractured surface for visual examination. With a little experience, one can tell rather accurately, from the appearance of a fracture, the physical condition of the material.

Through a conscientious application of the tests available one can determine quite accurately the suitability of any given lot of material for any given application of service.

General Types of Materials

Cast Iron. Cast iron has for years been the basic material in engine construction, largely because of its low cost and the ease with which it can be cast into intricate shapes. One does not have to look back very far to the time when it was simply a remelted pig iron, modified perhaps as to silicon content to obtain reasonable casting conditions. Eventually, the introduction of steel scrap, in small percentages, gave the cupola operator an opportunity to control the carbon to some extent and to produce a better graphite formation with corresponding increase in strength. From this beginning high-strength irons were soon developed in which the entire charge consisted of steel scrap. The term "semi-steel" has now little significance, as varying amounts of steel scrap are used in practically all cast iron made.

The strength of any given piece of cast iron is a function of its section, and this fact must be taken into consideration when making comparisons. When cast in a standard test-bar section, the high-strength irons will show tensile figures in excess of 50,000 lb per sq in. With the introduction of nickel, the strength may be increased to over 60,000 lb per sq in. and machineability be reasonably maintained. All cast irons, unfortunately, are inherently brittle, and this fact must be taken into consideration when evaluating the usefulness of the high strengths.

In plain cast irons the hardness is largely a function of the thickness of section and the silicon content. If the casting is quite irregular in shape, the heavy sections may be soft and the thin sections hard and unmachineable. When the silicon is increased to keep the light sections soft, the heavy sections are likely to be open-grained and spongy. Also, straight silicon irons oxidize readily at elevated temperatures and the combined carbon tends to break down into graphite and ferrite. Such a reaction involves an increase in size, or growth, and a decrease in strength. It is advisable, therefore, to keep the silicon as low as possible.

The addition of nickel, while supplementing the silicon in reducing chill, refines the iron and produces greater structural uniformity. It also reduces the tendency for draws and slightly increases the combined carbon, making the iron harder and more wear resistant. However, such irons are readily machineable, and a fine surface finish may be produced. Through the use of nickel the casting of many difficult sections is made readily possible.

Chromium, when added to cast iron, has the opposite effect to that of silicon and nickel, in that it tends to develop hardness through the retention of carbon in the combined forms. Such an alloy is most helpful in developing materials for use at elevated temperatures. It must be used in judicial amounts; otherwise, there will be an excess of hard combined carbon. This constituent is an excellent abrasive, and if present in a liner, for instance, may spall out in service and accelerate wear. Its presence also makes the machining exceedingly difficult.

Chromium and nickel, in suitable proportions, develop excellent characteristics in iron, particularly where resistance to wear and growth are important factors.

Molybdenum has a beneficial effect on cast iron. Its effect is intermediate between nickel and chromium, in that it tends to develop stability, without undue chilling tendencies, and a compact graphite formation which is quite beneficial in resisting fatigue. It may be used alone or in combination with nickel

Strains may be set up in irregular-shaped castings due to the differences in rate of cooling throughout the mass. If the castings are left in the sand, however, until fairly cold, the natural annealing effect of the slow cooling should effectively relieve the strains. If not relieved prior to machining, the piece may distort in service. A safe procedure, in cases where distortion may affect the operation of the unit, is to definitely anneal the castings in a suitable annealing furnace.

With facilities for annealing available, the additional step to heat treating is a comparatively simple matter. The desire for greater hardness in parts subjected to wear, such as cylinder liners, has resulted in the development of irons which are susceptible to heat treatment. Such irons ordinarily contain from 2 to 5 per cent of nickel, plus the proportionate amounts of chromium, and, like steel, may be heat treated to Brinell figures of 400 and upward. Although comparatively strong and tough, the materials are still cast iron, and as such are inherently brittle. The treatment is best confined to pieces of regular sections because of the possibility of cracks developing through the strains set up during quenching. The machining procedure follows that of treated alloy steels. The softer pieces may be machined after treatment, whereas the harder ones must be first roughed out, treated, and then finished off by grinding. Iron, heat treated for hardness, softens under heat, and hence will lose its hardness if operated at a temperature in excess of that used in tempering to develop the given hardness. However, the range is not seriously restricted, as tempering temperatures of around 900 F are usually employed during heat treatment.

The necessity for greater resistance to corrosion and stability at high temperatures has led to the development of a still more highly alloyed iron known as Ni-Resist. This iron contains some 15 per cent nickel and 7 per cent copper, with chromium varying from 1 to 6 per cent. The matrix is austenitic in type, somewhat similar to that of the 18 and 8 types of stainless steel. It is non-magnetic, tough, non-hardenable by heat treatment, and if the chromium content is not too high, reasonably machineable. Its coefficient of expansion is relatively high, approximating that of aluminum. Although comparatively soft, it will work harder in service, and develop appreciable resistance to wear. It can be produced in the foundry under standard conditions, although irregular sections may be difficult to cast because of the shrinkage.

Steel Castings. Steel castings, because of their combination of high strength and ductility, when suitably treated, and the fact that they can be produced in shapes not possible in forgings, serve many useful purposes. Like any other class of material,

they have their peculiar faults, which must be guarded against if trouble is to be avoided. Blowholes, shrinks, and segregation are the principal ailments, and unfortunately they are sometimes difficult to detect.

Cast steel lacks a constituent like graphite to counteract the effect of the metal shrinkage, and unless uniform sections are provided or provisions made to feed the heavier sections while cooling, shrinkage cracks may result. As a result, complicated castings with irregular sections are much more difficult to obtain in steel than in iron.

The beneficial effect of heat treatment on a carbon-steel casting is illustrated by the following table:

Condition	Tensile	Yield	Elongation	Reduction	Brinell	Impact
As cast.....	74,800	37,500	19.5	29.0	156	17
Normalized.....	75,650	42,000	25.5	44.0	143	21
Treated.....	84,200	57,400	31.5	65.0	160	44

Alloy-steel castings are now being produced in almost as many varieties as obtainable in bar stock, and if suitably heat treated, serve a useful purpose. The greater strength permits lighter weight, and the lighter sections tend to promote soundness. Stainless-steel castings of many grades are also being produced, although their use in Diesel engines, because of the cost, is rather limited. These steels are difficult to cast, and because of their high solidification shrinkage, usually require special pattern equipment and simpler designs.

Structural Steels. Aside from the straight carbon steels, the number of alloy steels available for structural purposes are legion. Many have been standardized by the S.A.E., and are designated by their numbers, while others, the special product of some steel company, are sold under brand names. While many of these steels have distinctive characteristics and serve useful purposes, nevertheless the number could be materially reduced with profit to the industry. As a matter of fact, most of the troubles encountered, where the steel is supposed to be at fault, are due to improper heat treatment rather than any inherent fault in the type. The fact that better results can usually be obtained in the long run through the suitable conditioning of a standard type of material, rather than through the hit-and-miss tries of special types, cannot be too strongly emphasized. All steels, particularly the alloy types, must be suitably heat treated to develop their maximum physical characteristics.

Special Steels. Numerous types of special alloy steels are also available. Such steels are of value for their ability to perform some special purpose well, such as air hardening without distortion, resisting wear, or standing up under corrosion. Chromium is the principal element in most of these steels, although nickel, molybdenum, vanadium, copper, silicon, and aluminum may play important parts.

Stainless iron, a low-carbon 12 per cent chromium material, has many useful applications. If properly treated, it has excellent physical characteristics and reasonably good resistance to corrosion. With properties quite comparable to those of S.A.E. 3140 steel, it may be safely used where strength is required. For greater stability, under severe conditions, one is forced into the austenitic steels carrying appreciable percentages of chromium and nickel, with, in many cases, an additional element such as silicon. Many modifications, of course, are available.

Nitralloy, a special type of steel which may be processed to develop remarkable surface hardness, stands up well under severe conditions of wear. Two grades, the chromium-aluminum, having exceedingly high hardness, and the chrome-vanadium, with lesser hardness but greater toughness, are in general use. The material retains its hardness through the temperature range encountered in the Diesel engine, with the possible exception of the exhaust valve. It is quite resistant to both atmospheric and alkaline corrosion, but not to acids. However, from records available, it would appear that neither type is seriously attacked by the usual fuels.

Forgings. Forging assures a sound section in the finished piece, free from blowholes or draws. Further, the quality of the stock used may be carefully checked before the forgings are made. However, drop forgings have their peculiar faults in the form of laps and cold shut. Fortunately, these, unlike blowholes and shrinks in castings, can usually be detected quite readily by visual inspection. Drop forgings have an advantage over castings in that they may be made in light sections and still retain a clean-cut surface appearance.

Unless care is exercised in forging, the structural condition of the material, despite the beneficial effect of the added working, may be utterly ruined. If steel, for instance, is soaked at a high temperature in an oxidizing atmosphere, the gases penetrate the section and oxidize the grain boundaries. As a result, the material becomes quite brittle, and the condition is not corrected by any normal heat treatment. Some indication of this condition, if present, can be detected by examining the pickled surface, but the only safe method is by an actual fracture test.

As the material is elongated under the hammer it tends to accentuate the directional grain flow normally found in bar stock. The ductility, when tested across the direction of flow, is usually much lower than when tested with the grain. In some pieces, such as a crankshaft, the directional grain is an asset, providing the fibers are continuous throughout the length, while in other parts such as gears, where the direction of stress application is continually changing, it is not. The directional tendency may be broken up by suitable cross-forging, and the desirability of so doing must be considered when designing stressed members.

Forgings, like castings, must be properly heat treated to secure uniformity of structure, relieve strains, and obtain the maximum combination of strength and ductility. Light sections should be fully heat treated, while heavy sections may be normalized and annealed.

Aluminum Alloys. Aluminum in its pure condition, like iron, is a soft, tough metal, but when suitably alloyed with copper, magnesium, and other elements, develops many desirable physical characteristics. All of the alloys may be cast into useful form, while some, in addition, may be forged, rolled, or extruded. Many of the alloys are naturally soft and tough, others are comparatively strong but brittle, while some, when heat treated, develop a good combination of strength and ductility. In the wrought form, treated, tensile strengths of over 50,000 lb per sq in. with elongations of around 20 per cent may be obtained. The modulus of elasticity, however, is only one-third that of steel.

The aluminum alloys are unique in that their physical characteristic tends to run to extremes, and, unfortunately, some of them conflict in their usefulness. Light weight, the outstanding characteristic, is offset to a certain extent by the low modulus. The high heat conductivity in many cases is an advantage, while in others it is not. The high coefficient of expansion, common to most of the alloys, does not work out well with cast iron. The bearing characteristics are not good, and the resistance to alkaline corrosion is low.

While the aluminum alloys are regularly used for many purposes, particularly where weight is a consideration, the selection of the most suitable type must be given careful consideration.

Bronzes. The various types of bronze alloys available under various standard specifications are exceedingly numerous, The range of characteristics, which is dependent on the elements alloyed with the copper, is probably greater than for any other class of material.

Some of the alloys are soft and tough, others are hard and

brittle, while still others have a good combination of strength and toughness. A few, such as the aluminum bronzes, respond to heat treatment.

Many of the bronzes are excellent bearing metals, particularly when tin and lead are present. In general, all the bronzes are resistant to atmospheric corrosion and, with suitable modification, can be made to resist the attack of any of the conditions encountered in Diesel-engine work.

The majority of the bronzes are used in the cast form, although many may be die cast and forged. The ease with which they can be fabricated and their resistance to corrosion tend to make them attractive as structural members, but these advantages are in most cases offset by the low modulus of elasticity, weight, and comparatively high cost. In bearing service, however, their use is well established.

The bronzes, as a class, oxidize readily when molten, and unless care is used in the foundry to prevent such a condition developing, their physical characteristics may be seriously impaired. Such a condition will show up in a tensile test, largely through low ductility, although the condition can readily be detected by means of a simple fracture test. Oxidation in a tin-bearing bronze is particularly bad, as any of the extremely hard tin oxide formed, and scuffed out in running, may seriously score the shaft. The temperature of casting also has an important effect on the strength and ductility of many of the bronzes.

Monel metal, because of its combination of strength, when suitably processed, higher modulus of elasticity, wear resistance, and ability to withstand most forms of corrosion, is an excellent intermediary material between the bronzes and steel. It is readily available in wrought form, but sound castings, if irregular in shape, are not so easily obtained.

Bearing Metals. An ideal bearing metal consists of two constituents. There should be a tough base metal which acts as the body or matrix to carry a hard, finely dispersed element or alloy, which, in the aggregate, acts as the bearing surface. The hard constituent resists wear; the soft constituent provides the degree of conformability necessary to relieve any local high spots, and through its natural tendency to recess, assists in maintaining the oil film. Cast structures, because of the natural arrangement of the grains, are usually better than those which have been mechanically worked. Bearing materials should have a low coefficient of friction, although under running conditions the actual friction is dependent on the pressure, speed, temperature, and lubrication.

The white metals, which are essentially alloys of tin, lead, copper, and antomony, are the commonly recognized bearing metals. The tin-base or babbitt metals have a better combination of strength and plasticity than the lead-base alloys, and hence are more generally used in Diesel work, even though more expensive. However, considerable work is being done with the lead-base alloys, and some worth-while results are being obtained.

The white metals are sensitive to variations in composition, and small changes in the hardening elements or some impurity may either render them brittle or lower their melting point appreciably. The metals oxidize readily when molten, and hence extreme care must be exercised in both the melting and pouring operations.

The high-lead bronzes, with or without small percentages of tin, are closely associated in application with the white metals. Such alloys do not conform to the true concept of a bearing metal, but are actually the reverse, in that the copper matrix is harder than the dispersed load. However, they seem to function quite well within their working range, which is the fusion point of the lead. A rather high coefficient of expansion and possible lead segregation are the two features most likely to give trouble in the use of this material.

The harder cast bronzes, carrying an appreciable amount of tin, lead, and possibly some phosphorus, are good bearing metals when operating against hardened steel. Such bronzes have good strength and resistance to wear, but good alignment and adequate lubrication are essential to their satisfactory performance. If the oil film fails, the resulting seizure is likely to be disastrous to the shaft. Numerous types of bearing bronzes are available, and the selection is largely a matter of past experience. The structural condition of the material used should be checked by fracture tests.

The beryllium-copper alloys, capable of being treated to a Brinell hardness of around 400, are being advocated as bearing metals. Their use, however, has not spread sufficiently to warrant any satisfactory conclusions being drawn as to their probable performances.

Chromium Plating. "Hard" chromium plating is being used quite extensively because of its excellent wear-resisting characteristics. The type must not be confused with "decorative" plating, wherein an extremely thin film of chromium is applied, more or less as a lacquer, to preserve the copper-nickel plating underneath.

The hard plate is applied directly to the base material in thicknesses varying from one-half to three-thousandths of an inch, depending on the conditions of service. Either steel or bronze, as a base material, plates quite satisfactorily; cast iron, because of the graphite, does not provide such a suitable base.

When properly finished by grinding and polishing, the plated chromium presents a hard, low-friction, wear-resisting surface quite suitable for bearing purposes. The plate, although sufficiently tough and adherent for normal bearing conditions, is not suitable for use under shock or extreme pressures such as might be applied through high spots in a hard packing material. Chromium works best as a bearing when mated with a soft packing such as a leaded bronze; cast iron, particularly if it contains free cementite, is likely to score the plate through an indentation of the softer base material.

The hardness of plated chromium is dependent on the presence of the hydrogen deposited during plating, and if heated sufficiently high, in the neighborhood of 1400 F, the plate becomes soft.

Unless the conditions of plating are carefully controlled, strains are set up in the deposited metal sufficient at times to cause it to crack. Such cracks may, in the extreme, extend through the entire depth of plate, in which event the tendency to spall is increased and the resistance to corrosion decreased. However, because of a better knowledge of the variables responsible for this tendency, the plate being produced at the present time is comparatively free from these defects and should perform well in service.

Chromium works well on shafts, pins, and similar parts where good bearings and adequate lubrication are provided. It also functions well as a protection against galling on liners or plugs which are assembled with a close fit and which may later have to be removed. Worn surfaces may in many instances be built up with chromium and satisfactorily salvaged. Further, chromium-plated surfaces, when worn, may be renewed at small expense.

Chromium plating, like all other materials, has its field of useful application and, when rightly applied, gives excellent results.

Welding. The technique of welding has reached the point where it has ceased to be a metallurgical problem in so far as steel construction is concerned. With suitably coated rods of the same composition as the base material, a skilled operator can lay welds having substantially the same physical characteristics as the original plates. A low-temperature strain-

relieving treatment, however should follow the welding of all complicated sections that are likely to be subjected to vibratory stresses. In the case of the higher carbon and alloy steels, the temperature should be carried above the critical transformation point of the material, so that, in addition to relieving the strains, there may be a refinement of the coarse grain developed during the welding operation.

Cast iron can be welded under favorable conditions, but on the whole the results are not overly satisfactory. The structure of the iron matrix is badly upset, and incipient cracks may develop in and about the weld unless the operation is most carefully controlled.

Design, in built-up structures, is all important and far outshadows the metallurgical considerations. The liberal fillets in castings, to avoid shrinkage draws which tend to develop at sharp corners or sudden changes in section, and a reasonable thickness of metal to prevent misruns automatically serve to stiffen the structure and smooth out any tendency for concentration of stress. As a result, castings are usually quite rigid because of foundry considerations and so many of the engineering problems disappear. In built-up structures, however, the reverse is true. The material is strong and tough, and as far as static stresses are concerned, can be used in light sections; the fillets naturally are small. Unless proper precautions are taken, therefore, to stiffen such a structure by suitable bracing, flexing in service is likely to occur, and with it concentration of stresses in the welds. Such a condition usually results in ultimate failure, not because of a faulty weld, but because of insufficient design.

Where weight is an important consideration, welding undoubtedly provides a means of keeping it down.

While the welding of structural members represents sound practice, the building up of worn or undersized working parts is open to question. Many failures have been traced to this cause. Active parts are usually made of medium-carbon or alloy steels which do not weld as satisfactorily as the low-carbon structural steels. The higher strength materials are more likely to be spongy when welded, and a suitable structural condition after welding can be developed only by heat treatment. Such a heat treatment, accompanied by surface scaling and distortion, is of course ruinous to a finished piece. Where the stresses are inherently low and the welding is not being performed at a change in section, the procedure may not be harmful if done under competent supervision, but otherwise it should be looked upon with disfavor.

Failures

Failures may be due to either faulty material or faulty design. A material may be considered at fault if some irregularity, not common to a properly conditioned piece, has contributed to the failure. The irregularity may be a cold shut, segregation, poor structure, deep tool marks, or similar defect. The design or selection of material may be considered at fault when a sound homogeneous piece, properly conditioned, fails for no apparent reason. As a rule, occasional failures denote faulty material, whereas repeated failures tend to throw suspicion on the design.

While it is not practical to positively establish the soundness of each piece of material, nevertheless, with close supervision over the methods of manufacture and adequate inspection in the shop, the chances of faulty material getting into service are not great.

Changes in material can be made without any outward evidence of it having been done. Changes in design, however, usually involve not only the piece actually in trouble, but other mating parts as well, to say nothing of tools, fixtures, and the interchangeability of parts. Consequently. where a trouble may be corrected through the substitution of another type of material, it is logical that such a change should be made. The result, however, is that many problems that are fundamentally ones of design automatically become ones of metallurgy. Although a piece of alloy steel may be substituted for a carbon steel or a piece of Nitralloy for hardened tool steel, the extent to which material changes may be made is comparatively limited, and hence considerable thought must be given to the original design and selection if trouble is to be avoided.

Operating problems in which the material is involved may be grouped under three general headings: first, breakage; second, excessive wear; and, third, corrosion and oxidation.

Breakage. The first class, breakage, assuming the material is correct, is of course the result of overstress. Pieces made of excellent material and operating under low calculated values of stress occasionally fail. The cause, although not always apparent, can usually be traced to a concentration of stresses. In irregular-shaped pieces the average stress is of little practical importance, as failures are the result of maximum stresses. Each type and condition of material has a certain maximum stress value, called the endurance or fatigue limit, under which the piece will withstand an indefinite number of applications of stress without failure. If this stress is exceeded, however, even though over a minute area, a crack will ultimately form and failure occur.

Stress concentrations may be due to sudden changes in section, deep tool marks, inclusions, or anything which tends to break the smooth continuity of the section, or they may be the result of misalignment and excessive vibrations.

While the intensity of the stress concentrations due to changes in section may be difficult to calculate, their probable location and approximate magnitude can often be obtained through the use of bakelite models and a polariscope. Under the action of polarized light, the stress distribution in a section of bakelite may be observed by color variations in the material. Such a method of investigation is simple and instructive.

As stress concentrations are the most prolific cause of breakage, effort should be employed to keep them to a minimum. Fillets at shoulders in stress sections are recognized as essential, but slow tapers, if practical, are better. Similarly, splines distributed around the surface of a shaft are better than a single key. Whitworth threads are better than vee threads because, in the case of bending, the stress concentration at the roots is much lower with the smooth radius than with the sharp notch. For the same reason, studs, with the body turned down to the root diameter of the thread, are much better than those with full-sized bodies; the reduced diameter of the body, aside from reducing the stress at the roots of the thread, provides greater flexibility along the entire length and less tendency for concentration at any point.

There is a tendency in design to lay too much emphasis on the elastic limit and tensile strength and not enough on the so-called toughness of the metal. The fatigue or endurance limit, obtained with a smooth, symmetrical specimen running without impact, seems to be a function of the ultimate strength. In actual service, however, such conditions rarely exist, and aside from the irregularities of operation, parts are seldom regular in section. Stress concentrations, not considered in the fatigue limit, are therefore an important consideration in actual practice. If the material is brittle, a crack once started will progress very rapidly, whereas if the material is tough, the condition of overstress may be relieved through a local readjustment of the grains in the area of stress. Even though the formation of a crack cannot be avoided, the tougher material may seriously retard its progress.

Unfortunately, the toughness which tends to retard the

formation and subsequent development of a fatigue crack is not indicated by any available mechanical test. Pieces having the same tensile and impact values may show decidedly different performances in service. The microscope seems to be the most useful method of checking this characteristic. In steels, for instance, a coarse-grained, pearlitic structure, under many conditions, is not as resistant to fatigue as a normalized structure consisting of a fine-grained intimate mixture of ferrite and pearlite. The soft, tough ferrite seems to provide the mobility which is so important in resisting the spread of cracks. High-tensile alloy steels, not suitably treated, may fail more quickly than a comparatively soft, low-strength material.

Good balance must be maintained between strength and toughness, and when specifying some of the more highly stressed parts, the microstructure should be included.

Excessive Wear. The second class of failure, excessive wear, may be due to faulty material, although high pressures, due to abnormal conditions of operation, and inadequate lubrication are more likely to be the real causes.

Wear is the result of a rubbing, metal to metal, contact between two surfaces. If the surfaces are continually separated by a thin film of oil, wear is largely eliminated, and is reduced to a mere matter of erosion, and possible chemical attack, by the oil used. If the film is periodically broken, however, wear may be quite rapid.

The normal working pressures are readily calculated and may be amply provided for in the original design. Abnormal stresses due to periodic vibrations or the whipping of shafts are not so easily determined, and the resulting pressures may be sufficiently high to break the oil film. Misalignment, surface finish, and dirt are potential sources of wear, although all three may be readily controlled, and hence should not be serious factors in actual operation.

Numerous types of metals suitable for operating in contact with one another are available, and if worked within their corresponding limitations of pressure and temperature, under adequate conditions of lubrication, a satisfactory performance is assured. The selection of the most suitable combination of metals should be given careful consideration, as in addition to obtaining long life as a whole, the conditions should be such that the least expensive of the two pieces in contact will wear most. A hard bronze bushing may readily score and gall a soft steel shaft, whereas a soft bushing will not. A soft material in the presence of dirt, however, may load up and, as a result, lap away a hard material quite rapidly.

In order to maintain a good oil film, assuming proper alignment, finish, and rigidity, it is necessary that the elements of temperature, pressure, feed, clearance, and oil viscosity be held within limits. The viscosity of an oil decreases with an increase in temperature, and if thinned down too much, the resulting film may not be sufficient to carry the load. The pressure, taking into consideration the surface velocity, should be within the limits which experience has proved most satisfactory for the metal combination employed. The clearances must be adjusted to suit the viscosity of oil being used; if the clearance is too small, the film will not be established, and if too large, the oil may work out and not maintain a continuous film. The arrangement of the feed must be such that the distribution, developed in the unloaded area, is uniformly directed to the zone of pressure.

Perfect lubrication is never attained in actual operation; hence, metal-to-metal contact does occur, even though only at the time of starting, and therefore wear is inevitable. Through careful utilization of available materials under proper conditions of operation, it may be maintained at a very small rate, however.

Corrosion and Oxidation. The third class, corrosion and oxidation, is more likely to be a metallurgical problem than the first two, as, despite the best the designer can do, the materials must contact with the corroding or oxidizing medium.

The exhaust valve is probably the hot spot in all internal-combustion engines and may reach a rather high temperature. The other parts, however, which can be more readily cooled, do not become excessively hot.

The Diesel engine, however, unlike the gas or gasoline engines, draws in air undiluted with fuel and compresses it to a temperature at which the oxygen becomes active before the fuel is admitted. The oxygen, therefore, in the absence of fuel, is free to attack the iron or other materials forming the surface of the combustion chamber. Fuel oils for Diesel engines are usually not as highly refined as gasoline, for instance, and may contain appreciable amounts of sulphur which become quite active chemically.

Although the injected fuel is carefully atomized, there is in many cases a tendency for flame impingement. In the case of iron pistons, if the heat is localized, particularly at the center, growth takes place due to the combination of temperature variation, and a crack may develop. The upper portions of the liners, also, wear more rapidly than the lower portion, largely because of this same condition.

Material Application

Frames. Important considerations in the design of a frame are strength, rigidity, and weight. Frames may be made of cast iron, aluminum, or welded steel. The ease with which cast iron can be fabricated and its comparatively low cost account for the majority of the engine frames being made in that material. Welded steel frames, however, are being given consideration because of their possible lighter weight.

If made of cast iron, the composition may vary all the way from a straight silicon to a high-test alloy, depending upon the particular requirements of service. Care must be exercised to keep the sections reasonably uniform, if casting shrinks and strains are to be avoided.

Welded frames are usually made of soft steel and are finally annealed. If made of high-carbon or alloy steel, the entire frame should be heat treated to eliminate the brittle area which develops near the welds and to fully develop the advantage of the higher strength materials. Rigidity must be considered if weld failures are to be avoided. The frames of small units, where the minimum weight is desired, may be made of aluminum.

Cylinders. In smaller units the cylinders may be integral with the frames, whereas in the large units they are made separate. When the cylinder is to provide its own liner, considerable attention must be paid to the type of material used if minimum wall wear is to be obtained; otherwise, the casting may be made of any type of material that will develop the required strength. Care should be taken to see that a smooth water-jacket surface is provided with as little attraction for scale as possible. The casting should be given a low-temperature anneal to relieve strains and prevent subsequent distortion.

Liners. Cylinder wear is an important consideration in the operation of a Diesel engine. The rate of wear is dependent upon one or more of several factors such as lubrication, ring pressure from the gas reaction, dirt, oxidation, and type of material.

Lubrication and ring performance are matters of design, while dirt, if a factor, may be controlled to a large extent by the use of a suitable air cleaner. Oxidation, while influenced by the fuel, is more dependent on the temperature attained, and hence varies with the conditions of cooling. Thin liners, for this reason, are better than thick ones.

Liners may be either of the dry or wet type. The dry type is mostly confined to the small sizes because of the thicker section

of the two walls and the difficulty of providing the close fit essential for good heat conductivity. The metal conditions are much more difficult to control in the dry type than in the wet type because of the larger casting involved.

To resist wear, the liner material should have a reasonably hard, dense structure, with good bearing characteristics and stability under the conditions of operation. To meet these requirements the majority of liners are made of cast iron, although some are being made of steel and hardened Nitralloy. In cast iron, this necessitates low silicon to resist growth, low total carbon to develop denseness, and eutectoid ratio in combined carbon and fine grain size to develop resistance to wear. Nickel and, if the size will permit, chromium are helpful in obtaining the desired conditions.

Because of their symmetrical section, liners may be cast centrifugally and a denseness and uniformity of structure obtained not easily reproduced in a sand casting. If cast in the ordinary manner, the mold should be vertical and the casting should be machined, so that the bottom, or drag end as cast, will be uppermost in the cylinder. In this way the denser metal will be placed at the point of maximum wear and excellent results obtained.

Under usual conditions the Brinell hardness of a straight silicon iron cannot much exceed 180 and that of the regular chrome-nickel iron 220, without free carbides being present. Such carbides are objectionable because of their abrasive characteristics when scuffed free of the iron. If a higher hardness is desired, it should be obtained through the use of a heat-treated iron. Nickel, plus some chromium, in a cast iron, when suitably treated, will permit the development of an excellent structure with Brinell hardness figures of 400 or more.

A 60-point carbon steel, normalized or heat treated for greater hardness, has, when adequately lubricated, proved quite satisfactory as a liner material. Thin sections may be employed and excellent cooling obtained.

Ni-Resist cast iron offers excellent possibilities as a liner material in combination with aluminum, as the coefficient of expansion is about the same in both materials. It is quite resistant to attack from either heat or the normal gases, and its soft, austenitic matrix tends to take on a work-hardened glaze which makes it quite resistant to wear.

Nitralloy steel, nitrided, is being used on many of the smaller units with apparently good results, and, aside from possible attack where high sulphur fuels are used, there is no reason why its use should not spread.

Heads. Cylinder heads are usually made of cast iron, although steel may be used on the larger units and aluminum on the smaller ones. The head must be sufficiently strong to resist the combustion pressures and have the necessary stability to resist the attack of the air and gases of operation.

Head sections on the single-acting engines are usually rather complicated because of the manifold passages and water-cooling chambers. As a result, the heat stresses may be rather high, and this fact must be taken into consideration in drawing up the design and selecting the material. If made in cast iron, the head may provide its own valve seat; if made in aluminum, some other material must be inserted. Aluminum bronze and Ni-Resist are frequently used for this latter purpose.

The water jacket should be smooth and clean to resist the adherence of scale. Attention must be paid to the cooling water used with aluminum; otherwise, porosity, the result of corrosion, may develop.

Pistons. The conditions imposed upon the piston, like those of the liner, are quite severe. A piston should have light weight, strength, good bearing characteristics, an expansion rate comparable to the liner, good heat conductivity, and resistance to the attack of the hot air and gases. There is no single material which will meet all these requirements; hence the one selected must be a compromise.

Cast iron is largely used, however, because its coefficient of expansion may be the same as the liner, its bearing characteristics are good, and it is comparatively inexpensive. The pistons, like the liners, should be made of a dense, fine-grained material. The majority are sand cast, although some of the smaller sizes may be made in semi-permanent molds.

The straight silicon irons oxidize readily in contact with hot oxidizing gases, and if there is a tendency for a local hot spot in the center of the piston, it may fail rapidly through growth and fatigue. The use of a nickel or nickel-chromium iron will delay this type of failure.

In the event of seizure with a cast-iron piston, the resulting damage to the liner may be rather disastrous.

Aluminum, because of its light weight, is being used extensively for pistons in the smaller high-speed units. It may replace iron on larger sizes where water cooling would otherwise be necessary. Its rather high coefficient of expansion can be controlled in the smaller units through the use of inserted Invar rings, and the tendency to slap, when cold, can be eliminated. In some cases, the expansion is taken care of through the use of a split skirt.

Aluminum has the disadvantage of being a comparatively poor bearing material, and hence requires good lubrication in a place where it is not easily maintained. Also, being a soft material, it loads up with dirt and acts as a lap on the liner. Seizure is not as severe, however, as with the iron piston.

Pistons of the aluminum type are usually made of a copper-magnesium-nickel alloy, heat treated to develop a Brinell hardness of around 120. Because of the low modulus, there is a tendency for distortion under load, which must be taken care of with additional clearance in the vicinity of the piston pin.

Steel is not generally used as a piston material because of the difficulty of obtaining satisfactory castings and its tendency to gall under failure of lubrication. Piston heads may be made in steel and used with iron skirts, however.

Piston Rods. One of the most likely sources of trouble in a double-acting Diesel engine is the piston rods. The diameter of the rod must be kept small, in order to interfere as little as possible with the capacity of the lower combustion chamber. This means that the working stresses must be high. The rods also must act as carriers for the piston-cooling medium. If the rod is hollow bored to permit a circulation of the cooling liquid, the temperature gradient stresses may be quite appreciable. If the cooling liquid is carried by means of an external sleeve, the actual rod diameter must be kept smaller than would otherwise be necessary, and the resulting stresses are again high. If water is the cooling medium, the matter of corrosion must be given consideration. Corrosion not only lowers the endurance limit of the steel, but it causes pits which localize the stresses which further accelerate failure.

Because of the high working stresses and packing wear, the natural tendency is to make the rods of alloy steel, suitably heat treated to develop strength and hardness. However, such rods do not seem to stand up as well as those made of the softer steels, because they apparently lack that inherent toughness which is so essential to resisting the progressive type of failure. Either straight carbon or some soft alloy of the nickel-manganese type seems to work out best.

Piston Rings. Piston rings are usually made of a straight silicon cast iron regardless of the type of liner used. Excellent results have been obtained through the use of nickel iron, and its use for this purpose should spread. Steel, of approximately spring temper, may be used in the smaller sizes. Experiments

have been conducted with other types of iron and hardened steels, but the results have not been sufficiently outstanding to cause any general change.

Piston Pins and Bushings. The chief requirements of a piston pin are that it be light, have sufficient strength to carry the load, and the proper hardness to resist wear. With adequate lubrication and a suitable fit, a piston pin should resist wear indefinitely. Pins may be made of hardened tool steel, although they are more usually made of carburized and hardened steel. Nitrided and chromium-plated pins have also been used with considerable success. A smooth, highly finished surface is essential to long life.

Bearing bronzes of the composition type are most generally used for bushings, although the copper-lead type seems to be meeting with favor because of the smaller damage to the pin in the event of seizure. Either, under normal conditions, works satisfactorily.

Crankshafts. One of the desirable characteristics in a crankshaft is that the directional grain flow of the metal follow the contour of the section and be continuous throughout its length. Such a condition eliminates the transverse stressing of the material. Single-piece forgings are preferable to built-up sections because of the better stress distribution at the changes in section. Failures, when they do occur, however, are usually the result of critical vibrations, and the stresses developed in this manner are generally more than any metal can be expected to carry.

In order to obtain the necessary rigidity in a crankshaft, the section is large, and hence the working stresses are low; consequently, the question of strength is not such a serious matter. Bearing wear is one of the important considerations in selecting the type of material to use.

Most crankshafts are made of a straight carbon steel, although on some of the higher speed units a harder material may be desirable. In such cases the bearings may be hardened by case carburizing or by nitriding. The latter, although more expensive, is perhaps the preferable method.

Connecting Rods. Connecting rods present few, if any, metallurgical problems. The majority are made of carbon steel, suitably treated, although on some of the smaller high-speed units alloy steels may be required.

Aluminum may be satisfactorily used, but as a rule the expense is not warranted. With a careful distribution of the metal in a steel rod, the section can be made quite light and suitable results be obtained at much less cost.

Bearings. In the small high-speed units the bearing conditions in the Diesel engine may be quite severe, but on the whole they are quite normal.

As a rule, the crankshaft bearings, both main and connecting rod, are made of a thin layer of white metal backed up with a strong steel shell. The tin-base alloys, because of their greater strengths, are preferred. In the fabrication of the bearings care must be taken to see that there is a good bond between the white metal and the base and that the metal is not overheated in pouring. A poor bond usually results in cracking and higher temperature of operation. The serious disadvantage to the use of the white metals is their low-temperature operating range.

When the operating temperatures are higher than the white metals can safely stand, the leaded bronzes may be used. Their operating limit, reached through a fusion of the load, is in the neighborhood of 600 F. Because of a high coefficient of expansion, these materials must either be used as a thin lining on a steel back or else fitted with appreciable clearance. Seizure, if it occurs, is quite positive, but usually not destructive.

The leaded bronzes are frequently used on small plain bearings and in guides where the conditions of lubrication are not good. In the presence of dirt, however, they tend to load and act as a lap:

Camshafts and other accessory bearings are taken care of by any one of a number of the standard copper-tin-lead mixtures regularly available. Such bearings give excellent results when the hardness is suitably adjusted to the mating steel. Good alignment, smooth finish, and adequate lubrication are essential requirements. In the event of seizure, the shaft may be badly scored.

Valves. The exhaust valves, being perhaps the hottest spot in the engine, naturally must be made of a material that will withstand the temperatures involved. In the larger engines, cast iron does very well, particularly if alloyed with chromium and nickel. In the small units, where the conditions are more severe, better materials, such as the chrome-nickel-silicon steels, may be necessary. Ni-Resist, stainless steel, and the straight chrome-nickel steels are also used. To reduce costs, a heat-resisting alloy head may be welded to a carbon-steel or alloy stem.

The heat-resisting materials are rather poor conductors of heat, and the heat dissipation has in some cases been improved by inserting a cone of salt or copper within the head and stem. Under severe conditions, such as are found in the smaller high-speed units, such types may serve a useful purpose.

The inlet valves, being continually cooled by the incoming air, do not require particular attention. Good results are obtained with cast iron on the larger units and with the straight alloy steels, such as a medium carbon-chrome-nickel, on the smaller units.

The clearance between the valve stem and valve guide must necessarily be small, and the conditions for lubrication are not good. As a result, the valve stems must be made as hard as practical. Attempts have been made to do this through the use of chromium plating, but the results have not been overly successful. The guides are, as a rule, made of a medium-hard cast iron or a bronze.

Valve seats are usually cut directly in the cylinder-head material, and, generally speaking, the results are quite satisfactory. Inserted seats of some harder or heat-resisting material may be used, but mechanical difficulties in keeping them tight make their use undesirable.

Fuel System. Because of the high pressures involved, and at times the corrosive nature of the oils handled, the spray nozzles and pump parts must be made of a hard, corrosion-resisting material. The stainless steels, suitably heat treated, seem to meet the requirements quite well.

Conclusions

The metallurgical problems of the Diesel engine are not serious and consist more in selecting the most suitable material for a given application than in trying to find the solution for a chronic condition. Most of the problems encountered in operation are fundamentally ones of design, but for economical reasons they often become metallurgical. There is no complaint on this score, but it is believed that a more careful study of the details of design, with a view to eliminating localized overstress and providing better lubrication, will eliminate most of the troubles now encountered. As a matter of fact, where such considerations are given, the performance of the Diesel engine is most reliable, and many engines are building up records which rival those of the older reciprocating types of steam engines.

Discussion

J. S. Vanick.[2] The author has presented a splendid summary of the actual and potential applications of various metals to Diesel construction, and such comment as can be offered will be

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more in the nature of elaboration upon items of special interest which he has mentioned. The critical point of view with which we examine Diesel materials of construction is concentrated toward bringing out their best performance, without changing the basic character of the material we work with. If cast iron is widely applied, it is possible to accentuate certain desirable qualities of the material by means of alloys without changing the method of construction or design.

The discussion bases a comparison of fabricated welded-steel structures with cast iron possessing a strength of 24,000 lb per sq in. and modulus of 12,000,000, but the writer happens to know that the author's firm specifies for engine parts a 40,000 or 45,000 lb per sq in. tensile cast iron which is alloyed and can produce a 50,000 or 60,000 lb per sq in. cast iron. Such cast irons possess better than double the strength and modulus of 24,000 lb per sq in. tensile and 12,000,000 modulus and are commercially producible in progressive foundries. Tentative specifications for cast irons to possess 40,000, 50,000, and 60,000 lb per sq in. tensile are being prepared for presentation before the committee on cast iron of the American Society for Testing Materials. Such high strengths can be obtained by using 60 to 90 per cent of steel in the cupola charge along with 1 or 2 per cent of silicon, manganese, and nickel. Such an iron will analyze about 2.80 to 3.20 C, 0.80 to 1.75 Si, 0.80 to 1.10 Mn, and 1.25 to 2 Ni. Its application has been most useful for frames, crankcases, and pistons.

Strength is not so important for other parts of the engine, and liners are an example where less than 40,000 tensile is satisfactory, but a high, uniform hardness over 200 Brinell becomes essential to reduce wear. Wear tests in our laboratories and others have shown clearly the need to maintain a high uniform hardness in cast iron for maximum wear resistance. Such irons may be patterned after the high-test type, except that cupola grades can contain 30 to 60 per cent steel and analyze 2.90 to 3.40 C, 0.80 to 1.75 Si, 1.25 to 2.50 Ni, and 0.40 to 1 Cr. Limitations of present-day machine tools keep the upper level of hardness under 240 to 260 Brinell. Pistons and rings frequently require a composition similar to liners, but higher in silicon and lower in nickel and chromium to compensate for their thinner sections; that is, 2.90 to 3.40 C, 1.25 to 1.90 Si, 1.25 to 2.50 Ni, and 0.30 to 0.50 Cr. Heads and exhaust manifolds require resistance to heat, with hardness and strength unimportant; hence low-silicon compositions higher in chromium than liners are more useful—as represented, for example, in the following composition: 3.10 to 3.45 C, 1.10 to 1.75 Si, 1.25 to 1.75 Ni, 0.50 to 0.90 Cr.

These modifications of cast iron are intended to accentuate some useful property such as strength, hardness, resistance to wear, or resistance to heat inherent in the material without changing its primary, useful properties. From our position on the skirmishing line in the development of newer cast-iron compositions we can see opportunities for further increasing the resistance to wear, to heat, and to corrosion by raising the alloy level from a limit of about 3 per cent nickel plus chrome, as at present, to $4^1/_2$ per cent nickel plus $1^1/_2$ per cent chrome, and as the limit of maximum machinable hardness is pushed up, from a top of 240 Brinell to possibly 300 or 350, liner compositions will follow the progress in machining. As an alternative, heat-treated cast irons, machined prior to hardening by oil quenching, offer opportunities for future development.

The author mentions the tendency for designers to lay too much emphasis on strength and not enough upon toughness. This factor is worthy of reemphasis, especially with respect to steels used in large units. It is impossible to work to millimicrometer tolerances, and the perfect fit, like perfect lubrication, becomes an ideal rather than a practical fact. Here, Diesel designers might take a page from the experience of locomotive builders who realize that heavy rods and castings must possess toughness and resistance to fatigue to endure the weave and whip of a locomotive bouncing over the rails. The result is that they guard the toughness factor in their steels jealously, and specifications in such parts as locomotive frame castings require high ductility or toughness to absorb dimensional irregularities in assembling and later to resist the vibrational stress and fatigue set up during operation.

These comments are intended merely to emphasize that much benefit remains to be derived by taking advantage of qualities inherent in the materials now used in engine construction.

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Inertia Supercharging of Engine Cylinders

By E. S. DENNISON,[1] PHILADELPHIA, PA.

A theory is developed for the supercharging effect produced in an engine cylinder when a long suction pipe is attached. As a preliminary to this theory, equations are given for suction and exhaust flow where no inertia effects are present. Results of the theory are compared with experiment, and some data are given on engine performance when supercharged in this manner.



A SMOOTH straight pipe of proper proportions, attached to the suction of a four-cycle engine cylinder, will effect an increase of its volumetric efficiency and hence of its capacity to develop power. This fact is known through various publications and needs no introduction. The supercharge is produced by accelerating the air column to a high velocity in the mid-part of the suction stroke; the kinetic energy so developed is then expended to force a surplus of air into the cylinder before valve closure. The process differs from supercharging with an auxiliary blower in that the compression power used is abstracted directly from the indicated power of the cylinder instead of being taken from the shaft; there are no extra mechanisms, no moving parts.

If the pipe is very long, oscillation of the column in its natural frequency plays a dominant part. Theoretically, maximum supercharge is attained when resonance occurs between pipe frequency and engine speed, i.e., when the period of free oscillation of the column is equal to the time required for the crankshaft to turn through 180 deg. With still greater lengths, the effect of oscillations may be favorable or adverse.

Based upon the condition of resonance as thus stated, Capetti[2] gives a formula for the most favorable length of intake pipe:

$$L = 7.5\, w_s/N$$

in which

L = length of pipe, inches
w_s = velocity of sound, about 1127 ft per sec at 70 F
= 1127 × 12, or 13,524 in. per sec
N = rpm of engine.

This formula leads to inconvenient lengths of pipe. At 500 rpm it gives 200 in., at 1000 rpm 100 in., etc. These lengths are such that they ordinarily could not be tolerated in practice.

Referring to actual tests, including some made at Langley Field by the N.A.C.A., Capetti finds that actual optimum lengths are less than given by the formula. This is still the case, as he finds, even after correcting for the effect that motion of air in the column has upon its period of oscillation. The resonance assumption alone is not a reliable guide in estimating a desirable pipe length.

Nevertheless, reported tests have usually shown the necessary length of pipe to be so long as to be impracticable for many purposes. Klüsener[3] obtained about 7 per cent maximum increase in volumetric efficiency using an 88-in. pipe attached to a small engine (4.73 in. bore by 7.09 in. stroke) operating at 750 rpm. He concludes that ordinarily a pipe several meters long would be required to obtain the best result. Such an addition to the engine is not easily justified if the gain is to be limited to 7 per cent or thereabout. Klüsener does not mention the diameter of pipe used in his tests.

Aside from possible resonance, the pipe may be considered merely as a device for storing kinetic energy. Then the diameter of pipe used is readily seen to be of first importance.

The mass of air in the column may be expressed as $M = C'LD^2$, its velocity as $w = C''/D^2$, and its energy as $E = CL/D^2$, in which D = pipe diameter, C, C', C'' = constants. Therefore, other things being equal, we should expect a higher supercharge if a small pipe is used. Other things, of course, are not equal; both resonance and pipe friction are affected in more or less significant ways.

Capetti applies a factor to the optimum length as found by the simple resonance formula, to correct for air motion in the pipe. It is derived by averaging the velocity of wave propagation outward contrary to the direction of motion, and inward with the motion. The factor is:

$$z = \left(1 - \frac{w_m^2}{w_s^2}\right)$$

in which w_m = mean air velocity in the pipe.

If w_m has even a high value, say $^1/_3\, w_s$, the factor $z = 0.89$; that is, it is not far from unity. But so far as it goes, it is in the right direction; for it reduces the length of pipe at which optimum conditions are to be expected.

The pipe-friction formula (Equation [21]) can be reduced to the form:

$$\Delta_p = \text{const.} \times \frac{G^2 L}{D^5}, \text{ approx.}$$

for a given density of air, in which G = rate of flow, lb per sec.

Because of the inverse fifth power of D in this formula, we must expect friction to set a limit to the useful reduction of pipe diameter. Also, since friction is directly proportional to length L, it helps to explain why the optimum length as found by test is less than the calculated, especially if the diameter is relatively small.

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[2] Antonio Capetti, "Effect of Intake Pipe on the Volumetric Efficiency of an Internal-Combustion Engine" (Translation), N.A.C.A. Tech. Memo no. 501, Feb., 1929.

Contributed by the Oil and Gas Power Division and presented at the Fifth National Oil and Gas Power Meeting, Pennsylvania State College, State College, Pa., June 8 to 11, 1932.

[3] Dr.-Ing. Otto Klüsener, "Saugrohr und Liefergrad," V.D.I. "Dieselmaschinen V," 1932.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Preliminary experiments were made using a cylinder of 8¼ in. bore by 12 in. stroke. The engine could be operated at speeds up to 900 rpm. There was no means of knowing volumetric efficiency with accuracy. Instead, the compression pressure was used as a measure of the air charge. Some results of these tests are given in Table 1.

TABLE 1

Compression pressures, 8¼-in. by 12-in. cylinder
(Lb per sq in. abs, corrected to standard barometer)

Tests at 800 Rpm			
Pipe diameter, in.	3.75	2.50	2.00
Ratio of cylinder section/pipe section	4.85	10.9	17.0
Pipe length, in.:			
0	395	395	395
12	...	...	430
36	414	445	432
48	...	466	428
54	...	468	...
60	...	474	418
72	...	474	411
Tests at Variable Speed			
Pipe diameter, in.		2.50	None
Pipe length, in.		48	0
Engine rpm:			
300		381	376
400		387	379
500		406	385
600		421	390
700		444	394
800		466	395
900		464	401

Comparing results at 800 rpm with 36-in. pipe, the advantage of the 2.50 in. diameter over the 3.75 in. is striking. Tests with large diameters were carried no further. Similarly the 2.00 in. diameter fails to give an advantageous result in any length; friction is excessive. The optimum pressure increase with 2.50-in. pipe is 20.0 per cent; the corresponding increase in weight of charge was roughly estimated at 14 per cent.

It is interesting to compare the best length of 2.50-in. pipe at 800 rpm with estimates based upon resonance:

Optimum length, test, in.	66
Resonant length, no velocity correction, in.	127
Resonant length, with velocity correction, in.	119

Optimum length by test is 52 per cent of simple resonant length. Actually little is gained by going beyond 48 in., due to flattening of the curve.

The outcome of these tests was so promising as to justify carrying the experiment further. Accordingly a cylinder head for an engine 9 in. bore by 12 in. stroke was so designed that inertia supercharging might be readily applied to it. Based upon the foregoing tests, together with an analysis, a diameter of 3 in. was selected, and the head arranged for connection of this size tubing. It was estimated that this diameter, in proper length, would be suitable up to a speed of 1000 rpm, or a piston speed of 2000 ft per min.

Tests of volumetric efficiency, also indicator cards and other data, have since been obtained from a single-cylinder engine fitted with the new head and attached suction pipe. At the same time an effort has been made to develop the theory of such a system so far that optimum pipe diameter and length, and a dependable estimate of volumetric efficiency, might be precalculated for any new design. The outline of this theory, with a comparison between calculated and test results, forms the subject of the present paper.

It should be observed that the particular type of application to which this discussion applies is the relatively high-speed engine, the piston velocity being 1500 ft per min or more. In such cases it is possible to obtain quite effective supercharging with a moderate length of suction pipe, provided the relative diameter of the pipe is small. Such results are obtained using a ratio of cylinder section/pipe section as high as 10 and in some cases perhaps higher. Incidentally, it usually follows that the pipe section is less than the area of the valve ports through which the air enters the cylinder.

Theory of Suction Process Without and With Inertia

It has been shown that the most favorable length of suction pipe is much less than was to be anticipated from the theory of resonance. Allowing for flattening of the volumetric efficiency curve near its maximum, a length only about half the resonant length may be preferable. Such being the case, the possibility of pipe resonance has little more than academic interest.

The theory of resonance is based upon a consideration of the harmonics of the piston motion and the resultant waves produced in the column.[4] In such a calculation, flow resistance at the valve, as well as pipe resistance, is usually ignored. This is probably justifiable with low-speed engines in which the valves are relatively large. It is not the case with engines operated at high piston speeds such as are discussed here. Instantaneous pressure drop at the valve may exceed 5 lb per sq in. Air being an elastic fluid, the movement of the piston is not directly effective to propagate waves in the column. Wide discrepancies are likely to arise from a computation based upon harmonics, unless an allowance for the valve resistance is properly made.

No attempt has here been made to do this. Instead the thermodynamic conditions affecting flow into the cylinder (or in some cases, out of it) have been made the starting point of the discussion. These are investigated first for cases in which no air column inertia is present. With this foundation established, the column is made to respond directly to forces estimated to act upon it; the case is treated as one of purely forced vibration. In general, a check of results indicates that the assumption is reasonably accurate for a pipe well under the resonant length.

In developing the theory, a little sacrifice of accuracy has been thought justifiable whenever it resulted in much easier use of the method. A calculation of this kind is of value only if it can be used as design procedure; this is impossible if complications are excessive. In any event it is well to avoid an appearance of accuracy which may be fictitious. It is thought that the method has been brought within limits of ordinary procedure.

System of Units

An unconventional system of thermodynamic constants is introduced for the sake of simplicity in calculation. It is obtained by applying consistently the following units:

Length	Inch
Force	Pound
Time	Second

The usual state equation for a gas has the form

$$p'v' = R'T$$

in which

p' = lb per sq ft
v' = cu ft per lb
R' = gas constant, ft-lb per lb per deg F. For air the numerical value is 53.35; the pound-mol constant for any gas is 1544
T = deg F abs.

In the present system pressure is expressed in lb per sq in. and specific volume in cu in. per lb; hence, the new constant R has the unit of in-lb per lb per deg F. Numerically it is $12 \times 53.35 = 640.2$ for air; the pound-mol constant for any gas is $12 \times 1544 = 18{,}528$.

Our state equation for air is then

$$pv = 640\,T \quad \ldots\ldots [1]$$

[4] P. Voissel, *Z.V.D.I.*, vol. 56, p. 721 (1912).

in which

p = pressure, lb per sq in.
v = specific volume, cu in. per lb.

The so-called mechanical equivalent of heat as determined at the Bureau of Standards is 778.54 ft-lb = 1 mean Btu. Converting this figure to the smaller unit, we have 9342.5 in-lb = 1 Btu; that is $J = 9342.5$.

The use of the constant J or its reciprocal A can be avoided by expressing all heat quantities in the mechanical unit, in-lb. In the present case this is accomplished by converting the specific heats of air:

At constant pressure c_p = 0.241 Btu per lb per deg F
= 2245 in-lb per lb per deg F
At constant volume, c_v = 0.172 Btu per lb per deg F
= 1605 in-lb per lb per deg F

$$\text{The ratio } k = \frac{c_p}{c_v} = \frac{2245}{1605} = 1.40$$

$$\text{Also } c_p - c_v = R$$

$$= 2245 - 1605 = 640.$$

This confirms the value for R directly calculated above. The gravity constant $g = 386$ in. per sec^2 completes the list of constants.

The adoption of these constants avoids once for all the recurring factors 12, 144, and 1728; likewise no further attention need be given to converting heat and work quantities where they occur in the same equation.

We may now proceed to a consideration of the several cases which arise in calculating air or gas flow either in intake or exhaust. First will be given the forms which the equations take when no inertia effects are present.

Case I—Inflow, Without Inertia Effects (refer to the sketch, Fig. 1).

The piston P is moving away from the cylinder head and drawing in air through the valve, of which the effective area is A_n. At a given instant the volume to the left of the piston is V, and the contained gas is at temperature T and pressure p; the latter is less than atmospheric pressure. Atmospheric conditions T_a and p_a exist outside the valve. We will consider the effects of an infinitesimal change; for convenience, the various quantities may be referred to an element of time dt.

It is evident that the process is irreversible; it involves the degradation of mechanical energy to thermal energy. Moreover, it is discontinuous; it cannot here be assumed that air passing through the valve with considerable pressure drop will issue into the cylinder at atmospheric temperature. That assumption is good only where the rate of flow is constant—as it is, for example, normally through a throttling valve. It is necessary to build up suitable equations from first principles, but it will be seen that this does not lead to any great complexity.

Setting up the elementary equation for energy changes in the fluid contained in the cylinder:

$$dU = dQ - dW \quad [2]$$

in which

U = total intrinsic energy of gas contained within the cylinder, in-lb
Q = energy added thermally, in-lb
W = external mechanical work done, in-lb.

To make use of Equation [2], we shall have to express the energy quantities in ascertainable common terms such as pressure and volume. For convenience, these quantities will be discussed in the reverse order. dW is work done against the piston. Then:

$$dW = pdV \quad [3]$$

in which dV is the increment of cylinder volume in time dt.

dQ is the thermal energy introduced by the incoming air. (Heat transfer from the cylinder walls is ignored.) Energy thus introduced is made up of two parts:

(*a*) Intrinsic energy of entering air. Assuming that c_v = constant within the range of temperature to be considered, this is $c_v \times T_a \times dG$, in which dG is an elementary weight of air, lb
(*b*) Flow work of entering air = $pvdG = RT_adG$

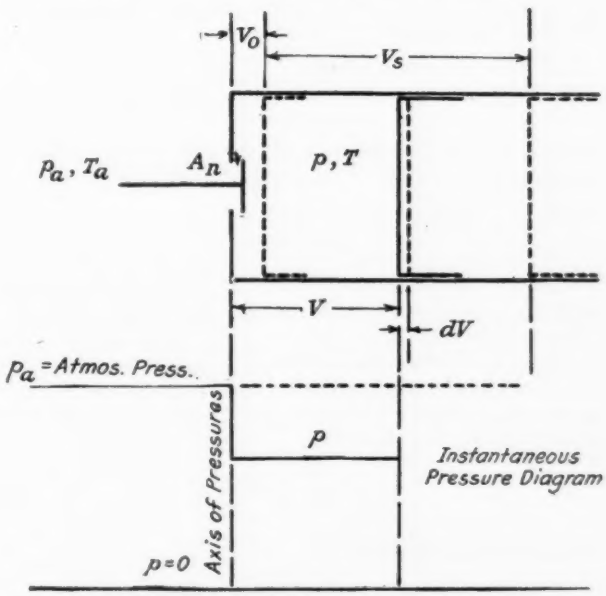


FIG. 1 DIAGRAMMATIC SKETCH, INERTIA EFFECTS ABSENT

Combining (*a*) and (*b*) we have:

$$dQ = (c_v + R)T_adG$$

$$= c_pT_adG = kc_vT_adG \quad [4]$$

dU is the change of intrinsic energy of the gas in the cylinder.

Following the assumption just made,

$$U = Gc_vT \quad [5]$$

in which G is weight of air in the cylinder, lb.

Now $pV = GRT$ and $GT = pV/R$. Substituting this expression for GT in Equation [5], we obtain:

$$U = \frac{c_v}{R}pV$$

But $R = c_p - c_v$, and $\frac{c_v}{c_p - c_v} = \frac{1}{k-1}$. Then:

$$U = \frac{1}{k-1}pV \quad [6]$$

That is, the intrinsic energy in volume V of a given gas is fully determined when p and V are known, the expression being independent of temperature T.

We may now combine Equations [3], [4], and [6] in Equation [2]:

$$\frac{1}{k-1}\,d(pV) = kc_vT_adG - pdV$$

$$pdV + Vdp = (k-1)\,(kc_vT_adG - pdV)$$

$$Vdp = (k-1)\,kc_vT_adG - kpdV$$

But $(k-1)c_v = R$, therefore, $Vdp = k(RT_adG - pdV)$, and:

$$dp = \frac{k}{V}(RT_adG - pdV) \quad \ldots\ldots\ldots\ldots\ldots [7]$$

To go further, it is necessary to express dG in terms of the variables which fix the rate of flow—namely, pressure p, valve section A_n, also T_a and p_a. A simple form of flow equation is desirable, for example:

$$G = c\mu A\,\frac{\sqrt{(p_a-p)p}}{\sqrt{T_a}}\,t \text{ lb}^5 \quad \ldots\ldots\ldots\ldots [8]$$

in which A = area of orifice, sq in., and μ is the flow coefficient. For the present purpose $\mu A = A_n$, by definition.

t = time, sec
c = a constant.

With the present system of units, $c = 1.094$ approximately, when $(p_a - p)$ is small relative to p_a; the formula is usually applied only to such cases.

It can easily be learned, however, how much error arises if the same formula is applied to a wider range of pressures. To take an extreme, assume $p = 0.53\,p_a$, the critical ratio. The exact theoretical formula then gives:

$$G = 0.53\,A_n\,\frac{p_a}{\sqrt{T_a}}\,t \quad \ldots\ldots\ldots\ldots [8a]$$

whereas the trial formula gives:

$$G = 1.092\,A_n\,\frac{\sqrt{0.53 \times 0.47}\,p_a}{\sqrt{T_a}}\,t$$

$$= 0.545\,A_n\,\frac{p_a}{\sqrt{T_a}}\,t$$

In other terms, the trial formula becomes accurate at the critical ratio if

$$c = 0.53/0.545 \times 1.092 = 1.062$$

Between the two extreme ratios of pressure, the variation of the adjusted value of c is nearly linear. For the present purpose we then take $c = 1.08$, and

$$G = 1.08\,A_n\,\frac{\sqrt{(p_a-p)\,p}}{\sqrt{T_a}}\,t \text{ lb} \quad \ldots\ldots\ldots [9]$$

This form may be used throughout with an error averaging about 1 per cent. Its accuracy is adequate, in view of possible errors for example in A_n.

Substituting in Equation [7], in the differential form:

$$dp = \frac{k}{V}\left[1.08\,RA_n\,\sqrt{T_a}\,\sqrt{(p_a-p)p}\,dt - pdV\right] \ldots [10]$$

A little inspection will show that this expression for dp does not lend itself to a direct solution. If t is taken as the independent variable, dV may be expressed in those terms, but A_n only with difficulty. Thereafter, the algebra is certain to become involved. A step-by-step solution must be accepted instead. This may be obtained in several ways. The procedure developed here is convenient and reasonably accurate; also it has the advantage that trial-and-error methods are avoided.

Let θ = engine crank angle. Divide Equation [10] through by $d\theta$; also factor p out of the bracket.

$$\frac{dp}{d\theta} = \frac{kp}{V}\left[1.08\,RA_n\,\sqrt{T_a}\,\sqrt{\frac{p_a-p}{p}}\,\frac{dt}{d\theta} - \frac{dV}{d\theta}\right] \ldots [11]$$

If n = rps of the engine,

$$\theta = 360\,nt$$

and

$$\frac{dt}{d\theta} = \frac{t}{\theta} = \frac{1}{360\,n}$$

Then:

$$\frac{dp}{d\theta} = \frac{kp}{V}\left[\frac{1.08\,RA_n\,\sqrt{T_a}}{360\,n}\,\sqrt{\frac{p_a-p}{p}} - \frac{dV}{d\theta}\right] \ldots [12]$$

Now $R = 640$, and we may assume $T_a = 530$ F abs (70 F scale).

Substituting,

$$\frac{dp}{d\theta}^6 = \frac{kp}{V}\left[\left(44.2\,\frac{A_n}{n}\,\sqrt{\frac{p_a-p}{p}}\right) - \frac{dV}{d\theta}\right] \ldots [13]$$

The quantities V and $dV/d\theta$ are fixed for any value of θ by engine dimensions: piston area, crank radius r, and ratio r/l of crank radius to connecting-rod length; the familiar equations need not be set down here. The quantity A_n is based upon instantaneous valve section A and flow coefficient μ. From known data this can be estimated for a given case. These three quantities may be tabulated for various crank positions θ.

Given then the cylinder pressure p existing at any point, Equation [13] enables one to calculate quite readily the *slope* of the pressure line with respect to crank angle. To avoid the difficulty of integration, it is now assumed that $dp/d\theta$ for any point a may be taken as a constant for the purpose of extrapolation to the next point b. That is,

$$(\Delta p)_{ab} = \left(\frac{dp}{d\theta}\right)_a \Delta\theta \quad \ldots\ldots\ldots\ldots\ldots [14]$$

and

$$p_b = p_a + (\Delta p)_{ab} \quad \ldots\ldots\ldots\ldots\ldots [15]$$

A check was made upon this procedure using a trial-and-error method developed from Equation [10] with results substantially identical. The results of Equations [13–15] converge in the sense that an error arising in one step tends to be corrected in

[5] S. A. Moss, Trans. APM-3, 1928.

[6] NOTE: Instead of dividing Equation [10] by $d\theta$, it may be divided by dV, resulting in a form simpler than Equation [13]. In place of Equation [11], we then have:

$$\frac{dp}{dV} = \frac{kp}{V}\left[1.08\,RA_n\,\sqrt{T_a}\,\sqrt{\frac{p_a-p}{p}}\,\frac{dt}{dV} - 1\right] \ldots [11a]$$

Reducing this as above, we obtain:

$$\frac{dp}{dV} = \frac{kp}{V}\left[\left(44.2\,\frac{A_n}{n}\,\frac{d\theta}{dV}\,\sqrt{\frac{p_a-p}{p}}\right) - 1\right] \ldots [13a]$$

The product $44.2\,A_n(d\theta/dV)$ is predetermined by engine design and the routine calculation becomes very simple. However, this procedure was not used, for angle (i.e., time) increments are more advantageous than volume increments, especially when dealing with acceleration and velocity, as will later be done.

the next. At most, where the change in $dp/d\theta$ is rapid, the calculated line will be a "zig-zag" if $\Delta\theta$ is too large. Practically, intervals of 5 deg are required near top center; at 40 deg and over from top center, intervals of 10 deg or more may be used.

A sample calculation based upon the foregoing equations is given in Fig. 11. The usual procedure is to start at top center, assuming atmospheric pressure in the cylinder, and to continue to the point of closing of the inlet valve.

Case II—Outflow, Without Inertia Effects ($p < 2p_a$).

This case arises in considering the exhaust process, also sometimes near the close of the suction process.

Equation [7] applies here as in Case I. But to find dG we must use:

$$G = -1.08\, A_n \frac{\sqrt{(p - p_a)p_a}}{\sqrt{T}}\, t \text{ lb} \quad \ldots\ldots\ldots [16]$$

Proceeding as before, we obtain:

$$\frac{dp}{d\theta} = \frac{k}{V}\left[-\frac{1.08\, RA_n\sqrt{T}}{360\, n}\sqrt{(p - p_a)p_a} - p\frac{dV}{d\theta}\right]$$

$$= -\frac{k}{V}\left[\left(\frac{7.36\, A_n}{n}\sqrt{T\,(p - p_a)}\right) + p\frac{dV}{d\theta}\right] \ldots [17]$$

For exhaust gas at 1200 F, $k = 1.36$ approximately.

To be exact, we should have to take account of the relation

$$\frac{T}{T_o} = \left(\frac{p}{p_o}\right)^{\frac{k-1}{k}} \quad \ldots\ldots\ldots\ldots [18]$$

in which p_o, T_o refer to any initial state, such as the point of release. This can be conveniently done for the purposes of Equation [17] by plotting the relationship in the form:

$$\sqrt{T} = \text{constant} \times (p)^{0.1325} \quad \ldots\ldots\ldots [19]$$

in which the exponent $0.1325 = \dfrac{k-1}{2k}$

Case IIa—Outflow ($p \geqq 2p_a$).

This case arises immediately after the point of release, the cylinder pressure being then high.

The flow formula for this case is that of Equation [8a]. The equation reduces to the form:

$$\frac{dp}{d\theta} = -\frac{kp}{V}\left[\left(\frac{0.943}{n}\, A_n\sqrt{T}\right) + \frac{dV}{d\theta}\right] \quad \ldots\ldots [20]$$

To secure reasonable accuracy in this case, the temperature relation, Equation [19], must be used.

Case III—Inflow Including Effect of Inertia in Attached Pipe.

Mention has already been made of basic assumptions used in this case, particularly the neglect of possible resonance. However, the working assumptions require a detailed statement and some discussion. Not all variables can be taken in account. The problem is to make such a choice that the principal factors shall be allowed for and that the resulting equations shall not be unwieldy.

Assumptions:

1 Mass flow continuity. At any instant the mass rate of flow past each point in the path is assumed constant
2 Density changes of air *in the column* are assumed negligible. For purposes of the column calculation, air is treated as an incompressible fluid having the density of atmospheric air
3 Flow resistance in the pipe is assumed negligible. This is to say, in effect, that if preliminary calculation shows that friction will be a considerable item with a particular pipe size, a larger size should be selected. However, see Appendix A
4 Velocity pressure in the column is assumed to be completely converted to static pressure before the valve.

Actually, assumption 2 follows from 1. For, manifestly, we may not allow density changes in the column without also providing that at some time air entering is not equal to air leaving.

The function of the column is to serve as a reservoir of kinetic energy. The effect of assumption 2 is to underestimate somewhat the energy which will be stored under given conditions. As an example, assume that air is flowing through a tube of 3 in. diameter at a rate of 1.5 lb per sec. Calculated from the assumption,

$$\text{Velocity } w = 4900 \text{ in. per sec}$$

and kinetic energy

$$E = \frac{1}{2} \times \frac{7.07}{23{,}100} \times \frac{(4900)^2}{366}$$

$$= 9.52 \text{ in-lb-per inch of length.}$$

The figure 7.07 (sq in.) is the area of a 3-in. pipe; 23,100 (cu in. per lb) is the specific volume of atmospheric air.

Taking account of the compressibility of air, we find that the actual conditions in the pipe will be:

$$p = 13.18$$
$$T = 513$$
$$v = 24{,}900$$
$$w = 5280 \text{ in per sec}$$
$$E = \frac{1}{2} \times \frac{7.07}{24{,}900} \times \frac{(5280)^2}{386}$$
$$= 10.24 \text{ in-lb per inch of length.}$$

Thus the energy is underestimated by about 7 per cent in this case.

Quite high air velocities can be used without resulting in excessive pipe friction. To take again the example used in discussing kinetic energy (1.50 lb per sec through a 3-in. pipe) we will refer to the first case, in which atmospheric density of air is assumed to hold. The pressure drop in a pipe, for a fluid in turbulent motion, is given by:

$$\Delta p = \frac{2\, fL\rho w^2}{gD} \quad \ldots\ldots\ldots\ldots\ldots [21]$$

in which

L = pipe length
ρ = density of fluid
w = velocity of flow
D = diameter of pipe
g = acceleration of gravity
f = coefficient of friction

and in which any consistent set of units may be used.

Using the data of McAdams,[7] we find in this case $f = 0.0043$ for ordinary steel pipe. Then $\Delta p = 0.0078$ lb per sq in. per inch of length. For a length of 50 in. this becomes only 0.39 lb per sq in.; this is quite small as compared with the pressure loss simultaneously occurring at the valve. A rule might be to exclude from consideration by preliminary estimate any case

[7] Walker, Lewis, and McAdams, "Principles of Chemical Engineering," pp. 73–88.

in which maximum pipe resistance would exceed 0.50 or 0.75 lb per sq in., and in the calculation to neglect resistance. This is done in the derivations here given. Appendix A shows how a resistance correction may be applied to the calculation.

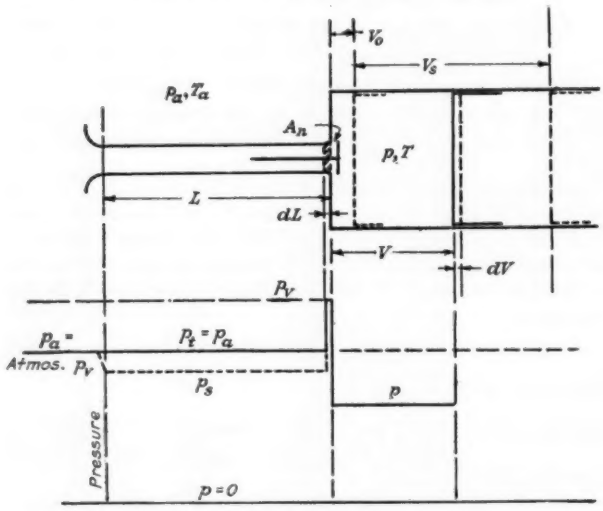


FIG. 2 DIAGRAMMATIC SKETCH, SUCTION PIPE ATTACHED

Assumption 4 does not involve any approximation when the junction between the pipe section and the ports is well shaped. But both assumptions 1 and 4 fail to hold if a considerable dead space exists between the column and the valve. In that case large losses may intervene, and the methods of calculation here given would require modification.

Accepting the assumptions as laid down, the derivation proceeds as follows:

The case is represented schematically by Fig. 2. Atmospheric pressure p_a exists at one end of the air column; pressure p_v (and temperature T_v) at the inner end in the last element of pipe length before the valve. The difference in the two pressures at any instant is a measure of the force acting to accelerate the column. Applying the second law of motion,

$$a = \frac{dw}{dt} = \frac{F}{M} \text{ in. per sec}^2$$

in which

F = force acting on air column, lb
M = mass of column, lb/g.

Evaluating,

$$F = A_p (p_a - p_v)$$

and

$$M = \frac{LA_p\rho_a}{g}$$

in which A_p = cross-section of pipe. Note that velocity and acceleration are taken positive in the inflow direction.
Then

$$\frac{dw}{dt} = (p_a - p_v)\frac{g}{L\rho_a} \quad \text{[22]}$$

and

$$\frac{dw}{d\theta} = (p_a - p_v)\frac{g}{360\,L\rho_a n} \quad \text{[23]}$$

An expression for dG is obtainable from consideration of the pipe:

$$dG = wA_p\rho_a dt$$

A second expression is based upon valve conditions:

$$dG = 1.08\,A_n\frac{\sqrt{(p_v - p)p}}{\sqrt{T_v}}\,dt$$

Equating the two expressions, we have:

$$wA_p\rho_a = 1.08\,A_n\frac{\sqrt{(p_v - p)p}}{\sqrt{T_v}}$$

$$\frac{(p_v - p)p}{T_v} = \left(\frac{A_p\rho_a}{1.08} \times \frac{w}{A_n}\right)^2$$

$$p_v - p = \frac{T_v}{p}\left(\frac{A_p\rho_a}{1.08} \times \frac{w}{A_n}\right)^2 \quad \text{[24]}$$

Referring to Equation [12], we obtain a similar expression for the present case:

$$\frac{dp}{d\theta} = \frac{kp}{V}\left[\frac{1.08\,RA_n\sqrt{T_v}}{360\,n}\sqrt{\frac{p_v - p}{p}} - \frac{dV}{d\theta}\right] \quad \text{[25]}$$

If a general analytical solution is sought, this must be built up from Equations [23], [24], and [25]. The difficulty here exceeds that of the previous case. Instead, the procedure for a step calculation is given.

A simplification of Equations [24] and [25] will first be made. In those expressions T_v is a function of p_v given by the relation $T_v/T_a = \left(\frac{p_v}{p_a}\right)^{\frac{k-1}{k}}$. If Equation [24] is used to solve for $(p_v - p)$, the variation of T_v must be allowed for by trial and error. But the variation of T_v is actually small, and its average approximates closely to T_a. Substitution of T_a for T_v simplifies the work. A trial has shown that the error so introduced is very slight.

Now introducing $T_a = 530$ and substituting other numerical values, the required equations can be summarized as follows:

$$\frac{dw}{d\theta} = 24{,}800\,\frac{(p_a - p_v)}{Ln} \quad \text{[26]}$$

$$\frac{dp}{d\theta} = \frac{kp}{V}\left[\left(44.2\,\frac{A_n}{n}\sqrt{\frac{p_v - p}{p}}\right) - \frac{dV}{d\theta}\right] \quad \text{[27]}$$

$$p_v = \frac{8.53 \times 10^{-7}}{p}\left(A_p\frac{w}{A_n}\right)^2 + p \quad \text{[28]}$$

The procedure in a step calculation is now given. Assume that for point m the quantities p, p_v, and w are known; other quantities determined by design are also known. We will proceed to point n.

(1) Using Equation [26], calculate $\left(\frac{dw}{d\theta}\right)_m$

(2) By extrapolation, obtain w_n:

$$\Delta w = \Delta\theta\left(\frac{dw}{d\theta}\right)_m$$

$$w_n = w_m + \Delta w$$

(This extrapolation of w is similar in principle to that of p.)

(3) Using Equation [27], obtain $dp/d\theta$. By extrapolation calculate p_n.

(4) Given w_n and p_n calculate p_{vn} from Equation [28].

This completes the step computation, which is then repeated. Fig. 12 gives a sample calculation. Steps of 5 deg must be taken until 40–50 deg past top center. Thereafter 10-deg or 15-deg intervals suffice.

Case IV—Outflow, With Attached Pipe:

This case arises where inertia in an exhaust column is used to improve scavenging of the clearance space. Some outflow occurs also just before closure of the inlet valve.

The derivations are similar to the foregoing and need not be repeated in detail. The equations take the following forms:

$$\frac{dw}{d\theta} = \frac{1.072\ (p_a - p_v)}{L\rho_p n} \quad \text{..... [29]}$$

Here ρ_p is density of gas in the pipe and may be estimated from average exhaust gas temperature.

$$\frac{dp}{d\theta} = -\frac{k}{V}\left[\left(\frac{7.36\ A_n}{n}\sqrt{T\ (p - p_v)}\right) + p\ \frac{dV}{d\theta}\right] \quad \text{.... [30]}$$

$$p_v = p - 0.0584\ T\left(\frac{wA_p\rho_p}{A_n}\right)^2, \text{ approx} \quad \text{..... [31]}$$

The procedure in calculation is similar to that in case III. Velocity w and pressure p are extrapolated, and p_v computed to complete the step.

Volumetric Efficiency

A step-by-step computation of weight G flowing into the cylinder can be made simultaneously with any of the foregoing, and volumetric efficiency μ_v estimated. This is unnecessary. The energy Equation [2], together with the expression for intrinsic energy, Equation [6], can be used instead, much more easily.

Apply Equation [2] to the entire period from start of suction to closure of the inlet valve. Transposing:

$$Q = \Delta U + W \quad \text{..... [32]}$$

in which

ΔU = increment of intrinsic energy within the cylinder

W = total work done by air pressure against piston

Q = thermal energy added. In this case, since heat transfer from the walls is ignored, this must all be accounted for by incoming air.

For estimation of W, refer to Fig. 3. The net work area to the point of closure is:

$$W = A_1 - A_2$$

$$A_1 = p_a V_s - A_1'$$

in which A_1' is the area $msnm$ and V_s is stroke volume.

$$A_2 = p_a \Delta V + A_2'$$

in which A_2' is the area $rqnr$ and ΔV is the partial return stroke volume.

$$A_1 - A_2 = p_a(V_s - \Delta V) - (A_1' + A_2')$$

$$= p_a\ (V_f - V_o) - p_m V_s \quad \text{..... [33]}$$

in which

V_o = clearance volume

V_f = cylinder volume at valve closure

p_m = mean indicated pressure from area $msnqrm$

Applying Equation [6],

$$U = U_f - U_o$$

$$= \frac{1}{k-1}\ (p_f V_f - p_o V_o)$$

Taking $k = 1.40$ and $p_o = p_a$

$$U = 2.50\ (p_f V_f - p_a V_o) \quad \text{..... [34]}$$

in which p_f is final pressure, lb per sq in. abs.

Combining,

$$Q = 2.50\ (p_f V_f - p_a V_o) + p_a(V_f - V_o) - p_m V_s$$

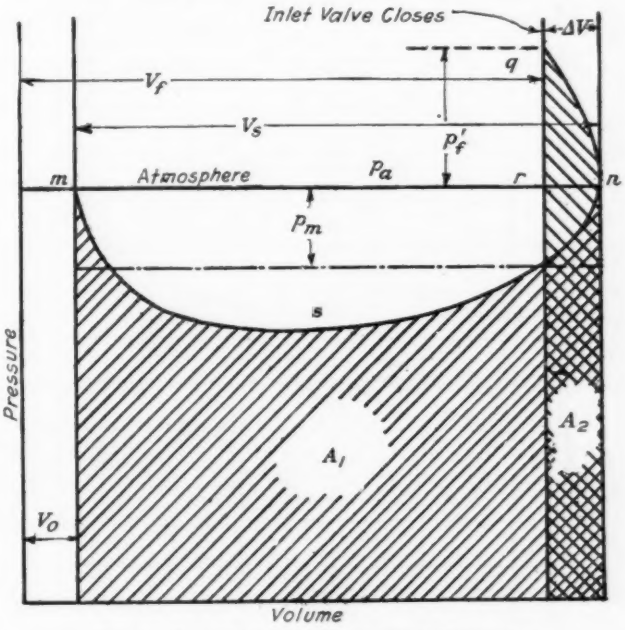


Fig. 3 Calculation of Volumetric Efficiency

Now let $p_f' = p_f - p_a$, so that p_f' = gage pressure at closure. Then:

$$Q = 3.50\ p_a(V_f - V_o) + 2.50\ p_f' V_f - p_m V_s \quad \text{.... [35]}$$

This equation expresses in in-lb the net heat which has entered the cylinder. The heat per lb of air is

$$q = c_p T_a$$

Therefore, the weight entering is

$$G = \frac{Q}{c_p T_a}$$

Volumetric efficiency is then

$$\mu_v = \frac{G}{G_s} = \frac{G}{V_s \rho_a} \quad \text{..... [36]}$$

In which G_s is the weight of atmospheric air required to fill stroke volume V_s.

This method may be applied to any case. A slight error arises from the fact that any small amount of air flowing outward carries with it a little more energy than atmospheric air flowing in; this can be overlooked. The method applies where a pipe is attached, since our assumptions provide that pipe velocity at valve closure shall be zero. Hence, no kinetic energy is unaccounted for.

The two pressures, p_f' and p_m, are seen to be the direct results of the computation. For a given trial pipe diameter, two or three values, together with those for $L = 0$, will suffice for plotting curves of each. From these the volumetric efficiency curve follows readily.

Results of Tests and Comparison With Theory

Both volumetric-efficiency tests and low-pressure indicator cards have been obtained with the 9-in. by 12-in. single-cylinder

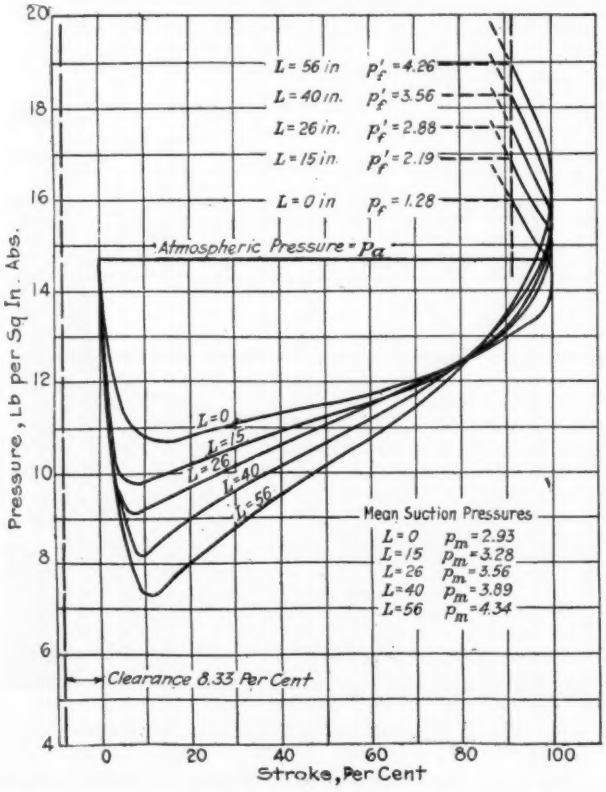


Fig. 4 Calculated Pressure-Volume Diagram of Suction Stroke

(Cylinder, 9 in. by 12 in.; 1000 rpm; pipe diameter, 3 in.; two-suction valves 2⅜ in. diam, open 10 deg B.T.C., close 42 deg A.B.C.)

test engine already mentioned. These data afford an opportunity for comparison with the results of theory, at least for one fairly typical case.

The engine is so designed as to permit adjustment of valve timing and of compression ratio. However, the test conditions selected for comparison were all made with one setting of valves, as follows:

Exhaust valve opens	45 deg B.B.C.
Exhaust valve closes	20 deg A.T.C.
Inlet valve opens	10 deg B.T.C.
Inlet valve closes	42 deg A.B.C.

This timing is not represented as necessarily the best; for example, certain results indicate that in this design the inlet-valve opening should be earlier. However, experiments showed that later closing of the intake valve would in no case have any marked advantage. The engine is fitted with two intake and two exhaust valves; port diameter is 2.625 in. and valve lift 0.625 in.

For the volumetric efficiency measurements, the engine was fitted with an equalizing tank attached to the suction. The tank volume was about 220 times the piston displacement, and it was suitably baffled to quiet the air movement within. Air was drawn in through one or another of a series of rounded nozzles, special attention being given to obtaining a smooth surface finish. Thus the tank was maintained a little under atmospheric pressure, the test calculations requiring correction for this fact. Owing to the large volume of the tank, pressure fluctuations as recorded by manometers were very slight, being limited to $^1/_{32}$ in. of alcohol, approximately. An estimate of errors indicated that individual readings should be correct to about 1 per cent. The scattering of points is greater than this. However, an independent check was obtained by overlapping the range of speeds tested with two different sizes of nozzle, so that the tank pressure and other conditions were quite changed. Owing to the good check so found, it can be said that average results are in error not more than 1 per cent.

Indicator cards were taken with a point-by-point balanced-pressure diaphragm-type indicator as made by the American Instrument Company; certain minor alterations have been made. Care was taken to employ a short passage from combustion space to diaphragm. Using a thin diaphragm (0.003 in.) and correcting when necessary for the pressure difference required to "click" the instrument under static conditions, very satisfactory and consistent results are obtained. Individual

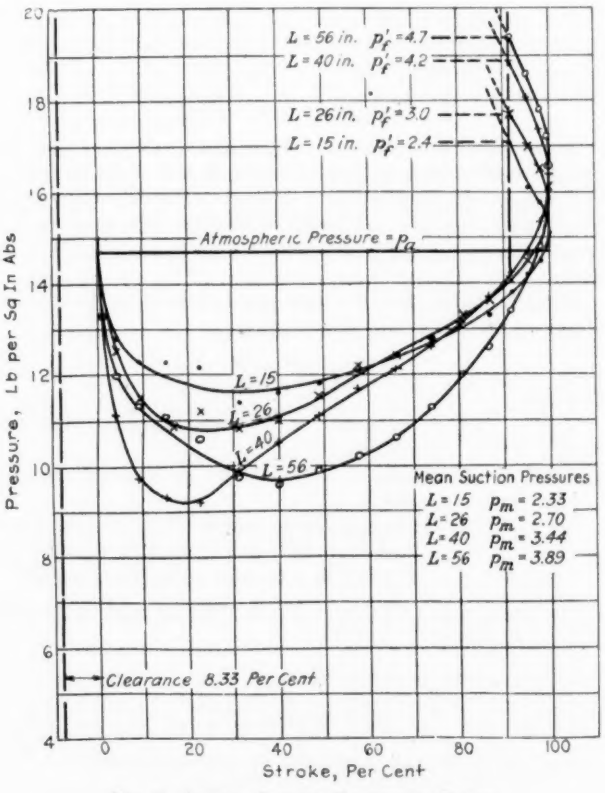


Fig. 5 Actual Suction Indicator Cards

(Tested under conditions assumed in calculating Fig. 4.)

readings on low-pressure cards appear to be good to 0.2 lb per sq in.

All experimental results involve a certain length of suction pipe before the valve and in all cases used in comparison the diameter is 3 in. The design of the head is such that about a 10-in. column is an unavoidable part of the system. After attaching the necessary rounded mouthpiece, the minimum length for which reliable data can be obtained is 15 in. So test results for pipe length $L = 0$ are unavailable. This prevents,

for example, a direct check upon the formula, Equation [13], for calculating a suction card where inertia is absent.

For indicator-card calculations a knowledge of the valve coefficient μ or the effective area A_v is necessary. In this case actual tests were available.[8] From those and other published data probably any design could be estimated within 5 per cent—a sufficient degree of accuracy for design purposes.

Fig. 4 represents a series of theoretical cards; the actual cards (Fig. 5) are taken under the same conditions assumed in the calculation. In general a fair agreement is obtained. Certain characteristic differences are evident:

(1) The calculated minimum pressure is always too low
(2) The calculated mean indicated pressure p_m is too high
(3) The calculated final gage pressure p_f' (at valve closure) is generally low. The error here is small, averaging 0.35 lb per sq in.

The principal sources of error are undoubtedly (*a*) heat transfer to the air, which is ignored, and (*b*) inaccuracy of A_v for a given condition of flow. It is unlikely that defective performance of the indicator contributes much to the discrepancy. An exception may be the case $L = 56$; here the test card seems inconsistent with the others and departure from the theoretical card is greatest.

A more detailed picture of the case $L = 40$ in. is given by Fig. 6. The characteristic differences mentioned are apparent here.

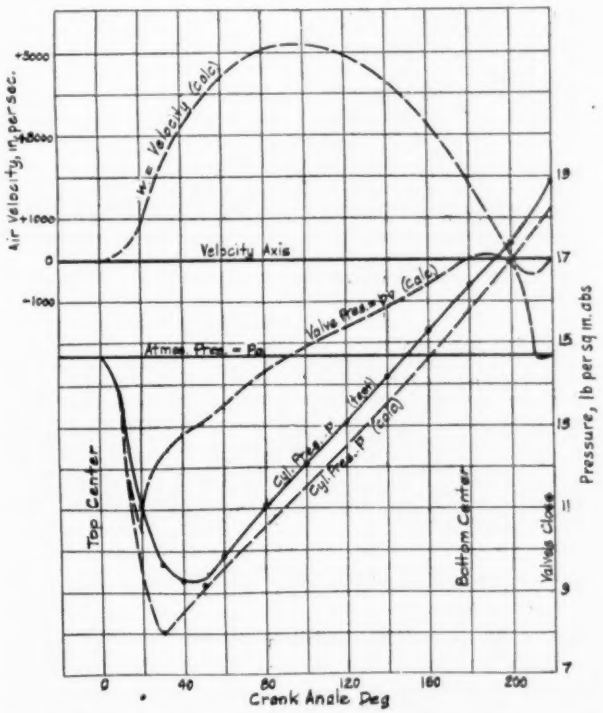


FIG. 6 CRANK ANGLE-PRESSURE DIAGRAMS BY CALCULATION AND TEST
(Rpm, 1000; $L = 40$ in.; also shows calculated valve pressure p_v and air velocity w.)

However, the agreement in the general shape of the curves for cylinder pressure p is satisfactory. The course of the calculated pressure p_v, before the valve, is of interest. No measurement of the corresponding actual pressure has been made. Similarly there is no direct check upon the computed velocity of air w.

[8] Trans. OGP-53-6 (1931).

Volumetric efficiency at 1000 rpm is shown by Fig. 7. As might be expected, the calculated figures are too high, the negative effect of heat absorption from the walls having been neglected. The average correction ($L = 15$ in. to 56 in. inclusive) is —8.0 per cent.

The test result curve is somewhat steeper than the calculated.

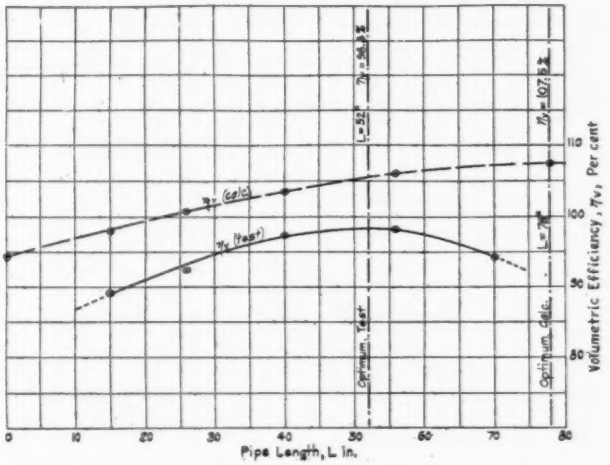


FIG. 7 VOLUMETRIC EFFICIENCY BY CALCULATION AND TEST
(Rpm, 1000; pipe diameter, 3 in.)

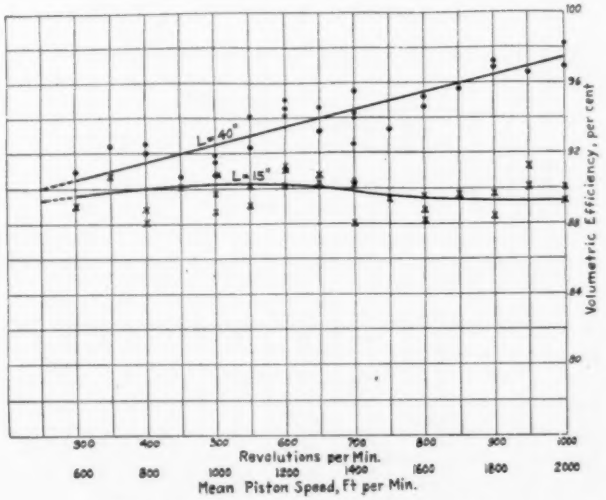


FIG. 8 VOLUMETRIC EFFICIENCY BY TEST
(Cylinder 9 in. by 12 in., with suction pipe 3 in. diam.)

This is possibly due to underestimating kinetic energy of the column; that error increases with pipe length.

The discrepancy in optimum length shows up again in this case. What part pipe friction may play is not known, for no calculation has been made in which this was allowed for. A comparison of best length can again be made:

Optimum length, test, in.....	52
Optimum length, present calculation, in.....	78
Resonant length, no velocity correction, in.....	101
Resonant length, with velocity correction, in....	94

Here again the best test length is 52 per cent of the simple resonant length; agreement with the first case can be only coincidence.

The actual maximum efficiency as tested (98.3 per cent) is itself of some interest. At $L = 15$ it is 89.3 per cent, or an in-

crease of 10.0 per cent in weight of charge. The test curve extrapolated to $L = 0$ gives about 82.5 per cent. While this figure cannot be confirmed, it is not unduly low for a mean piston speed of 2000 ft per min. If this estimate can be accepted, the increase of charge due to the pipe is 19.1 per cent.

Fig. 8 gives the effect of engine speed, with $L = 40$ in. and $L = 15$ in. The continual increase of air charge throughout the tested speed range, using $L = 40$ in., is striking. The effect is somewhat higher with light loads and lower when operating over 90 bmep. Increased load capacity is shown by exhaust temperature; the reduction is normally 100 F, and more with a large load. The limit of load with clear exhaust was raised by 15 bmep approximately.

In these tests compression ratio was adjusted to give 410 lb per sq in. gage compression pressure in each test. The fuel injection system has not yet been specifically adapted to supercharged operation.

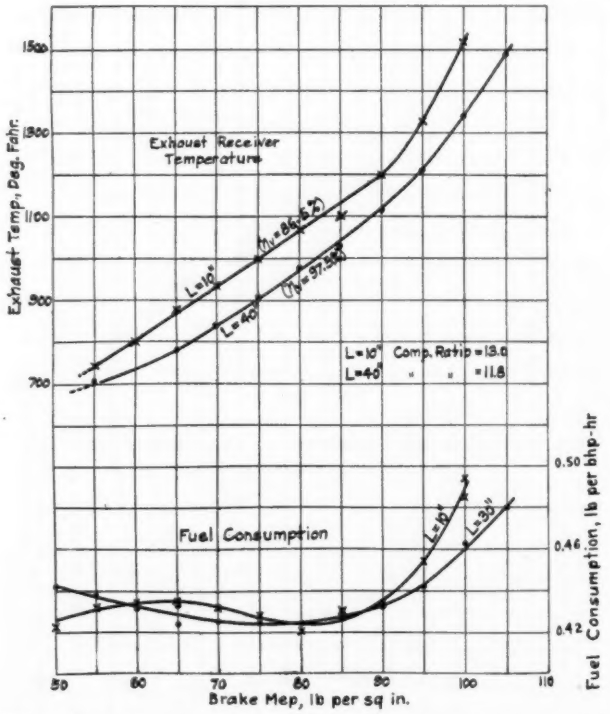


FIG. 9 FUEL CONSUMPTION AND EXHAUST-RECEIVER TEMPERATURE (Single-cylinder engine; suction pipe, 3 in. diam; L = suction pipe length; 9-in. bore by 12-in. stroke; 1000 rpm.)

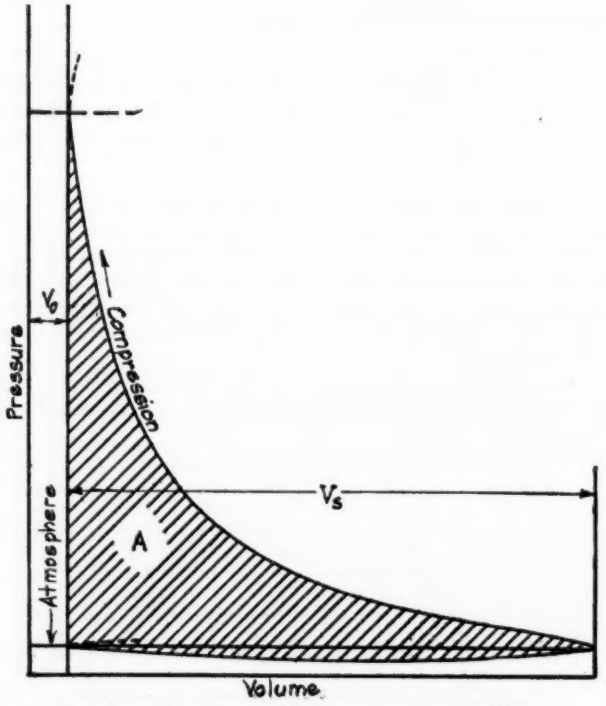


FIG. 10 DIAGRAMMATIC SKETCH, COMPRESSION WORK

N=1000 n=16.67 L=0

1	2	3	4	5	6	7	8	9
θ	V	A_n	$dV/d\theta$	p	14.7-p	( )	[ ]	$dp/d\theta$
0	63.6	0.30	0	14.70	0	0	0	0
5	64.5	0.60	0.794	14.70	0	0	-0.794	-0.25
10	71.2	0.90	1.556	13.45	1.25	0.727	-0.829	-0.22
15	80.4	1.30	2.25	12.35	2.35	1.50	-0.75	-0.15
20	93.4	1.70	2.97	11.60				

(13) $\frac{dp}{d\theta} = \frac{Kp}{V}\left[(44.2\frac{A_n}{n}\sqrt{\frac{14.7-p}{p}} - \frac{dV}{d\theta}\right]$ $\left(\frac{44.2}{n} = 2.65\right)$

FIG. 11 SAMPLE CALCULATION, SUCTION WITHOUT PIPE

N=1000 n=16.67 D=3.0" L=40"

1	2	3	4	5	6	7	8	9	10	11	12	13
θ	V	A_n	$dV/d\theta$	p	( )	[ ]	$dp/d\theta$	w	$P_v - p$	P_v	$P_0 - P_v$	$dw/d\theta$
0	63.6	0.30	0	14.70	0	0	0	0	0	14.70	0	0
5	64.5	0.60	0.794	14.70	0	-.794	-.253	0	0	14.70	0	0
10	71.2	0.90	1.556	13.43	0	-1.556	-.410	0	0	13.43	1.27	47.2
15	80.4	1.30	2.25	11.38	0.354	-1.896	-.374	256	0.12	11.50	3.20	119.0
20	93.4	1.70	2.97	9.51	1.810	-1.46	-.208	830	1.07	10.58	4.12	152.6
25	109.6	2.15	3.56	8.47				1593				

(25) $-P_v - p = \frac{8.53\cdot10^{-7}}{p}\left(A_p\frac{w}{A_n}\right)^2 = \frac{4.28\cdot10^{-3}}{p}\left(\frac{w}{L_n}\right)^2$ — Cols. 9,5,8,10.

(26) $-\frac{dw}{d\theta} = 24{,}800\frac{(P_0-P_v)}{L_n} = 37.2(P_0-P_v)$ — Cols. 12, 13 — 9.

(27) $-\frac{dp}{d\theta} = \frac{Kp}{V}\left[(44.2\frac{A_n}{n}\sqrt{\frac{P_0-p}{p}}) - \frac{dV}{d\theta}\right]$ — Cols. 10,5,3,4,2 — 5.

$\left(\frac{44.2}{n} = 2.65\right)$

FIG. 12 SAMPLE CALCULATION, SUCTION WITH PIPE

with $L = 15$ is to maintain practically constant efficiency with rising speed. The normal expectation is a drooping curve.[9]

Fig. 9 compares engine test results with $L = 10$ in. and $L = 40$ in. While not outstanding in themselves, they serve to show some typical effects of supercharging. Specific fuel consumption

Exhaust temperatures may appear excessive. They were taken in a lagged receiver in which the exhaust gases are mixed and brought to rest. A thermocouple in an uncooled exhaust pipe immediately adjacent to the engine reads 150 to 200 F lower, due to failure to represent the mean gas temperature. Aside from this, it will be noted that the specific operating speed is high.

[9] P. H. Schweitzer, "Is the High-Speed Two-Cycle Diesel Possible?" *Power*, June 19, 1928.

Power Cost of Supercharging

Extra indicated power must be expended in the cylinder to obtain the increased air charge. Whether this power cost is excessive can easily be estimated from engine-indicator cards, without detailed theoretical comparison. Refer to Fig. 10, which is illustrative and not to scale. The shaded area A, determined by suction and compression curves, is a measure of the work expended to draw in and compress the charge, preparing it for combustion. The ratio of this work to the weight of charge is the desired criterion. A comparison was made between $L = 10$ and $L = 40$, resulting as follows:

Pipe length L, in.	10	40
Compression pressure, lb per sq in. abs.	425	425
Compression ratio	13.0	11.8
Volumetric efficiency, per cent	86.5	97.5
Air drawn in per cycle, lb	0.0286	0.0322
Mean indicated pressure of compression, lb per sq in.	47.0	51.2
In.-lb work per lb air	1,255,000	1,214,000

We may disregard the slight advantage shown with $L = 40$ and assume the work amounts equal. Then the extra air charge is obtained with proportionate power expenditure only. This is exactly as if the engine size were increased in the ratio of the air charges with equal quality of design.

Conclusions

The long suction pipe is an effective means of increasing volumetric efficiency and load capacity of an engine. The best results are obtained with high air velocities, which may approach one-half the velocity of sound. A compact and efficient installation is obtained with high ratios between cylinder area and pipe area; this ratio may with advantage exceed 10.

The increase of air charge may be as much as 19 per cent under favorable conditions, over what is obtainable when inertia is absent. The engine work expenditure to obtain the extra air is nearly proportional to the increase of charge, and therefore justifiable economically.

The supercharge effect will normally increase with speed, resulting in a specific load capacity which rises with engine rpm.

Engine performance reflects the advantages of the increased air supply. The characteristics become those of an engine larger by the ratio of the increase in air charge.

The theory as developed offers a satisfactory method of anticipating indicator cards, suction power loss, and volumetric efficiency. Theoretical efficiency must be corrected by about 10 per cent in the size of engine tested. In larger engines the correction will be smaller, and in small engines larger.

Appendix A

ALLOWANCE FOR PIPE FRICTION

The friction coefficient f in Equation [21] is an experimental function of the Reynolds number. From data such as that of McAdams[7] obtain f for the estimated maximum rate of flow. With sufficient accuracy, f so found may be used as a constant. Then for a given pipe diameter, Equation [21] takes the form:

$$\Delta p = \text{constant} \times Lw^2 \quad \ldots\ldots\ldots\ldots [21a]$$

This Δp acts at all times to retard the column, whatever the direction of motion. Equation [26] then becomes:

$$\frac{dw}{d\theta} = 24{,}800\,\frac{(p_a - p_v - \Delta p)}{Ln} \quad \ldots\ldots\ldots [26a]$$

while Equation [29] becomes:

$$\frac{dw}{d\theta} = \frac{1.072\,(p_a - p_v + \Delta p)}{L\rho_p n} \quad \ldots\ldots\ldots [29a]$$

The effect of heat of friction upon the density of air in the pipe is neglected (Assumption 2, case III). However, the valve conditions are changed by the resulting temperature rise. The amount of this rise must be estimated.

$$\text{Work} = \text{Force} \times \text{Distance}$$

or

$$W = \Delta p A_p w \text{ in-lb per sec}$$

This energy is added as heat to the air simultaneously flowing through.

$$G = A_p p_a w \text{ lb per sec}$$

The temperature rise is then:

$$\Delta T = \frac{W}{c_p G} = \frac{\Delta p A_p w}{c_p A_p p_a w}$$

$$= \frac{\Delta p}{c_p p_a} = \frac{23{,}100}{2245}\,\Delta p$$

$$= 10.25\,\Delta p$$

Specific heat c_p is used here, not c_v, for the reason already pointed out, that we are concerned with the amount of energy carried through the valve, and this is Gc_pT in-lb per lb (see Equation [4]). We must have equality of energy as well as of weight passing the valve section at any instant.

In view of Equation [21a], we may reduce the expression for ΔT in a given case to

$$\Delta T = \text{constant} \times w^2 \quad \ldots\ldots\ldots\ldots [21b]$$

The calculation procedure as given in case III is then modified to the following (data for point m are assumed to be known):

(1) Compute Δp (Equation [21a])

(2) Compute $\left(\frac{dw}{d\theta}\right)_m$ and w_n (Equation [26a])

(3) Compute $\left(\frac{dp}{d\theta}\right)_m$ and p_n (Equation [25]; T_{vn} will not equal T_a)

(4) Compute $(\Delta T)_n$ (Equation [21b]); then $T_a + \Delta T_n = T_{vn}$, neglecting $T_v = f(p_v)$, as before

(5) Compute p_{vn} (Equation [24]).

Discussion

K. J. DeJuhasz.[10] Using the intake and exhaust pipe as active means for increasing the cylinder charge is an attractive problem which, if successfully solved, would result in increased power output of the engine. The paper, containing a theoretical analysis and experimental procedure with several novel features, and also experimental results of promise, is a valuable contribution to the existing literature on this subject.

Some years ago the writer took part in indicating tests carried out by a large European company manufacturing motor cars and internal-combustion engines. The purpose of these tests was to secure the greatest possible power output from a given engine. On a supercharged racing-car engine with a speed up to 6000 rpm, modification of the valve timing and suitable shaping of the exhaust pipes resulted in a power increase of about 4 per cent. On a car engine of normal design, a noticeable increase of mean effective pressure was obtained by highly polishing the inside surface of the intake manifold and by eliminating the obstructions in it.

A survey of the improvements attained in latter years by

[10] Assistant Professor of Engineering Research, Pennsylvania State College, State College, Pa.

suitably dimensioning the inside of the combustion chamber leads one to the justifiable expectation that benefits could be realized also by close attention to the design of engine parts extraneous to the cylinder—i.e., to the intake and exhaust system. Applied to Diesel engines, such gain would be especially valuable as it would tend to compensate one of the disadvantages of the Diesel, the comparatively low mean effective pressure.

With the object of exploring the possibilities in this direction an investigation has been started at the Engineering Experiment Station of the Pennsylvania State College on the inertia charging of a two-stroke cycle engine.

The author performed a timely service in focusing attention to this possibility of bettering engine performance.

RALPH MILLER.[11] Has the author investigated the effect of elbows and bends in the suction line? It is impractical to build suction pipes straight as shown on the illustration. Can the same effect be produced where manifolds are used as on multiple-cylinder engines without unbalancing the degree of supercharging for each cylinder?

The investigation is applicable to compressor design, but of little value in Diesel-engine design when it is remembered that engines with the most highly developed combustion cannot be operated satisfactorily with mean effective pressures above 80 lb, and this limitation is not imposed by lack of air.

HARTE COOKE.[12] In connection with the arrangement for pressure charging given by the author, the power required of course comes from the engine, and as the arrangement mentioned by another discussor accomplishes the same thing with greater efficiency, it just means that the supercharging was done as a smaller expenditure of power on the engine itself.

In regard to the vibrations in the piping, these are of considerable magnitude and of a very complicated nature. The writer at one time observed a very marked fall in pressure in the high-pressure cylinder of a certain engine immediately after the beginning of the exhaust stroke, apparently due to the inertia of the steam going out of the cylinder. Indicator cards were taken from the exhaust pipe, and very peculiar vibrations of considerable magnitude were noted in the pressure in this pipe which showed that the pressure immediately after release, due to the puff of steam, was reduced considerably below the normal receiver-pressure for the first part of the exhaust stroke; also, these complicated vibrations were repeated perfectly over a period of a minute or so.

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[12] Mechanical Engineer, McIntosh & Seymour Corporation, Auburn, N. Y. Mem. A.S.M.E.

AUTHOR'S CLOSURE

The progress of Professor DeJuhasz's two-cycle experiments will be of much interest. In an ideal system, all scavenging power should be taken from the exhaust "puff." Here also a difficulty will be found in making the process effective over a wide speed range, and to initiate it at a low speed.

In reply to Mr. Miller, the introduction of elbows, if properly formed, does not greatly influence the results obtained. The radius of the bend should be large, i.e., equal to about two pipe diameters. An engine has been operated in which a 90-deg elbow was present in the pipe, adjacent to the head. Increase of compression was only slightly less than in the preliminary tests quoted in the paper. A similar effect can be produced where a manifold supplies, for example, a group of three cylinders. Such results were quoted by Capetti. However, the pipe resonance frequency is encountered at a much lower engine speed, because of the multiple impulses. At higher speeds the amount of supercharge decreases rapidly.

Because an engine does not use in combustion all air present in the cylinder, it does not follow that it will not benefit from a further addition. The supercharged engine development of Mr. Büchi, for example, would be impossible if a limit such as is mentioned by Mr. Miller existed; fortunately it does not. The limit of clear exhaust for the present test engine was raised from 85 to nearly 100 bmep by pipe supercharging. High-speed engine operation is in fact limited by the possibilities of dissipating heat from the combustion space, and in this respect a further surplus of air is plainly helpful. In the present tests this is reflected in a reduction of exhaust temperature.

Exhaust pipe waves, mentioned by Mr. Cooke, were in one case found to have an amplitude as high as 3 lb per sq in. These waves may be utilized to aid scavenging, but unfortunately the favorable effect is confined to a rather small speed range.

OGP-55-6

Some Phases of Light-Weight Diesel-Engine Design

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The paper analyzes the economics of light-weight Diesel-engine design. This is justified if these points are satisfied: Lower cost per unit (cost per horsepower), lower cost of operation for the unit, reduction in time of work accomplished, and reduction in engine size. The proper coordination of three principles will produce an engine very much under conventional weights, these being increased mean effective pressure, increased piston speeds, and intelligent use of material having a high ratio of strength to specific gravity.

THE introduction of this paper is a condensed analysis of the economics of light-weight design. The economic soundness of any project must be determined early in its conception to insure a lasting success. The recognition of the importance of the economic side of a venture prompts the use of a short discussion of its phases before starting with the design problems.

The light-weight Diesel engine is justified economically if one or more of the following premises are satisfied:

1 Lower cost per unit (cost per horsepower)
2 Lower cost of operation for the unit
3 Reduction in time of work accomplished
4 Reduction of engine size.

The almost axiomatic clearness of the first premise need not be enlarged upon. Methods that accomplish this lower cost per horsepower are discussed later in this paper.

The second item in the foregoing list needs some exposition. The use of Diesel power in mobile equipment has been retarded because of the weight of this type of engine. The cost of operation of mobile equipment is measured in units of cost per ton-mile. The engine in many of these units constitutes the major portion to the overall weight and therefore should receive first consideration in a weight-saving program. The close relation that the weight of the engine bears to the weight of the balance of the unit results in a unit which either is smaller and lighter or a unit capable of handling a higher percentage of pay load. In either case there is a saving in the cost of transportation.

Reduction in the time of work accomplished is frequently desirable, but in every instance requires more power. There are many types of equipment which have reached the limit of size or weight, and unless increases in power are made without increasing the weight of the power plant, the possibility of reducing the time for a given amount of work is impossible. Two excellent examples are found in the Diesel dredge and the Diesel-electric locomotive. The dredge is limited in size and draft by the operating conditions; this, in turn, determines the weight of the power plant. The premium paid for light weight in a case of this kind is sound economically and is generally recognized by both the operators and the builders of this equipment.

The Diesel-electric locomotive requires weight for tractive effort, but there is a limit of load per axle, as well as a limit for the number of wheels. These limits predetermine the weight of the power plant, and, by this token, the specific weight of an engine becomes a very important factor. The loads pulled and the speed attained are both essentially a direct function of available horsepower. It is the opinion of many engineers that the future of the Diesel-electric locomotive will depend on the development of engines of large output and low specific weight.

The fourth item, reduction of engine size, has a twofold economic interest. Space is frequently the determining factor for engine installations, and engines of high output that will go into a small space will find new markets as well as satisfy the demand for relatively small installation dimensions that are dictated by the size of the unit in which the engine is to be installed.

There are many examples that would amplify the foregoing discussion, which has just barely touched the high points. The subject would make a very interesting paper in itself. Due to the amount of material to be discussed under the heading of ways and means of designing the light-weight Diesel engine, no more time will be given the economic side of the picture.

Assuming that the foregoing has justified the light-weight Diesel engine from the economic point of view and that the engine builders are recognizing many, if not all, of the advantages of this type of engine, the next step in the procedure is the investigation of various methods to attain the desired result. The proper coordination of three general principles will produce an engine very much under conventional weights. These principles are:

1 Increased mean effective pressure
2 Increased piston speeds
3 Intelligent use of material having high strength-specific-gravity ratio.

Each of these methods has problems which must be fully recognized and in which definite provision must be made for the proper functioning of the parts involved. It is the prime object of this paper to discuss the problems and to offer a solution of them.

Increased Mean Effective Pressure

Increasing mean effective pressure involves problems that may be classified under two general headings, i.e.:

1 Combustion
2 Heat.

The first, combustion problems, are going to be given little consideration in this paper. It is sufficient to say that the research work that is being done at State College, combined with that of the N.A.A.C., as well as of private enterprises, is producing results that will ultimately be reflected in the industry by increased m.e.p. rating.

The second problem, that of heat, is a vital one with present m.e.p. ratings. When these ratings are increased, the difficulties encountered are magnified, and steps must be taken to avoid failure of the parts that are subjected to the intensified conditions.

Increases in the m.e.p. ratings, at the sacrifice of specific fuel consumption, may or may not be justified. Unquestionably, the present low specific fuel consumption places the Diesel en-

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gine in an enviable position as a prime mover. By that token, the aim should be higher rating, with the same or a lower figure for specific fuel consumption. Assuming, then, unchanged specific fuel consumption and heat balance, an increase in m.e.p. would increase the amount of heat to be dissipated in direct proportion. It is this extra amount of heat that will cause mechanical failures if due recognition of its presence is not considered and means for its dissipation provided.

The four essential parts of the cylinder unit that are subjected to the heat of combustion are:

1 Cylinder head
2 Cylinder liner
3 Exhaust valve
4 Piston.

Each will be discussed under a separate heading.

Cylinder Head

In considering a cylinder head from the standpoint of heat dissipation, several things are apparent. The temperature gradient, between the hot and cold sides of the cylinder head, is a function of the thermal conductivity of the material and the thickness of the section through which the heat must flow. Due to the restraint offered by the rigid attachment of the cylinder head to the cylinder, the temperature gradient in the bottom face of the head sets up stresses which, unfortunately, are increased if metal sections are increased. By this same token, a reduction in the metal thickness would reduce the thermal stresses. However, the maximum cylinder pressure must be considered, and therefore is a determining factor for this dimension.

TABLE 1 PHYSICAL PROPERTIES OF SEVERAL ALUMINUM ALLOYS

Alloy No.	Ultimate T. S.	Yield point	Elongation, per cent
Casting Alloys			
195-HT-4[a]	28,000–38,000	13,500	6–12
195-HT-10[a]	36,000–50,000	27,000	0–5
195-HT-16[a]	30,000–40,000	21,000	3–8
43	17,000–22,000	7,000	3–7
112	19,000–24,000	11,000	1–2½
Forging Alloy			
25 ST	55,000–63,000	30,000–40,000	16–25

[a] These casting alloys are heat-treated.

The selection of a material having a higher thermal conductivity offers a method for the reduction of thermal stresses. Aluminum, with a conductivity factor some 4½ times that of cast iron, thus recommends its use. Other requirements, however, must be considered. Referring to Table 1, it will be seen that the alloy specified as 195-HT-16 has physical properties equal if not better than cast iron. The Brinell hardness, however, is low, which will necessitate a special design for a valve seat. There are two methods of incorporating a valve seat being used at the present time. An aluminum-bronze ring (90 per cent Cu, 10 per cent Al) shrunk into a recess or counterbore provides a valve-seat surface having a Brinell hardness of approximately 225. This is not only being used in Diesel practise, but has been the practise in both water- and air-cooled aircraft engines. The latter is much more severe service due to the high temperatures of the air-cooled cylinder head (500 to 550 deg. fahr.). Aluminum bronze was chosen because of its high Brinell hardness, its higher coefficient of expansion (approximately 0.00001 in. per deg. fahr.), and a higher thermal conductivity.

At least one American Diesel-engine builder is using steel valve seats cast into the aluminum head and has experienced no trouble from loosening in service. This construction becomes a necessity when valves are so close together that it is impossible to get sufficient material around the individual ring or valve seat. There is no objection to this practise from the foundry standpoint. All available information would indicate that there is little to choose between the two constructions.

The problems of corrosion must be recognized, but with proper precautions does not preclude this material for use in salt water. Differentiating between corrosion and electrolysis, tests indicate that corrosion alone is not serious, and with the use of inhibitors and protective coatings, trouble should not be encountered. Electrolytic action, however, is a most serious problem, and it is essential that materials other than aluminum, which are in contact with the cooling water, should be insulated by nonmetallic gaskets. Stud holes should be "blind" tapped, and copper-based alloys should be avoided as far as possible in the system. Where bronze pumps and water piping are used, the connections to the aluminum parts of the engine should be made with rubber hose.

All water-jacketed aluminum parts, such as cylinder heads, cylinders, and exhaust manifolds, should be treated with a corrosion inhibitor, as well as a protective coating which consists of baked linseed oil. To avoid erosion, designs should be so made that the water entering the engine does not impinge directly against the aluminum surface. If these precautions are taken, little if any apprehension should be felt. With fresh water cooling, the system is generally "closed," and highly alkaline water may be treated, thus avoiding all corrosion.

Cylinder Liner

The heat dissipated through the cylinder liner is considerably less than through the cylinder head, which, in turn, results in lower thermal stresses. The relative simplicity of the liner also readily lends itself to the use of stronger materials, should high thermal stresses become apparent. Steel liners could be made much thinner than the conventional cast iron, thus reducing temperature gradient, and with the exception of cost differential, the steel liner seems to be entirely practical.

Exhaust Valve

A very prominent aircraft-engine designer measures engine performance by a comparison of horsepower per inch of exhaust-valve size. This is just an indication of the heat problems that have been experienced in his field where high m.e.p. and high speeds are encountered. If the light-weight Diesel engine is produced by the methods suggested earlier in this paper, the exhaust-valve problem will approach the conditions existing in present-day aircraft engines.

When it is realized that the amount of heat which is absorbed by a valve is a function of the cube of its diameter and that the outlet for that heat varies only directly with its dimensions, it becomes apparent that this particular point will require careful consideration. There are two outlets for the heat in an exhaust valve—through the stem and through the seat. In either case the heat must pass through to the cooling water, and by the same reasoning as was used earlier in regard to the cylinder head, a greater amount of heat will be passed with the aluminum head than with the conventional cast iron.

In automotive practise 35 to 45 carbon exhaust valves have replaced, with better results, the special heat-resisting steels, when aluminum heads were used in place of cast iron. Unfortunately, there were no quantitative figures of the actual temperatures of these valves, but a comparison of the appearance of each valve indicated very clearly a cooler valve in the aluminum head.

Most valve-stem guides to date have been made of cast iron, but it is hoped that in the near future data will be available to show that aluminum valve-stem guides are practical. If this

can be shown, there will be further opportunity of reducing exhaust-valve temperatures.

The use of dual exhaust valves will go far in solving this temperature problem and will undoubtedly be a necessity in engines of large bore. Serious consideration should be given to the use of exhaust valves smaller than inlet valves. Data are available which show that the losses in an engine with an exhaust-valve diameter one-half that of the inlet-valve diameter were negligible with piston speeds up to 2300 ft. per min. While these figures were found on a gasoline engine, it is reasonable to believe that results in the same general direction would obtain in Diesel-engine practise. The advantage would be twofold—cooler exhaust valves and larger inlet valves. The advantages of both are obvious.

The design of the water passages around the valve parts can be so arranged as to give the much-desired high cooling-water velocities at this point. The insulating effect of the more stagnant water is eliminated, as well as steam pockets and the formation of scale due to localized high temperatures. The temperature reduction in the valve is very apparent.

Cast-iron cylinder heads weigh from 6 to 9 lb per horsepower. It is perfectly practical to design an aluminum head that will weigh from $2^1/_2$ to 5 lb per horsepower, weight well worth saving, as it affects overall engine weight.

Piston

Every Diesel-engine piston, regardless of design or material, shows the undeniable evidence of high temperatures. This evidence ranges from discoloration to heavy deposits of carbon on the under side of the head in cast-iron pistons. The next step is the cracked head, which in practically all instances is the result of excessive thermal stresses. Heat balance tests show that approximately 30 per cent of the total heat supplied an engine finds its way into the cooling water. Parday, supported by others, makes the claim that some 12 per cent of this heat is dissipated through the piston head. (These figures relate only to uncooled pistons.) Actual piston designs based on these data prove by their successful operation the essential truth of these figures.

A large manufacturer of aluminum pistons has established the fact that the greatest portion of heat leaving the piston is conducted through the piston rings (between 70 and 80 per cent is thus accounted for). These figures have been verified by Taylor at the Massachusetts Institute of Technology. With these figures established, B. Isidin has evolved formulas for the design of aluminum-alloy pistons which have been used in designs of pistons which function well in enough different sizes to justify their use as a standard regardless of the material used.

The important points which are established in this paper are:

1 Head thickness is a direct function of the heat supplied with any given material
2 The allowable temperature gradient for cast-iron pistons is 350 deg. (This produces a temperature stress of approximately 15,000 lb per sq in.)
3 The allowable temperature gradient for an aluminum piston is 130 deg. (The resulting temperature stress is 7000 lb per sq in.)
4 The thermal conductivity of cast iron taken as 1 gives a comparative conductivity of aluminum of 4.4
5 The head thickness of a piston varies inversely as the conductivity factor (allowable temperature gradient × thermal conductivity).

From these figures it can be shown that a cast-iron piston should have a head thickness some 82 per cent greater than an aluminum piston for similar service to be within safe design factors. Mr. Isidin places the limit of uncooled aluminum pistons at some 300 bhp per piston. The accuracy of this prediction should be confirmed within the next six months.

The large amount of heat going through the piston rings demands that careful consideration be given to the design of the ring belt. Laboratory and actual experience has shown that the ring width has little effect on heat dissipation. The number of rings and the depth of the ring in the groove are the two factors which should be carefully determined. The minimum number may be determined by formulas, while the depth of ring is dictated by the physical characteristics of the ring material. This should be 0.040 per inch of cylinder bore, and rings so designed can be furnished by any ring manufacturer who uses a good quality of iron.

Good practise on ring widths is as follows: 5 to 7, $^{15}/_{16}$ to $^3/_{16}$ in.; 8 to 10, $^{15}/_{16}$ to $^1/_4$ in.; 11 to 13, $^{15}/_{16}$ to $^5/_{16}$ in.; 14 to 16, $^{15}/_{16}$ to $^3/_8$ in.; 17 to 21, $^{15}/_{16}$ to $^7/_{16}$ in.; 22 and larger, $^1/_2$ in.

The top or fire ring should be placed a reasonable distance from the top of the piston; 10 to 15 per cent of the diameter gives satisfactory results.

When it comes to the selection of the alloy for the Diesel piston, a number of requirements are prerequisite to satisfactory operation. Of first importance is the selection of a material having sufficiently high physical properties at the operating temperature of the piston to withstand the stresses involved.

In the casting alloy identified as 142-HT, it will be noted that the drop in ultimate tensile strength and yield point of this alloy at elevated temperature is among the lowest. Its Brinell hardness is high enough to give long life to ring grooves, and a last consideration, and one of no mean importance, is the absence of permanent growth. There are no other commercial alloys that combine all these factors, and the only compromise in the selection of this alloy is slightly lower Brinell hardness than the alloy identified as 122-HT. This alloy has permanent growth and lower physical properties at high temperature.

The subject of pistons will be again discussed briefly under the heading "Higher Piston Speeds," but for the sake of continuity one or two points of piston design will be discussed at this time, even though the increased heat of the higher m.e.p. and piston speeds do not affect, to any marked degree, the remaining design problems.

The question of piston clearance has probably retarded the general acceptance of aluminum pistons more than any one thing. Conclusions based on the knowledge of the relative coefficients of expansion of aluminum and iron alone would indicate that approximately twice as much clearance is required for aluminum pistons as for cast iron.

Experience has proved that this, in general, is not the case. It is customary with solid trunk-type aluminum pistons to grind the skirt taper so that when the piston attains a balanced temperature the sides of the skirt will be essentially parallel. With this practise the clearance above the pin is from 0.0012 to 0.0015 per inch, while the bottom of the skirt will run with from 0.0008 to 0.0012 per inch. This range is caused by ratings, piston-ring set-up, water temperatures, combustion characteristics, etc.

In a well-designed aluminum trunk-type piston, where the skirt wall is relatively thick and the stiffness rib which runs from one wristpin boss to the other is wide at its base, the resulting skirt stiffness has a marked effect on the ability of the piston to stay round under operating conditions. This inherent roundness provides nearly ideal conditions for lubrication and results in long life.

Due primarily to the difference in expansion of steel and aluminum, full floating wristpins are much to be preferred. There is no tendency of the piston to distort due to the influence of the

pin, and the proper fitting of the pin is more readily accomplished with this construction. The loading of wristpins should not exceed 2000 lb per sq in. There are several instances where that figure has been exceeded, but temporary failure of lubrication, distortion of the pin, or variations in the hardness of the piston pin are apt to cause excessive wear if not a definite failure.

Several methods of retaining the piston pin have been used. The plug construction that seems to have more to recommend it than any other is one that allows the use of a long pin, prevents oil from free communication to the cylinder wall, and has practically an indefinite life. The plugs should be made from the same material as the piston to avoid variations in expansions.

The stepped wristpin boss should be proportioned so that the unit pressures in the piston and rod are reduced materially on the expansion stroke, while the unit pressures caused by inertia do not exceed those due to expansion pressures. Lighter piston pins are also possible with this design.

The lubrication of aluminum pistons is essential because of inherent characteristics of the piston alloy. "Scuffing" is the term used to describe the results of using aluminum on a dry cast-iron or steel surface. The aluminum tears and is deposited on the other surface. The heat generated from the high friction causes expansion. This expansion in the case of a piston causes a seizure. The same conditions with cast iron would cause excessive wear, but only in extreme cases would a seizure result. It is not the volume or quality of lubrication, but it is the early establishment and maintenance of lubrication that is the essential requirement.

Increased Piston Speeds

The reduction of specific engine weight by increasing engine speed produces results in exactly the same order as increased b.m.e.p. The limiting factors come under four general headings:

1 Combustion
2 Volumetric losses
3 Bearings
4 Heat.

Combustion problems at higher rotative speeds apparently are more easily overcome than those encountered when trying to increase m.e.p. Several engines in this country are running at 1000 r.p.m. and over with good economy. This in itself would indicate that this point is not insurmountable. It is the opinion of several Diesel engineers that the limit of speed increase lies within the remaining three headings.

Volumetric Losses

The volumetric efficiency of any internal-combustion engine reaches a point where speed increases do not increase horsepower. Valve lift and timing influence this critical point. It is possible to get good volumetric efficiency with mean air velocities of 13,000 to 14,000 ft per min. This would require a valve lift of sufficient height to provide an area under the valve seat 25 to 30 per cent in excess of the clear diameter of the valve. With a single valve having a clear diameter one-third the cylinder diameter, the allowable piston speed would be one-ninth of 13,000, or some 1450 ft per min. Two inlet valves per cylinder would increase this allowable piston speed to some 2900 ft per min.

From this reasoning it would appear that volumetric losses will not stand in the way of increasing Diesel-engine speeds for some time to come.

Mechanical Problems

The mechanical problems encountered when increasing engine speed are almost entirely those of crankshaft main bearings and crankpin bearings. The limit of speed increase is determined by allowable PV (pressure-velocity) values. Unfortunately there is quite a range of opinion as to just what this figure should be. It is, however, generally conceded that the ratio of P to V in the factor materially affects the maximum value, as do such design characteristics as rigidity of bearings, oil pressure, oil temperature, the maximum temperature at the bearing surface, and the ratio of maximum to mean pressure.

Neglecting combustion pressures, the work done on the crankpin increases as the cube of the speed. (Actual loads increase as the speed squared, while the V, or velocity, factor is a direct increase.) The maximum combustion and compression pressures act against the inertia forces, resulting in lower maximum crankpin-bearing pressures. When combining the two into a resultant crankpin load, it is found that the PV values do not increase quite in the ratio of cube of the speed.

An analysis of several conventional four-cycle Diesel engines using cast-iron pistons and steel connecting rods and running at piston speeds of from 800 to 1200 ft per min showed that from 60 to 80 per cent of the total work done on the crankpin was the result of inertia and centrifugal force. This was determined by making a diagram the ordinates of which represent resultant crankpin pressures, while the abscissa is equally divided and represents crank angles. The compression and combustion pressures are additive to inertia and centrifugal forces, with the exception of a portion of the crank travel some 76 deg before and after top dead center. If the area of a diagram for centrifugal and inertia forces is taken for the four strokes less 76 deg at each end of the diagram and compared with the original diagram, the results will be in the order mentioned.

Inertia and centrifugal forces are a direct function of the weight of the parts, and a reduction of those weights is desirable if engine speeds are to be increased. The limit of this speed increase is dependent on possible weight saving in the piston and connecting rod and the allowable PV values.

Take a more or less specific case where the following conditions are assumed:

1 Present PV value not to be exceeded
2 Allowable weight reduction of present piston and rod 50 per cent
3 Present piston and rod account for 80 per cent crankpin work.

It may be estimated that the mean pressure has been reduced a total of 40 per cent. Having established a figure of 0.6 of the original mean pressure, it becomes apparent that the value of the velocity factor in the PV factor may be increased as $3\sqrt{1/6}$, or some 18 per cent. The inertia forces due to this increase in speed will not be sufficient to overcome compression and combustion pressures to the same extent as with the original heavy pistons and rods, but the increase in mean pressure due to this is so small that it may be neglected.

It is practical to design an aluminum piston that will weigh less than one-half the weight of a well-designed cast-iron piston; in fact, experience has shown that the weight of an aluminum piston with pin and rings actually is about one-half that of a cast-iron piston in the same condition. With connecting rods it is somewhat more difficult to predict the weight saving due to the wide range of stresses used in the design of steel rods. The comparison in weights of a well-designed steel and aluminum rod complete with bolts and bushings will show a weight saving of from 45 per cent to 35 per cent. With this reduction in the weight of the rod and piston, the foregoing reasoning appears to be correct.

Aluminum Connecting Rods

The prime object of using aluminum for connecting rods is of

course the saving in weight. This weight saving must not, however, be accomplished with sacrifices in factor of safety or lack of rigidity in either the wristpin or crankpin end of the rod. The low modulus of aluminum (10,000,000) requires a redesign of the bearing ends of the rod to provide the added material for stiffness. The stiffness of a beam is a direct function of the amount of inertia of the section and the modulus of elasticity of the material. The moment of inertia varies as the cube of the section, and the ratio of the elastic modulus of steel to aluminum being 3, the additional depth of the section to give equal rigidity is $3\sqrt{3}$, or 44 per cent. This section will weigh 49 per cent less than a steel section of equal rigidity. This increase in section reduces the stress due to bending more than 50 per cent, a feature in all-aluminum design which is extremely desirable from a safety factor standpoint.

The direct babbitting of aluminum connecting rods has many desirable features. Its use allows for an increase in the diameter of the crankpin equal to twice the thickness of a bronze- or steel-backed babbitt-lined shell, which materially increases the torsional rigidity of the shaft and reduces the unit bearing pressures to some extent.

The heat in a bearing is generated at the two rubbing surfaces and must be absorbed by the oil and the material making up the bearing. The removable shell-type bearing restricts heat flow through to the rod due to the high resistance of the joint between the rod and shell. An experiment several years ago revealed that some 20 times as much heat passed out of a direct-babbitted aluminum rod as with a steel rod with a separate shell. This figure was influenced to a large extent because of the high thermal conductivity of the aluminum rod. One objection to a direct-babbitted steel rod is the injury to the crankshaft should the babbitt melt out. This objection does not hold with aluminum due to the fairly good bearing characteristics of the alloy. Experiments have shown that aluminum connecting rods may be run on nitralloy crankshafts without babbitt.

The special technique required to babbitt aluminum rods is readily acquired. The only disadvantage in a direct-babbitted rod is the replacement feature, which should not exclude its use.

The material used for aluminum connecting rods is a wrought alloy having a minimum tensile strength of 55,000 lb per sq in., 17 to 25 per cent elongation, and a fatigue limit of 15,000 lb per sq in. in the heat-treated condition. It may be hand-forged or die-forged as the conditions require. The advantages to be gained are apparent and justify any increase in cost.

Valve Gear

Increased rotative speed imposes added problems in the design of valve gear. The weights of all of the moving parts must be reduced to a minimum in order to avoid heavy valve springs. This is particularly true with high lift valves. The careful design of cams will do much to reduce the stresses in the system.

Aluminum may be used for rocker arms and pushrods with savings in weight in the order of 50 per cent for these parts. Unfortunately, the temperature and wear conditions of the rest of the system preclude the use of light alloys anywhere else. One engine builder is using aluminum pushrods primarily to keep valve clearances uniform over a wide temperature range. This happens to be one of the few places where the high coefficient of expansion of aluminum is not bothersome.

Heat

The increase of heat input at higher rotative speeds is quite the same problem as that of higher b.m.e.p.'s. The points discussed earlier in the paper are applicable here and need not be further amplified.

Summarizing the advantages of obtaining light weight by increasing speed and b.m.e.p., it is very apparent that the changes in design and material specification will have relatively small effect on the overall cost of the engine. The reduction in specific weight is quite marked and may be estimated by the following reasoning. The use of aluminum for cylinder heads, pistons, connecting rods, and valve-gear parts will probably reduce the total engine weight 10 per cent. If the original engine was rated at 100 bhp at a piston speed of 1000 ft per min, and weighed 50 lb per hp, or a total of 5000 lb, the new weight would be 4500 lb. An increase in mean effective pressure of 15 per cent would increase the bhp to 115 at 1000 ft per min. The increase in speed of 18 per cent would further raise the bhp to 138 at 1180 ft per min; a specific weight of some 33 lb per bhp (4500 ÷ 136) results.

The size of the engine would not be changed in any way, and the resulting economy in space would be of value in many installations. The life of the engine should be just as long due to the adherence to similar design standards in each case.

The foregoing analysis of weight reductions by means of increasing the horsepower for cubic inch of piston displacement was based on very conservative figures, particularly in the case of the possible increase in speed. If this increase in output can be sold for the same price per horsepower as was received for the original engine, the profit obtained per engine would be very attractive. This fact, coupled with the reduction in installation dimensions, should result in a readily salable engine, which in the final analysis is the aim of every manufacturer and engineer.

Intelligent Use of Material Having High Strength-Specific-Gravity Ratio

The principle reason for the use of aluminum alloys for the parts discussed in the foregoing is that the peculiar properties of these alloys are preeminently suited to the application for which they are recommended. The saving in weight, as it affects total engine weight, is a second or third consideration. The balance of the discussion will be confined to the use of materials that will reduce the weight of the parts fabricated from the light alloys.

The outstanding and most generally recognized property of aluminum is its low specific gravity. It weighs 0.1 lb per cu in., as compared to 0.26 for cast iron, 0.28 for steel, and 0.31 for bronze. While these figures are very attractive, the low weight of aluminum alloys is only one of many physical characteristics that must be recognized when using this material in engine construction. The relatively low modulus of elasticity should be given careful consideration, particularly when replacing steel, whether in cast or forged form. For rigidity equal to steel, the moment of inertia must be increased three times. This increases sections approximately 30 per cent, with a resulting saving in weight of some 40 per cent and a reduction of stress of 56 per cent.

When determining sections of equal rigidity of cast iron the moment of inertia must be increased 50 per cent, which results in sections 10 per cent larger than the cast iron. The actual aluminum castings designed with these increased sections will be approximately 55 lb lighter than cast iron and have a stress factor 26 per cent less than the iron castings. It is well for the designer of aluminum parts to become thoroughly familiarized with these relating facts. Stress is laid on this point, because the majority of parts which may be made of aluminum are stressed by bending, and the rigidity of the design is the most essential consideration. Invariably, when parts are sufficiently rigid, the stress is so low as to eliminate any need for anxiety.

There are two general methods of fabricating aluminum Diesel-engine parts—casting and forging. There are a large variety of

commercially available casting alloys, each suitable to a particular purpose. They in turn are divided into two groups and may be classified as heat-treated and un-heat-treated castings. The heat-treated castings are used where high physical properties are necessary. The treatment may be varied to give great ductility at the sacrifice of ultimate strength and yield point or low ductility with a marked gain in both yield point and tensile strength. Table 1 shows the properties of an alloy designated as 195. The figures following the HT designate the treatment that varies the physical properties as soon in the list. This particular alloy has a very wide use in the mechanical field, the technique of casting it is highly developed, and its use to replace cast ferrous materials entails very little of an experimental nature.

There are several other available heat-treated alloys which have special physical characteristics, such as high Brinell hardness, good physical properties at high temperatures, etc. Their application, however, is limited to very few parts, such as piston castings and bearings, and in special cases they are used for particularly difficult casting to help the foundry.

The un-heat-treated alloys are used for parts which carry light loads. The nature of the casting as well as the work it has to do determines the selection. Alloy 43 is used where oil-tightness is a requirement. It casts well, and extremely thin castings may be made from it. A casting 6 ft sq can be made with a metal thickness of $^1/_4$ in. if there are no heavy sections. Its machine characteristics, however, are not as good as the heat-treated alloys nor the alloy designated as 112.

No. 112 alloy is the general-utility material for large and small castings which require little shock resistance. It is easy to cast, machines well, and has had many years of use. The physical properties are very near those of ordinary cast iron.

The use of forgings in Diesel construction is more or less limited to connecting rods and valve rocker arms. The Diesel aircraft piston might be added to that list, but the development of the art for this type of forging is quite new. As stated in another section of this paper forgings either may be hand forged or die forged, depending on the economics of quantities, size, etc.

One very attractive feature of reducing the specific weight of an engine by substituting aluminum for iron and steel is that bearing sizes, cylinder centers, and in fact all the important characteristics of the engine remain unchanged. The engine builder who has built a successful engine can reduce its specific weight a material amount without going through a long stage of development. Several builders are making both aluminum and cast iron parts from the same pattern equipment. The saving in weight when this is done is approximately 62 per cent.

From 70 to 80 per cent of the total weight of a Diesel engine is available for aluminumization, and when it is realized, that saving will average some 55 per cent. It is apparent that this method of attack should be given serious consideration. The reduction in specific weight by this method is something better than 40 per cent. Its only drawback is that it increases the cost of the engine, but it must not be forgotten that low engine weight has a very definite economic appeal in many fields. Experience has shown that the cost of reducing weight by this process is approximately 25 cents per pound saved.

In conclusion, it is of interest to estimate the total saving in engine weight possible by using the three methods discussed in the foregoing. Going back to the specific example of the 100-hp engine which weighed 500 lb as built in the more conventional materials the total saving in weight by using aluminum is 2000 lb, resulting in a weight of 3000 for the light engine. The horsepower increase was estimated as 36, which when added to the original power results in a total of 136 hp. The weight per horsepower obtained from these figures is almost exactly 22 lb, a saving of 56 per cent.

From the cost standpoint, the saving of 2000 lb at 25 cents a pound would increase the cost of the engine $500. If the original engine cost would be $5500, dividing that figure by the new horsepower rating, the resulting cost per horsepower will be $40.50. This represents a saving of 19 per cent. This weight and cost analysis, if found correct in practise, should satisfy the economic phase of the light-weight engine. Experience in actual applications of all the various parts of the engine which have been discussed has proved by their successful operation, the mechanical soundness of those parts.

It is hoped that this paper will in some small way stimulate and encourage the Diesel-engine industry to produce light weight engines that will open new and profitable markets.

OGP-55-7

Spray Characteristics of a Jerk-Pump Fuel-Injection System

By O. F. ZAHN, Jr.,[1] STATE COLLEGE, PA.

Tests were conducted with special spray-research apparatus in the Oil Spray Laboratory at Pennsylvania State College, aided by the loan of jerk-pump injection equipment. The application of the conclusions reached is summarized as follows: Variables that could be changed in engine installation are speed, load, and nozzle-opening pressure. These would affect the combustion line on the indicator card only as they changed temperature, turbulence, and spray characteristics. Within the range tested of these operating variables, only opening pressure would change spray characteristics enough to produce a notable effect on the combustion. It would also limit the minimum load that the engine could carry. The effects of speed and load on the spray would be negligible compared to their effects on temperature and turbulence. Combustion would be practically free from the effects of changing spray characteristics with a constant nozzle-opening pressure.



THE special spray-research apparatus of the Oil Spray Laboratory at the Pennsylvania State College and the loan of jerk-pump injection equipment through the courtesy of the United American Bosch Corporation made possible these experiments. Acknowledgments are due to these two institutions and their personnel for assistance and supervision. The laboratory apparatus, shown schematically in Fig. 1, is fully described in References 1 and 2. The injection equipment was made up of the following standard commercial units:

Fuel Pump. This was a constant-stroke, single-cylinder unit of the general design shown in Figs. 2 and 3. Oil enters the plunger space through a cylinder port and discharges through a spring-loaded delivery valve. Metering is effected by a channel cut in the plunger overrunning the inlet port. Cut-off can be adjusted to take place earlier or later by rotating the plunger with the metering rack, according to the load.

The manufacturer's specifications give the maximum pump discharge as 350 cu mm (0.0214 cu in.), which would mean 0.30 g when using oil of 0.864 specific gravity. The maximum discharge corresponds to about 100 per cent overload of a suitable engine, and never would be used. A discharge of 0.15 g with 40 per cent excess air corresponds to a cylinder size of about $5^1/_2$ in. by 7 in. Further specifications are: Plunger diameter, 9 mm, or 0.354 in.; plunger stroke, 10 mm, or 0.394 in.

Oil Line. This was a steel tube connecting pump and nozzle. The length was 36 in.; inside diameter, 0.10 in.; outside diameter, 0.25 in.

Fuel Nozzle. The nozzle, shown mounted for spray tests in Fig. 4, was of the spring-loaded differential-valve-stem type. The valve carries a small pintle which projects through the orifice. This is shown in Fig. 5. Oil issues through the annular space between pintle and orifice walls. The pressure at which the valve opens is adjusted by turning a bolt compressing the spring. Through the bolt and spring passes a small feeling pin which may be used to feel the lifting of the valve. The valve diameter was 5 mm, or 0.1965 in.; orifice diameter, 1.094 mm, 0.0431 in.; pintle diameter, 1.043 mm, 0.0411 in.; opening pressure, adjustable, 0 to 5000 lb per sq in.

Test Conditions. Table 1 is a comparison of conditions under which solid-injection equipment may operate in engines in one case and with the test apparatus of the Oil Spray Laboratory in the other case, for the three main spray tests: rate of discharge, velocity of spray, and dispersion of spray.

Oil. The oil used was Diesel fuel oil having 0.864 specific gravity and 48 sec Saybolt viscosity at 79 F.

Miscellaneous Tests

Before the spray tests proper, tests were made on leakage, line pressure, temperature rise, and metering.

Leakage and Residual Pressure. The nozzle was connected to a hand pump, and the leakage was noted. When the nozzle opening pressure was set at 900 lb per sq in., leakage was observed at all pressures down to about 300 lb per sq in., but was so slight below 400 lb per sq in. that it took some time to appear. When the nozzle opening pressure was raised to 4000 lb per sq in., the leakage became similarly slight at a pressure of about 3500 lb per sq in.

The nozzle was then connected to the jerk pump and oil line

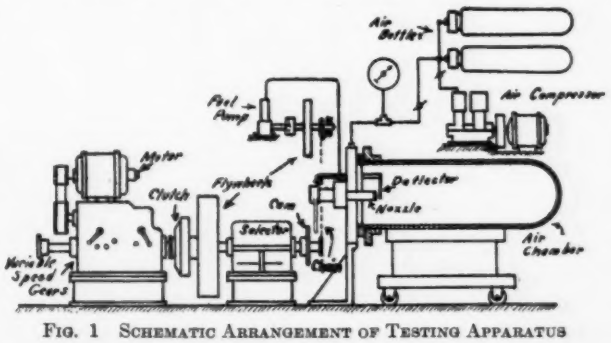


Fig. 1 Schematic Arrangement of Testing Apparatus

described, and a spring-balanced-diaphragm electrically recording pressure indicator (Ref. 3) was placed in the line at a distance of 10 in. from the nozzle. The position of the pump shaft where approximately 500 lb per sq in. started and stopped was recorded. It was found that this pressure existed for 360 deg for the range of speeds tested, 131 to 506 rpm, with a nozzle-opening pressure of 900 lb per sq in. This means that while the pump is running, the residual line pressure does not drop below approximately 500 lb per sq in.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

While thus normally operating, the equipment showed no leakage whatever between sprays, indicating that with this residual pressure leakage was so slight that it did not have time between successive injections to make its appearance.

Temperature Rise. Due to "working" of the oil, the issuing spray will be hotter than the oil in the pump-supply reservoir.

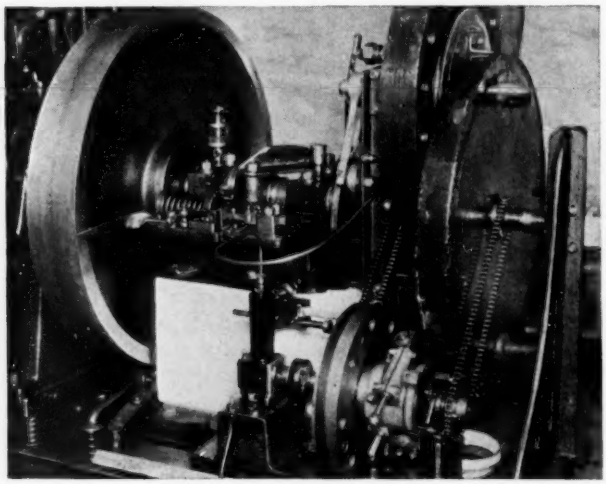


FIG. 2 MOUNTING OF INJECTION EQUIPMENT
(In this picture the spray stroboscope is also set up for rate of discharge tests.)

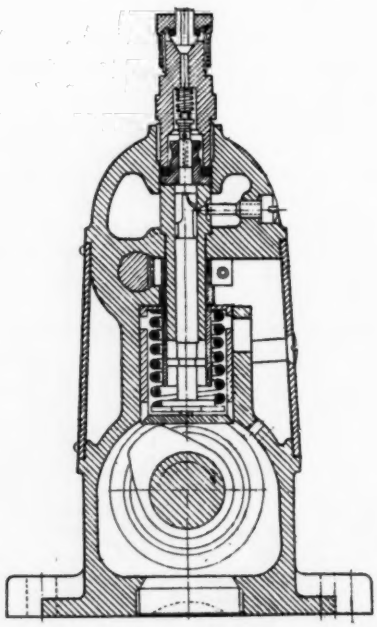


FIG. 3 TYPE OF PUMP TESTED

TABLE 1

Comparison	Spray test	Laboratory conditions	Engine conditions
Injection equipment	All	Identical	Identical
Pump speed	All	50–800 rpm	25–1000 rpm
Fuel oil	All	Identical	Identical
Air temperature	All	80 F	1500–2500 F
Air pressure, Air density	Dispersion	0–300 lb per sq in.; 0.073–1.46 lb per cu ft	450–850 lb. per sq in.; 0.95–1.17 lb per cu ft
Air pressure, Air density	Velocity	0–300 lb per sq in; 0.073–1.46 lb per cu ft	450–850 lb per sq in.; 0.95–1.17 lb per cu ft
Air pressure, Air density	Discharge rate	0 lb per sq in. (gage); 0.073 lb per cu ft	450–850 lb per sq in.; 0.95–1.17 lb per cu ft
Air turbulence	All	Not present	Present
Combustion	All	Not present	Present
Vibration	All	Less	More

The temperature rise was found to be independent of the amount of discharge, but it varied with speed, as is shown in Table 2.

TABLE 2

Speed, rpm	Reservoir temperature, F	Spray temperature, F	Temperature rise, deg
131	76	82½	6½
202	76	83	7
314	76	84	8
506	76	86	10

At higher speeds more energy is imparted in a given time; hence the oil temperatures are higher. At 506 rpm, this rise changes the viscosity of the oil from 49 to 46 sec Saybolt.

Metering. The effect of the air pressure at the nozzle and the speed of the pump on the maximum discharge were next examined. Five injections were caught in a cotton-filled cup and weighed. The averaged results are given in Table 3.

TABLE 3

Air pressure, lb per sq in.	Pump speed, rpm	Fuel discharge, grams
0	131	0.2874
	202	0.2870
	202	0.2887
	314	0.2788
	314	0.2877
	506	0.2941
Average		0.2873
200	131	0.2869
	202	0.2784
	202	0.2823
	314	0.2806
	506	0.2785
Average		0.2813
Mean of averages		0.2843

The position of the pump-throttle rack which rotates the plunger and meters the discharge was tested for partial discharges. The results are plotted in Fig. 6. "Metering rack displacement" is the distance from dead-end position, and "Discharge" is an average of several injections. At high nozzle-opening pressures there is a practical limit of minimum discharge because of missing. An injection does not take place at every revolution of the pump when the amount of discharge is too small to properly overcome the compressibility of the oil. This region is indicated by the dotted portions of the curves.

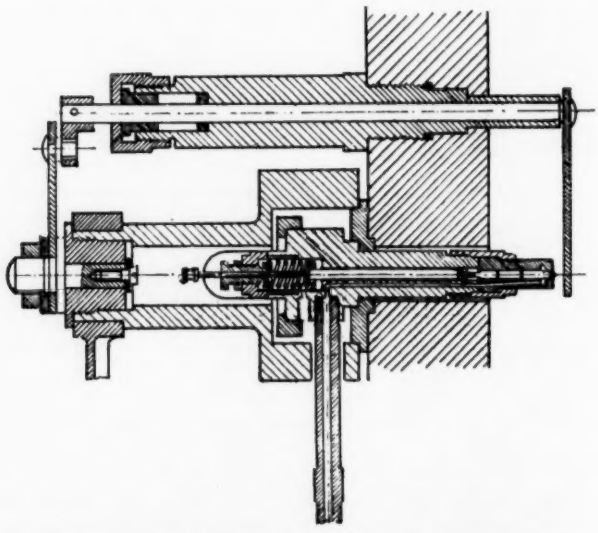


FIG. 4 NOZZLE IN MOUNTING WITH SPRAY DEFLECTOR ARM

At higher opening pressures the atomization of the spray is much finer. The best accuracy of metering was observed at an opening pressure of 2500 lb per sq in., but seems independent of speed or amount of discharge.

In the ensuing spray tests the percentages of discharge signify the following amounts of discharge:

100 per cent discharge	0.28 gram
75 per cent discharge	0.21 gram
50 per cent discharge	0.14 gram
25 per cent discharge	0.07 gram

Rate of Discharge of Spray

The rate of discharge of the injection, in grams per degree of pump-shaft rotation, was found with the spray stroboscope (Ref. 1). This apparatus consists of a disk having a small slot 2 deg wide, which revolves 1 in. in front of the nozzle. The disk is enclosed by a guard casing holding a cup (see Fig. 2). A part of the spray passes through the slot into the cup, but the remainder is stopped and drains away. The disk was chain driven at three times the pump speed through a differential gearing which permitted its phase relationship with the pump to be changed while rotating. Tests were made at atmospheric air pressure, at 131 rpm pump speed, and at 25, 50, 75, and 100 per cent discharges.

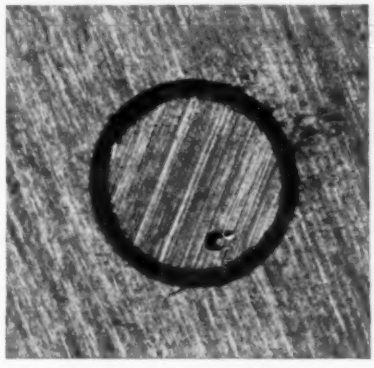


Fig. 5 Annular Orifice of the Nozzle Tested

[The annulus has an outer diameter of 0.0431 in. The inner wall is formed by the pintle (0.0411 in. dia) in the center.]

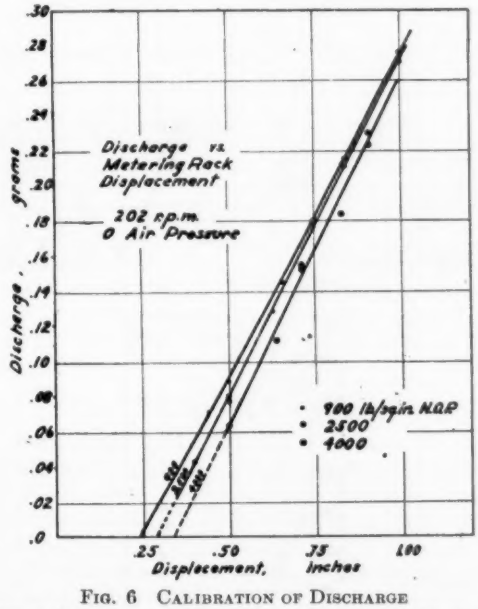


Fig. 6 Calibration of Discharge

(Metering rack displacement is plotted against the weight of discharge at three settings of the nozzle-opening pressure: 900, 2500, and 4000 lb per sq in. The dotted portions indicate the lower limits of weight of discharge.)

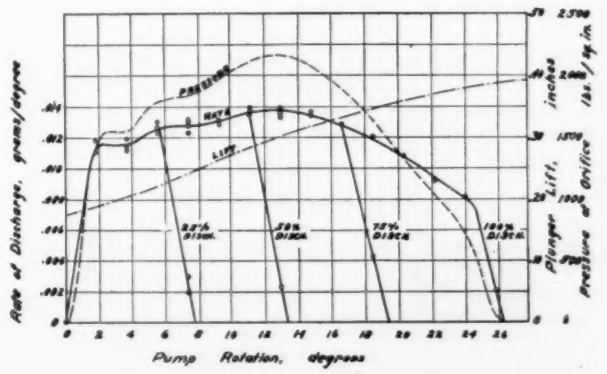


Fig. 7 Rate of Discharge

[The rate, in grams per degree of pump rotation, is plotted against pump (camshaft) rotation in degrees. The total discharges are 100% (0.28 gram), 75% (0.21 gram), 50% (0.14 gram), and 25% (0.07 gram). Speed, 131 rpm. Air pressure, 0 lb per sq in. gage. Calculated pressures at the orifice and the plunger lift have also been plotted.]

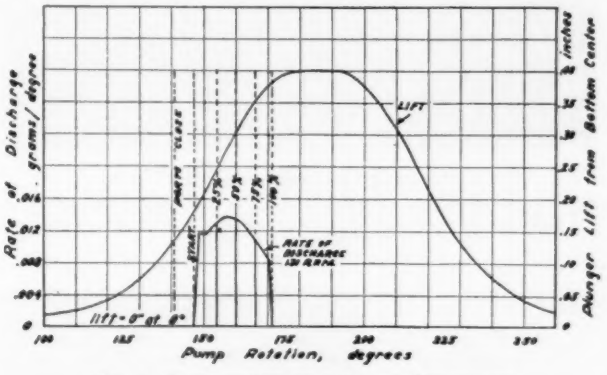


Fig. 8 Plunger Lift and Rate of Discharge

(The rate curve of Fig. 7 is here on a different scale, showing the relation of the injection period to the plunger lift and the closing of the pump ports.)

The results are plotted in Figs. 7 and 8 and show sharp beginning and cut-off of the spray. Cut-off is sharp, particularly for discharges lower than 75 per cent. No tendency to leakage is shown by the curves. Injection always begins at the same angle of pump cam, irrespective of weight or speed. The areas under the rate curves represent the weights of discharge, and upon measurement gave 0.287 g, 0.216 g, 0.139 g, and 0.068 g, at 100, 75, 50, and 25 per cent discharges, respectively, which agree within the limits of accuracy with the metering tests. The mean rate of discharge is highest at 75 per cent discharge and lowest at 25 per cent discharge, but is fairly uniform over the whole injection period.

The period of injection is nearly directly proportional to the weight of discharge, up to 75 per cent discharge.

Assuming a coefficient of discharge of 0.8 and a constant orifice area, the oil pressures back of the orifice were calculated from the rate curve and plotted in Fig. 7. The nozzle-opening pressure was 900 lb per sq in., but the maximum calculated pressure is over 2100 lb per sq in.

The lift of the plunger, obtained with a dial micrometer, is plotted in Figs. 7 and 8 to show the relative position of the cam during the injection period.

Tests at 202 rpm, not included here, gave curves very similar to the tests at 131 rpm, except that the peaks in the early part of injection were amplified.

Velocity of Spray

The velocity of the spray was found by a ballistic pendulum (Ref. 2) shown in Fig. 9. Spray impinges on the 6-in. plate and the momentum mv is recorded by a pencil line on the drum. Knowing the mass of the spray, the mean velocity was determined by

$$v = 1/m\ (MV + K)$$

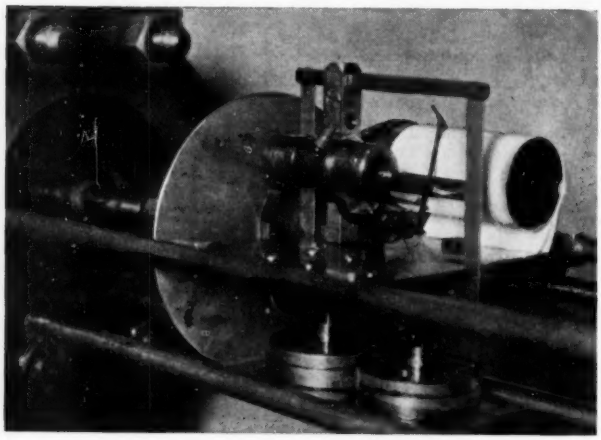


Fig. 9 Ballistic Recording Pendulum

[The spray momentum is found at various distances from the nozzle. The deflection of the 6-in. disk under the impact of the spray (from left) is recorded on the drum.]

where

- v = mean spray velocity
- m = total spray mass
- M = mass of pendulum + spray
- V = velocity of pendulum after impact
- K = frictional correction.

K was experimentally found to be 10.4 gram-meters. Unfortunately, the corrections for velocity become comparatively large when using spray masses as small as 0.28 g and 0.21 g, and introduce an uncertainty in the absolute values of spray velocity. The pendulum was designed for much greater weights of discharge.

The mean velocities were found with the pendulum at increasing distances from the nozzle in atmospheric and dense air, and plotted in Fig. 10. The issue velocities vary between 80 and 130 m per sec (262 and 426 ft per sec). In atmospheric air the velocities drop but slightly in 14 in., while in air of 200 lb per sq in. gage, corresponding to the density of air compressed 14 to 1, the drop is much greater. In both cases the curve is close to a straight line. In some tests in atmospheric air the recorded velocity is greater, not at the nozzle as would be ex-

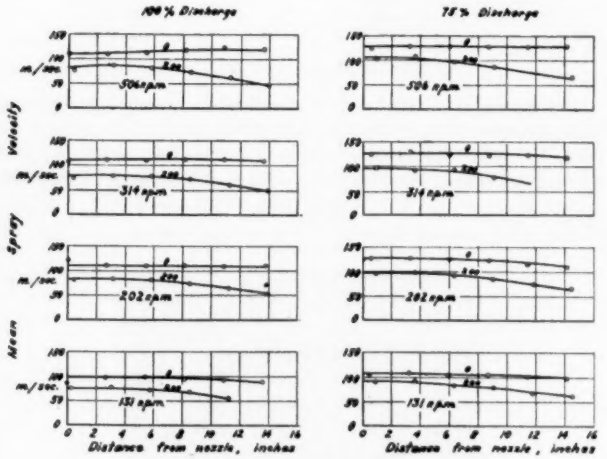


Fig. 10 Spray Velocity

[The mean velocity of the spray in meters per second is plotted against distance from the nozzle in inches. Air pressure, 0 and 200 lb per sq in. gage. Pump speed, 506, 314, 202, and 131 rpm. Weight of discharge, 100% (0.28 gram) and 75% (0.21 gram).]

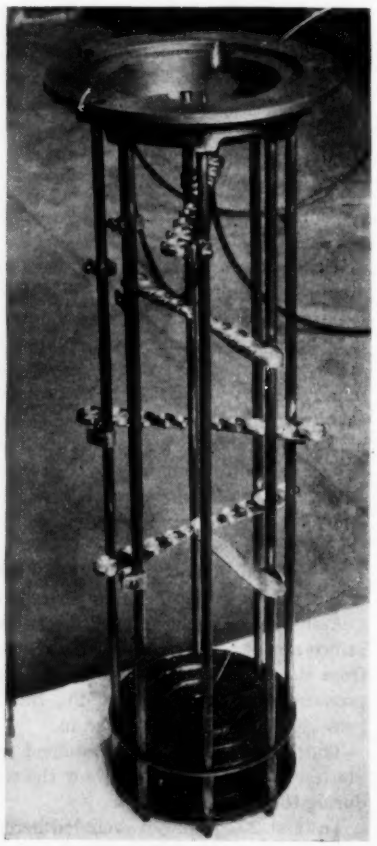


Fig. 11 Dispersion Rack

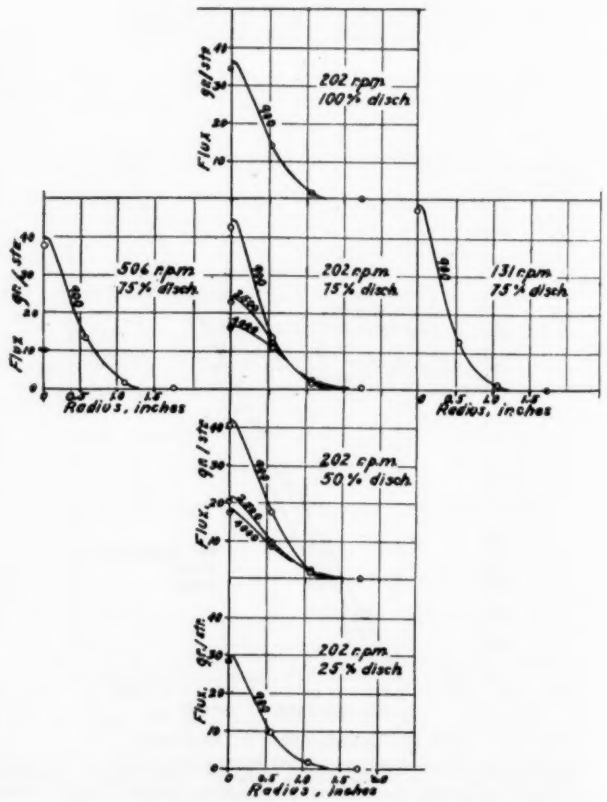


Fig. 12 Specific Flux of Spray

[Flux, in grams per solid angle (steradian), is plotted against radius from axis of spray at 7 in. from nozzle. Air pressure, 200 lb per sq in. gage. Pump speed, 506, 314, 202, and 131 rpm. Nozzle opening pressure, 900, 2500, and 4000 lb per sq in. Weight of discharge, 100% (0.28 gram), 75% (0.21 gram), 50% (0.14 gram), and 25% (0.07 gram).]

pected, but at some distance from it. This is obviously an error, caused by variation in air movement or elastic impact when the pendulum plate is too close to the nozzle.

The velocities at 75 per cent discharge are slightly higher than at 100 per cent discharge. This may be attributed to the fact, seen in Fig. 7, that the average rate of discharge at 75 per cent discharge is higher than at 100 per cent, and to possible errors in the calibration factor K. Otherwise, these curves show about what would be expected, i.e., negative acceleration becomes greater with increased distance and air pressure.

The spray velocities seem unaffected by pump speeds except at 131 rpm, at which speed all velocities are slightly lower.

It is to be borne in mind that the pendulum records the mean value of velocity; each oil particle is traveling at a different speed, ranging from zero to much greater than that recorded. Where the mean velocity is zero, however, it means that all of the oil has zero velocity; hence, it is the maximum penetration of the fuel spray. This would be at the intersection of the velocity curve with the "distance" axis.

In most actual engines it is certain that with this nozzle and setting and fuel oil, the spray would hit the walls of the combustion space. Ignition lag is considered very short if it is not more than 0.002 sec. In that time the spray would have traveled $0.002 \times 100 = 0.2$ m, or a distance of about 8 in.

At higher nozzle pressures the discharge velocity will be even higher at the nozzle, but due to finer atomization and consequent increased air resistance, it no doubt falls off more rapidly with distance.

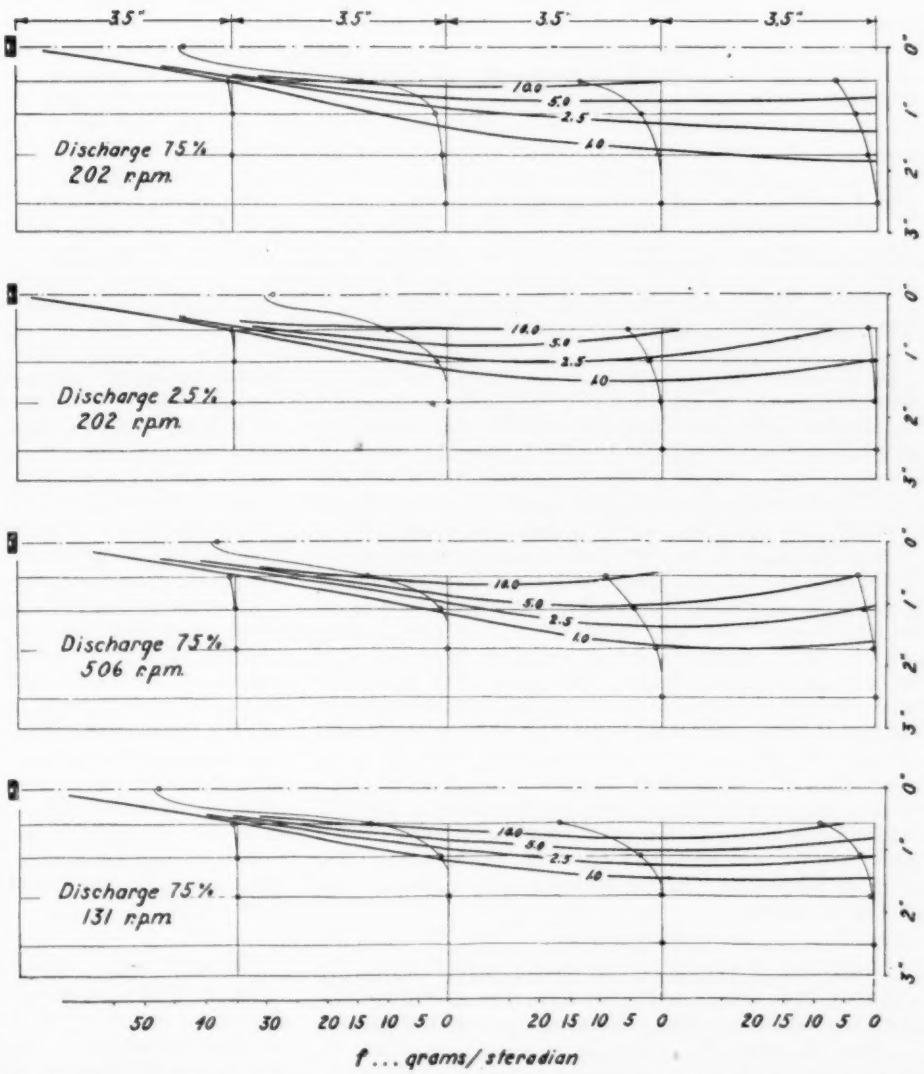


FIG. 13 SPECIFIC FLUX OF SPRAY AT SEVERAL DISTANCES, CONNECTED BY LINES OF CONSTANT FLUX

[Flux, in grams per solid angle, is plotted against radius from spray axis at 3.5, 7, 10.5, and 14 in. from nozzle. Lines of constant flux of 10, 5, 2.5, and 1 gram per solid angle connect the four flux curves. Weight of discharge, 75% (0.21 gram) and 25% (0.07 gram). Speed, 506, 202, and 314 rpm.]

DISPERSION OF SPRAY

The shape of the spray was determined with a "dispersion rack" (Ref. 2) which held small blotting pads in various positions in front of the nozzle. Two arms, each holding five pads, were placed on the rack at 3.5, 7, 10.5, and 14 in. from the nozzle, as shown in Fig. 11. For some tests, four of these arms and a fifth arm holding one pad in the center line of the spray were placed 7 in. from the nozzle.

Before each test, the axis of the rack was made coincident with that of the spray by the aid of a preliminary injection in atmospheric air. Then each pad was weighed and placed in position on the rack. The chamber was closed and filled to the desired air-pressure, and one or more injections were made. After opening the chamber, the pads were immediately weighed again. The difference in weight divided by the number of injections is the amount of oil which has been intercepted by the pad.

The weights of intercepted oil from several duplicate tests were added and divided by the total weight of discharge, giving a weight corresponding to 1 g of discharge. This specific weight, divided by the area of the pad, gives the density in grams per square inch. Dividing the density by the square of the distance of the pad from the nozzle gives a density in grams per unit solid angle, termed hereafter "specific flux."

Fig. 12 shows the distribution of spray flux (grams per steradian) at 7 in. from the nozzle. Specific flux is plotted against radius, or distance in inches from the spray axis. Lines having the properties of probability curves (Ref. 4) have been drawn through the points.

Four of these curves are again shown in Fig. 13, with similar ones taken at other distances from the nozzle.

Heavy lines connect the locations having flux values of 10, 5, $2^1/_2$, and 1 g per steradian, giving a picture of the longitudinal distribution of the spray in space.

TABLE 4 EFFECTS OF TEST VARIABLES ON INJECTION CHARACTERISTICS

	Increasing test variables					
Injection characteristics	Weight of discharge	Pump speed	Air pressure	Nozzle-opening pressure	Distance from nozzle	Remarks
Metering		None	None	Decreases	.....	See "Metering"
Mean rate of discharge	Increases slightly up to 75%	None	No test	No test	.....	See "Rate of discharge"
Start of discharge	None	None	No test	No test	.....	See "Rate of discharge"
Duration of discharge	Increases proportional to 75%	None	No test	No test	.....	See "Rate of discharge"
Velocity of spray	Slight	Decreases below 131 rpm	Decreases greatly	No test	Decreases	See "Velocity of spray"
Cone angle of spray	None	Increases slightly	Increases greatly	Increases greatly	.....	See "Dispersion of spray," Fig. 13
Portion of spray traveling 7 in.	Depends on N.O.P.	None	Decreases greatly	Depends on discharge	.....	See "Dispersion of spray"
Atomization, fineness of	Slight, if any	Slight, if any	No test	Increases greatly	Increases	Observed visually
Temperature rise of oil	None	Increases	No test	No test	.....	See "Temp. rise"

Fig. 12 shows that the nozzle-opening pressure is the most influential variable on dispersion. Increasing the opening pressure flattens out the flux curves (increases the cone angle of the spray). Speed has practically no effect. Weight of discharge has some effect. Analysis of Fig. 12 indicates that at 202 rpm a much smaller portion of the spray travels 7 in. at 25 per cent discharge than at higher discharges with the nozzle opening pressure 900 lb per sq in. At nozzle-opening pressures of 2500 and 4000 lb per sq in., this difference is noted between 50 per cent and 75 per cent discharges. The explanation of this lies in the air resistance offered to the spray; with a given opening pressure, the atomization and cone angle, and hence the total air resistance, remain constant. Decreasing the amount of issuing spray therefore increases the unit resistance encountered, and less of the spray travels a given distance. Increasing the opening pressure increases both atomization and cone angle. Hence the total air resistance increases, and the effect of decreasing the amount of issuing spray just mentioned is amplified.

Tests made in atmospheric air showed that the spray had a much narrower cone angle and penetrated much farther. The increased penetration is shown by the velocity curves at 0 air pressure in Fig. 10.

Conclusions

For the sake of conciseness, the conclusions are summarized in Table 4. The headings at the top of the table are the variables of the test conditions, while those at the side are the observed injection characteristics. The body of the table contains the effects of the test variables, considered as increasing, on the injection characteristics. For example, increasing "Nozzle opening pressure" increases greatly the "Cone angle of spray."

Application of Conclusions

The variables that could be changed in an engine installation are speed, load, and nozzle-opening pressure. These would affect the combustion line of the indicator card only as they changed temperature, turbulence, and spray characteristics. Within the range tested of these operating variables, only opening pressure would change spray characteristics enough to produce a noticeable effect on the combustion. It would also limit the minimum load that the engine could carry. The effects of speed and load on the spray would be negligible compared to their effects on temperature and turbulence. Combustion would be practically free from the effects of changing spray characteristics with a constant nozzle-opening pressure.

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3 P. H. Schweitzer, "Oil-Spray Research at Penn State," paper presented at the Second Oil Power Conference at Pennsylvania State College, 1928. Pennsylvania State College Technical Bulletin No. 5, pp. 175–187.

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Discussion

E. T. Larkin.[2] The paper is an interesting disclosure of spray characteristics with certain equipment. There have been many other papers presented by other investigators over a period of years regarding various spray investigations, and as one interested in engines, the writer has often wondered how this data can be actually applied to engines and combustion-chamber design in combination with the phenomena of combustion.

Investigations of spray characteristics are only a preliminary step in the determination of oil-engine performance. Combustion is influenced by the air conditions, principally those of temperature and air motion in combination with the shape of the air envelope surrounding the spray or sprays. The air made available to that part of the spray where combustion is first started may cause an acceleration of spray velocity, or a counterflow opposing the spray movement may be formed of sufficient velocity to substantially counteract the movement of a considerable portion of the fuel. It is suggested that more advanced work correlating spray characteristics with combustion characteristics offers a very desirable field of investigation. The work up to date of spray analysis forms a very desirable background for the research which must ultimately be performed to demonstrate the possibilities of developing and controlling high efficiency in oil-engine combustion.

Edgar J. Kates.[3] The writer believes that this paper has much practical value, despite the fact that the tests were conducted in a research laboratory instead of in an actual engine. The tests showed, for one particular make of fuel pump in rather general use, the effect of variations in pump speed, fuel quantity, and nozzle-opening pressure upon such important spray characteristics as velocity of spray, cone angle of spray, and duration of discharge. Many of the conclusions, such as that spray velocity is practically independent of pump speed, are full of meaning for an engine designer and are quite valid, even though there was no combustion or air turbulence.

In actual engines there are so many variables peculiar to the engine design that fuel pump tests on any one engine cannot be used as a basis for predicting results obtainable on another engine design. In the author's tests, these extraneous variables were eliminated by testing upon an apparatus where they did not exist; this now enables an engine designer to make whatever modifications his own engine type entails.

The writer hopes that this valuable research will prove to be the forerunner of others on other makes of fuel pumps, so that engine designers may be able to choose and adapt their fuel pumps to their engines intelligently.

Edward Adams Richardson.[4] The question has been raised regarding the value of tests such as these to the engineer. Exact

[2] Chief Engineer, Sterling Engine Company, Buffalo, N. Y.
[3] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.
[4] Bethlehem, Pa. Jun. A.S.M.E.

information regarding performance characteristics is of great importance. However, the engineer is interested not merely in the characteristics of a piece of equipment when new and in perfect condition. The precision of the performance of the pump system tested by the author and here reported is rather amazing. The writer would like to know just how old, from the standpoint of actual use, the equipment here tested was. He would like to have tests made on the same or similar equipment after a reasonable and definite amount of service. It is very important to know, when dealing with apparatus depending largely on the precision of its construction for satisfactory performance, just how well such apparatus retains its characteristics during service. In other words, the engineer is not so much interested in the maximum performance obtainable under laboratory conditions as in the average good performance which may be expected during a reasonable life. This point has been neglected, and it is of the utmost importance. It is to be regretted that the representatives of the manufacturer saw fit to ignore one or two queries of this nature at the meeting.

Author's Closure

It is true that this paper deals only with a small part of the problem of correctly using the fuel in Diesel engines. The many designs of engine combustion chambers, each with turbulence of the air resulting from the movement of the piston and the movement of the flame front, would make a great number of tests necessary to examine these other factors with our present technique. Incidentally, a knowledge of the spray dispersion in quiet air, together with information on operating dispersion characteristics, such as can be obtained by overloading the engine and observing the soot markings, may give the engine manufacturer qualitative information on his cylinder turbulence. It is hoped to test other designs of injection systems, and loans of additional equipment for this purpose are desired.

The pressures of the air into which the test sprays are injected have little intrinsic meaning. What is desired in dispersion tests is that the density of the air shall be the same as under operating conditions. A pressure of 200 lb per sq in. was used, because at room temperature the density of such air corresponds to that in an engine compressed 14:1. Again, the issue velocity of the oil depends on the pressure difference between the oil-line pressure and the air pressure. Operating conditions may be closely approximated, therefore, simply by changing the pressure of the oil at the nozzle to compensate for the air pressure.

While the author does not know how many hours this particular pump system was operated prior to making these tests, he knows that it is about three years old. This paper does not pretend to deal with all of the important considerations of injection systems, such as wear, cost, etc., but only with the spray characteristics.

PME-55-1

Progress in Petroleum Mechanical Engineering

OIL and its products have become so powerful a factor in transportation the world over that the long-continued depression has had little effect in disturbing the petroleum market. As a result the oil industry gives every indication of leading the world out of its economic slump.[1] Of course, actually, it has affected the volume of business, and it is only because the refiner was supplied with crude contracts at low prices that he was able to maintain a profit on refined products, although even this is not generally true throughout the country. Likewise, the reduction in volume and consequent profit over operating expense and overhead has been partly offset by cuts in salaries and wages, as well as reduction in labor by reason of operation of larger units.

It is in times of depression that the refiner must look to additional economies that may be obtained, and for this reason the forward-looking refiner has continued his research work where it seemed most promising in results.

While it is the one major industry enjoying a sustained and comparatively intact market, the arrest in normal expansion due to the depression has emphasized economic weaknesses particularly during the past year, which have commanded considerable time and thought toward their elimination on the part of leading executives.

To reduce the hazards of unnecessary and dangerous competition in export fields, two conferences of major world petroleum leaders were held, which resulted in establishing a working agreement among British, Dutch, Rumanian, and American interests. The Soviet failed to join.[2] The most obvious effect of the bitterly fought tariff on crude imports to this country that was made a law last June is shown in the readjustment of domestic corporations with foreign concessions and facilities. Organizations mainly interested in the domestic market are selling their foreign holdings to those already well established in export fields. There has been a marked increase in foreign refining capacity in the year, and diversion of South American crude from our shores undoubtedly will provide an added stimulus in this direction.

In our domestic field the industry has failed to take full advantages of the sustained market due primarily to continual disturbance in crude-oil production, which did not balance supply with demand. Control systems have been set up in the various producing states, but either through lack of proper legal authority or weaknesses in the agencies, the attempts to curtail production in line with demand have not been very satisfactory. So serious is this problem that voices of leading executives have been heard at many public gatherings offering plans to curb waste and balance production with demand. Arresting attention, particularly of the engineering profession, is that of Mr. W. S. Farish, presenting "A Rational Program for the Oil Industry."[3] Briefly, his program of balancing supply with demand recognizes a compelling right of the public to a proper supply of oil and its products at reasonable prices; it recognizes the right of the public to an intelligent conduct of the oil business to insure security of employment both for the dollar and for the man; it demands the protection of the small independent operator as well as the large integrated unit; and it demands the application of sound engineering principles. Improvement toward stabilization through present efforts at control is evident and will be more so when the industry adopts some such broad program.

A disturbing factor to the domestic market is due to the ease with which taxes may be collected on motor fuels and lubricants. The ever-widening margin between retail sales price and superimposed tax is producing an additive stimulus to illicit production. Through the bootlegger, the state not only loses the taxes, but is responsible for subjecting a law-abiding industry, in which it has a direct interest, to a severe and punitive competition.

To aid in relieving general distress, the oil industry is rapidly adopting the "share the work movement," and many of the larger units are now operating on schedules of a five-day week.

The year has been marked by mergers, notably the Standard Oil Company of New York and the Vacuum Oil Company in Socony-Vacuum Company;[4] Sinclair and Prairie in the Consolidated Oil Company,[5] with rumors of a combination of the Standards of New Jersey, California, Ohio, and Kentucky afoot.[6] Meanwhile the Consolidated and Standard of California are bidding for Richfield, and the Standard of Indiana has sold its Pan-American foreign holdings, including a refinery on the island of Aruba, D. W. I., to the Standard Oil Company of New Jersey.[7] Three major refining organizations, in cooperation with a large equipment manufacturer, have pooled their patent interests in the formation of the Petroleum Distillate Corporation. The Standard Oil Development Company (of New Jersey), The Atlantic Refining Company, the Standard Oil Company of Indiana, and the Foster-Wheeler Corporation are the organizations involved, the latter acting as exclusive licensing agent for the pooled patent interests. The Southwestern Engineering Corporation has been appointed agent for the Donnelly and Jenkins processes.

Refining

"Continuing failure to reduce refining operations to accord with market demand has added to economic difficulties of the industry. Blame is in part due to distress crude seeking a market at any price and in part to a competition for gallonage prompted by a surplus refining capacity. As long as refiners are willing to sell gasoline on too wide a margin, expand into new territory, and attract new outlets by further reductions, the present disproportionate distribution costs will continue."[8]

"Flexibility, operating economy, and avoidance of patent

[1] Fifth Annual Report of the Technical Oil Conservation Commission.

[2] *Oil and Gas Journal*, May 12, 1932, p. 14; June 9, p. 100; July 28, p. 11. Interview with Viscount Bearsted and Sir John Cadman, *New York Times*, Nov. 4, 1932.

Presented at the Lubrication Testing session, Thursday afternoon, December 8, 1932, during the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Contributions for the Progress Report were made by T. H. Hamilton, W. G. Heltzel, and David L. Trax.

Personnel of the Executive Committee of the Petroleum Division for 1932: W. G. Heltzel, Chairman; H. J. Masson, Secretary; Harold Adkison, Mid-Continent Petroleum Secretary; Walter Samans, J. W. Hays, E. H. Barlow, and D. L. Trax. Associate: A. H. Horne.

[3] W. S. Farish, "A Rational Program for the Oil Industry," before Petroleum Division, A.I.M.E., at Ponca City, Okla., Sept. 30, 1932.

[4] *Oil and Gas Journal*, March 3, 1932, p. 24.

[5] Ibid., Sept. 15, 1932, p. 8.

[6] Ibid., Dec. 31, 1931, p. 63.

[7] Ibid., April 28, 1932, p. 18.

[8] Ibid., Sept. 15, 1932, p. 28.

difficulties appear to be the guiding hands in the development and improvement of cracking-plant technology, irrespective of the value and application of cracking to the solution of the problem of maximum yield of gasoline possessing the highest anti-knock properties demanded by the automotive engine.

"The increases in both yield and anti-knock quality in cracked gasoline are attributable to a large number of new developments of the different methods of application of older ideas. A multiple type of cracking furnace, a viscosity-breaker furnace employed to produce clean cracking stock for the main furnace, and transmitting the greater part of the heat energy in radiant rather than the convection bank are some of the characteristics of new stills.

"Quenching the oil after passing the reaction chamber to reduce carbon formation; use of fractionating towers in direct connection with stills; cooking retorts . . . which yield more charging stock; reforming gasoline, naphtha, and cracking kerosene high-pressure and high-temperature operations; reforming lower hydrocarbons to produce heavier molecules of the gasoline range; use of vapor phase condition along with the liquid phase to increase the anti-knock value, are some of the features to attain the desired results in old as well as new installations."[9]

Attracting considerable attention in the industry is the recent commercial introduction of hydrogenated motor fuels and lubricants. These products are manufactured under pressures and temperatures ranging as high as 3500 lb per sq in. and 1000 F, meaning considerable advance in oil technology plant engineering, plant and safety control, and metallurgy.

The industry was skeptical of the practical operation of commercial plants because of the unheard-of advance in pressures and the hazards involved in hydrogen manufacture and handling in the process. Through the development of delicate controls and sensitive automatic safety alarms, commercial runs of one year's operation without shutdowns are cited to show the ruggedness of the catalysts and the workability of the process. Although working with oil-hydrogen mixtures at high temperatures and pressures, no lost-time accidents have occurred due in any way to the process. The commercial equipment is remarkably flexible from the points of view of feed stocks and products derived, the latter running from lubricants to aviation fuel with no coking in the process. Two new products of this process of interest are a high-flash safety fuel for aviation and marine use and high-solvency naphthas.[10] The final test of hydrogenated products, public demand in a competitive market, is now under way.

To meet the increasing demand for cracking capacity, one large Eastern refiner installed and is now operating a "tube and tank" unit of 10,000 bbl capacity. The unit has duplicate furnaces, each of 40,000,000 Btu heat capacity at normal rating, connected to a common fractionating and condensing system. This arrangement not only reduces capital investment but adds to flexibility of operation and ease of repairs to either heater without complete shutdown. The major portion of the heat is absorbed in the radiant section, and the furnace is so designed that the heat-absorption rate at no point exceeds 10,000 Btu per sq ft per hr.

The two soaking drums, designed for a working pressure of 1000 lb at 900 F, are each 70 in. in inside diameter by 40 ft high, with wall thickness of 7 in., and were hot forged and pierced from a single billet of steel—no seams or joints.

The separator is of the vertical type, 11 ft in diameter by 45 ft high, with a wall thickness of $1^3/_8$ in. designed for a working pressure of 150 lb. The fractionating tower is 12 ft in diameter and 90 ft high, weighs 325,000 lb, and has 25 bubble plates. The hot-oil charging pumps are of the crank and flywheel duplex type with outside packed plunger and are run through reducing gears by simplex, double-acting, tandem gas engines. The speed of these engines may be varied from 70 to 140 rpm by a throttling governor automatically maintaining the correct air-gas ratio.

A large mid-continent refiner is completing a 20,000-bbl combination unit for skinning and cracking, said to be the largest unit of its kind to date. This unit is designed to convert 70 per cent of the daily 20,000-bbl run into gasoline with an octane rating of 70. The only by-products of the operation will be the refinery gas and a superior grade of fuel oil having a viscosity of 150 to 200 sec at 122 F furol. Some idea of the size of this unit may be gained from the volume of one of the two furnaces, which is 74,000 cu ft, and the hourly heat energy estimated to be released is approximately 2,000,000 Btu. This unit will perform five operations: (1) Skimming. (2) Heavy virgin gasoline is separated and reformed by cracking to improve the anti-knock value and volatility. (3) Reduced crude is subjected to a viscosity-breaking operation to reduce fuel-oil quantity and provide cleaner cracking stock for the principal cracking operation. (4) The pre-processed fuel-oil charging stock is flashed to separate it into cracked gasoline, clean charging and recycle stock, and fuel oil. The clean charging stock is separately cracked in another operation to give maximum yield of high anti-knock value. (5) The various gasolines are blended and stabilized, the motor-fuel production representing 70 per cent of the mid-continent crude oil originally charged to the unit.

The demand for higher octane gasoline, calling for increased cracking capacity, has in turn called for gas- and vapor-recovery systems in economically operated plants. One large Eastern refiner has installed a recovery plant capable of processing 15,000,000 cu ft of gas daily and to recover upward of 50,000 gal of gasoline each 24 hours.[11,12,13] The year has been marked by the more general use of debutanization and stabilization plants.

Equipment manufacturers supplying the industry have kept pace with progress in refining by developing alloys resistant to heat and corrosion; new processes in heat treating and welding of pressure vessels; more sensitive instruments for measuring, controlling, recording, and providing safety in plant operation; and improved pumps and compressors to meet the increasing variety and demand of modern refinery operations.

A notable contribution by an Eastern equipment manufacturer is an improved type of heat exchanger, shown in Fig. 1. A feature is the elimination of tube sheets, so often the cause of leakage troubles, particularly where high temperatures and pressures are involved. Eliminating the tube sheets permits a better tube spacing with a view to increased heat transfer, permits the use of larger tube sizes, and by casing the tube bundle, does away with short circuiting around baffles (reduced efficiency), so often the case in the usual design. Several of these units have been in practical operation on high temperature and pressure work and have not only returned a remarkable increase in efficiency over the usual type, but demonstrated ease of inspection and tube cleaning.[14]

Because of the rapid increase in knowledge of the chemistry and technology of petroleum and its constituents made during the past year, the problem of efficient plant design for a given crude and market demand is becoming increasingly complex. "One thing is clear, however, and that is that technical advancement is fast. The difficulties of refining facilities are numerous. The problem becomes all the more intricate through knowledge

[9] *National Petroleum News*, July 27, 1932.

[10] "Recent Progress in Hydrogenation," *Ind. & Eng. Chem.*, Oct., 1932.

[11] "Permanent Recovery Systems," by George Reid, *Refiner*, Sept., 1932.

[12] "Modern Design in Tide-Water Cracking Facilities," *Refiner*, Oct., 1932.

[13] *Oil and Gas Journal*, March 3, 1932, p. 22.

[14] Name of manufacturer cited will be furnished on application.

of the fact that what is selected and constructed today will in all possibility be obsolete five or six years hence, for this industry has definitely abandoned its old-fashioned 'hammer and tongs' practices."[15]

The recently issued Recommendations on Refinery Heat Exchangers by the A.S.M.E. should prove a boon to the industry in clarifying difficulties between refiners and equipment manufacturers in an increasingly important phase of development of the modern plant. Other papers of interest contributed under the auspices of the A.S.M.E. during the past year are: "Properties of Hydrocarbon Mixtures at High Pressures," by W. K. Lewis and C. D. Luke, Massachusetts Institute of Technology; "Heat-Transfer Design Data and Alignment Charts," by T. H. Chilton, Allan P. Colburn, Raymond F. Genereaux, and Harcourt C. Vernon, E. I. du Pont de Nemours & Co., and "Flow of Fluids in Closed Conduits," by R. J. S. Pigott, Gulf Oil and Refining Company.

TRANSPORTATION PROGRESS

There has been very little expansion in oil-pipe-line facilities during the year 1932. One reason is that the capacity of pipe lines had been built up to meet the peak requirements for petroleum products of 1930 and 1931. The other important reason is that the only large oil field developed during the last year was the East Texas field, which is near the Gulf Coast, and much of this petroleum was moved over the existing trunk lines or short lines built to the Gulf. The decreased consumption in petroleum products during the last year has resulted in decreased pipe-line runs. Consequently, the pipe-line industry has not expanded to any extent—which has reacted on purchases of pipe machinery and equipment. A very large amount of second-hand equipment was used in the East Texas field. It appears now that there should not be any appreciable expansion of either the gathering systems or trunk lines during the year 1933, because

(1) The existing fields should be able to supply the crude oil requirements for the coming year
(2) The consumption of petroleum products during 1933 will very likely not exceed that of 1930 to 1931, for which the pipe line capacities were built.

An oversupply of crude petroleum has resulted in state control over some of the newer and larger oil fields. This control is generally known in the petroleum industry as "proration." Until about two years ago petroleum was produced from a well to its full capacity. This practice was characterized not only by a waste of natural gas, petroleum, the lifting power of the gas, and a disorderly development of the field, but also by an economic waste in the overexpansion of production, natural gasoline, and pipe-line facilities, which had to be depreciated in a comparatively short time. Proration means that a given field is allowed a certain amount of production, and the wells, with exceptions, are allowed to produce only a prorated amount of oil in a certain period.

From the pipe-line viewpoint there is a distinct advantage of extending the life of an oil field and eliminating the overdevelopment of pipe-line facilities to take care of peak loads. There have been certain disadvantages to pipe lines in operating in oil fields which are prorated; but the orderly development of oil fields over a period of years should result in more economical and better operating conditions. As a measure of conserving the natural petroleum resources some form of proration will probably continue.

Gathering Systems

The last year has marked the further use of the gasoline-motor-driven unit for gathering oil from the many points in a new oil field. This type of unit has definitely replaced the steam pump for this service.

Just as the Diesel engine replaced the steam engine for trunk-line pumps years ago, so the gas- or gasoline-motor driven pumping unit has displaced the steam pump in the gathering system during the last two or three years. The steam pump is obsolete for this service.

Electrical Motor Drive

About the year 1926 the electrical motor came into use for driving reciprocating pumps for field gathering stations and, later in that year and the following year, the use of electrical motors was extended to driving centrifugal pumps as booster units on trunk lines. In 1928 and 1929 the electrical motor came into rather extensive use for driving centrifugal pumps on newly built trunk lines. The electrical motor was also used extensively on trunk lines for driving reciprocating pumps. During that period, pipe-line engineers in general were inclined to favor the use of the electrical induction motor.

The last two years has marked a very decided change of attitude on the part of these engineers as to the use of electrical motors. This change has been brought about by the following reasons:

(1) The rates of the electrical power companies were very complicated, and did not fit into the operating conditions of the pipe-line industry
(2) Electrical-power rates were not in competition with the power developed by the Diesel engine
(3) The electrical motor has a constant-speed motor, lacking in flexibility in operation, which could easily be obtained without penalty through the use of the Diesel engine
(4) The demand charges set up in the rates of the power companies met with considerable disfavor on the part of the pipe-line engineers and executives
(5) The cost of Diesel engines during the years 1929, 1930, and 1931 was sufficiently low to make the power rates too high to compete. The years 1931 and 1932 were marked with substitution of gas- or gasoline-motor-driven pumping units for the electrical motor, and under the existing condition, this trend will very likely continue.

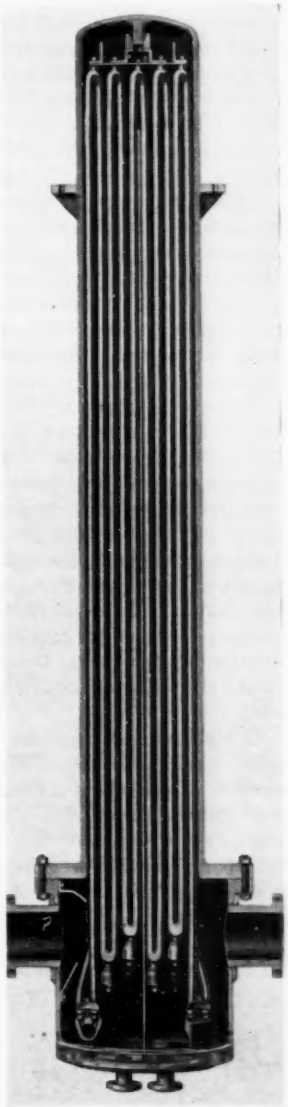


Fig. 1

[15] "The Problems of Technical Progress," *Refiner*, Oct., 1932.

Pipe

Only a small amount of pipe has been purchased by the pipe-line industry during the year 1932, because there has not been any great expansion in the industry and the surplus stock and second-hand supplies have been consumed.

In the future, seamless pipe will undoubtedly be used instead of the lap-welded pipe. In the larger sizes, both seamless and the electrically welded pipe will be used. Lap-welded pipe is almost obsolete in the pipe-line industry.

For years, the 3-in. pipe has been used generally, but it appears now that this is becoming an obsolete size, and it is likely that the industry will use 2-in. and 4-in. pipe, and 3-in. pipe will not be carried in stock after the supplies have been exhausted. Of course, 3-in. fittings will probably become obsolete with the 3-in. pipe.

During recent years there has been a tendency for pipe-line companies to use pipe with a wall thickness less than that which has been used for years. There have been some cases in which the wall thickness has been reduced very materially in the larger sizes of pipe. There has been some pipe laid with $^1/_4$-in. wall thickness, as compared with the $^3/_4$-in. wall thickness for standard pipe. It has even been suggested that $^1/_8$-in. wall thickness be used.

The wall thickness will be reduced, apparently in the 8-, 10-, and 12-in. sizes of pipe, from the standard thickness. It is the author's opinion that it may be reduced by $^1/_{16}$ in., or to the extreme of $^1/_8$ in., but the corrosion factor is of such importance that unless a satisfactory coating is developed, it cannot be reduced much lower than that. Besides the corrosion problem, there are certain mechanical problems involved that limit the reduction of the wall thicknesses of the pipe. It is known that the wall of the pipe fails from corrosion, not so much due to the pressure inside of the pipe, but to the wall being eaten through eventually; the thicker the pipe, the longer the life of it, but of course there is an economical thickness that should be used in the wall.

It has been customary in the construction of new pipe lines to purchase pipe in lengths of 40 ft, plus or minus, but the laws of the various states make it difficult to transport such lengths of pipe over the highways today. There are certain provisions also in the laws of the various states which limit the weight that can be transported on trucks. These regulations will undoubtedly limit the length of the pipe for new trunk lines which are to be built in areas where the state highways traverse the country. It is possible that some special permission could be obtained, but this remains to be worked out.

Fittings

Except in the small sizes, 2 to 4 in., the cast-iron and cast-steel fittings have become obsolete. Forged steel flanges, bends, tees, caps, etc., have come into use extensively, and undoubtedly have made the larger cast fittings obsolete. This class of material has been made obsolete by electric and acetylene welding.

Diesel Engines

Of course, the Diesel engine in the lower speed has been definitely established as the driving unit for trunk-line reciprocating pumps. One of the most important developments in the Diesel engine for pipe-line service has been the trend to the use of the solid-injection engine. This type of unit will undoubtedly be used in the future installations instead of the air-injection method.

Probably the most important development in the use of Diesel engines for pipe-line service has been the installation of some high-speed Diesel engines connected with gear increasers to centrifugal pumps. One large pipe-line company installed a 300-hp, 750-rpm Diesel unit to drive through a gear increaser a centrifugal pump with a capacity of 25,000 bbl at 430 lb pressure. This unit has the desirable features required for field-gathering pumping stations. Such a unit is portable, easily installed, has low investment cost, and has the flexibility of operation which cannot be obtained in the electrical motors. It is very likely that this type of unit will be used in the future for oil-field gathering stations, instead of the electrical motor. Its low operating cost puts the electrical motor out of competition for this service.

The high-speed Diesel-engine unit connected with a 3600-rpm centrifugal pump has brought into use the gear increaser. It looks as though a new field has been developed for gear increasers.

Many pipe-line engineers desire in a Diesel-engine installation for driving oil-pipe-line pumps to have the water-circulating pump driven off the individual unit, rather than having the circulating water supplied from some centralized pumping installation. The installation of the water-circulating pump on each unit gives more assurance that the engine will not be operated without this supply of cooling water. Generally, the pipe-line engineers have been required to figure out a drive for the circulating-water pump, and in many cases it has been driven off the gear of the pump. However, with the development of the closed water system making use of the heat exchangers, there is a desire on the part of pipe-line men to have the water-circulating pump driven off the engine, because the Diesel-engine manufacturer has not paid much attention to this feature, but it is one that is worth considering. In the future, it is likely that the pipe-line engineer will expect the Diesel-engine manufacturer to design a Diesel-engine unit which will come out equipped with a circulating-water pump and a heat exchanger for cooling the water.

Heat Exchangers for Closed Cooling-Water Systems for Diesel Engines

About three years ago the heat exchanger came into use as a part of a closed cooling-water system making use of the crude oil pumped for cooling the water. The use of the heat exchanger for this service has been extended further during the past year, and its use will very likely make the open cooling-water system obsolete. The general practice in the industry for years has been to make use of the open cooling-water system, which involved earthen or concrete reservoirs, spray systems, water towers, and an intricate system of piping. In many places pipe-line stations have inadequate or poor water supplies, and consequently the operating conditions are made difficult on account of this poor water supply or the lack of it. The heat exchanger, making use of the crude oil, has simplified this problem, because only a very small amount of water is required for the closed system, and the oil which is pumped through the trunk line does not reach a temperature much above 80 deg in the summer time, and of course as low as 32 deg in the winter time. One of the worst operating problems encountered with bad weather is cracked cylinder heads. The closed cooling-water system corrects this fault and remedies many others that are encountered, when cooling water is used for Diesel engines.

Most of the installations have been made with the heat exchanger on the suction line to the trunk line or the trunk-line crude-oil pump. Such an installation involves, usually, a booster pump on the suction line to feed the oil through the heat exchanger to the main discharge pump, because there is too much friction loss in the heat exchanger to permit the pumps to fill, if such provision is not made. During this year two installations have been made with the heat exchanger on the discharge line from the crude-oil pump. Such an installation simplified operations, and it is more economical. These heat exchangers

on the discharge lines have proved practicable and serviceable up to this time.

Motor Transport

For years, the slow heavy-duty truck has been used in the pipe-line industry, because their loads were heavy, and as a rule the roads were such that speed could not be made. The last three or four years have marked very important changes in the pipe-line industry, both as to the condition of roads and as to the load which was carried on them. A few years ago repairs in pipe lines were made by a large crew of men, and heavy materials and tools were required. The use of welding, both electrical and acetylene, has changed these methods so that today a comparatively few men are required for a pipe-line crew, and materials that must be transported are much lighter in weight. This change has eliminated the necessity of having the trucks carry a very heavy load. Likewise, during the last three years the expansion of concrete roads and graveled highways has made it possible for pipe-line trucks to make considerable speed along the route of the pipe line. For these reasons, the lighter weight truck with a higher speed is replacing the heavy-duty truck with a slow speed. The truck with a capacity of $1^1/_2$ to 2 tons, and capable of making 45 miles an hour, will replace the trucks of $2^1/_2$ to 3 tons with a speed of 18 or 20 miles an hour. This change has occurred primarily during this year, but it will be very marked within the next year or two.

It is the customary practice of the pipe-line industry to equip their trucks with either a hub or a body winch, with the necessary braking provisions on the winch. This feature should be considered in the design of trucks for pipe-line service.

One feature which is very important to pipe-line trucks is the proper clearance, because much of the time these trucks have to follow the course of the pipe-line right of way, which is not graded, and the necessary clearance must be in these trucks to avoid difficulties.

Welding

Electrical and acetylene welding has become thoroughly established in pipe-line practice. As pointed out in previous reports, welding has revolutionized the methods of constructing and repairing pipe lines. Each year marks an improvement in the welding methods, also extension of its uses. Welding has been speeded up, both in repairing lines and in the constructing of lines. An example of the efficiency of the acetylene welding in pipe-line service can be pointed out. During the last year one large pipe-line company constructed about 370 miles of line in which there were about 49,000 welds made on 8-in., 10-in., and 12-in. pipe. In those 49,000 welds only two small pinhole leaks developed. One of these pinholes in the weld was found while there was no pressure on the line. The other leak developed after oil was put in the line. This is a very interesting example of the efficiency of welding, because the welding was done over a stretch of about 700 miles, and the welds were made by many different welders under all kinds of working conditions.

Although there has been extensive use of both acetylene and electric welding in the pipe-line industry, there still remains a very worth-while field of development. Until recently, most pipe lines were laid with screw joints, and a considerable amount of mileage is buried in the ground with screw joints. These joints leak, and considerable maintenance cost is experienced on account of these leaking couplings. If it were only the leaks and the loss of the oil that the pipe-line companies have to contend with, the fault would not be so serious, but it is now known that when the oil leaks around these couplings and along the pipe lines buried in the ground, a very serious corrosion results within two to three years. Some pipe-line companies believe that 50 to 75 per cent of their underground corrosion problems is caused by the oil leaks from screw couplings. One pipe-line company has welded these couplings, so that they will not have this leakage problem during the years to come; other pipe-line companies have considered it. It remains, however, that leaking couplings develop frequently and that they must be repaired permanently. There is a certain amount of danger in making these welds by hand, because the coupling may spring a leak while it is being welded and a fire may follow. It is believed that the welding companies, either the acetylene or the electrical groups, should develop an automatic method for welding up both sides of pipe-line couplings which are on lines in service.

Automatic Pumping Stations

A few years ago there was considerable work being done by pipe-line engineers on automatic pipe-line pumping stations, and the results of their work were surprising. However, the last two years have marked an oversupply of labor, and consequently the development of automatic equipment has come to a standstill. The group of engineers working on this problem under the sponsorship of The American Society of Mechanical Engineers has felt that, until times change, when more men are required for other work, these automatic pumping stations should be held in abeyance.

Metering of Oil

The regulation of the oil fields by the states has brought into prominence the use of meters for measuring oil from oil-field gathering stations. Thus, the trend to the measuring of oil by meters will probably continue, not because of state regulations, but because of the need for such equipment in the pipe-line industry.

Pipe-Line Bridges

One of the principal maintenance problems in pipe-line work is creek and river crossings. The worst corrosion problems are found in creek crossings. Considerable trouble is experienced also in keeping these pipe-line crossings in place, because of the erosion of the banks and the changing of the channels of the creeks and rivers. To meet this problem pipe-line bridges have come into use, and it is likely that there will be a continued trend in this direction if the bridges can be made and installed at a cost which will be competitive with the replacement of the pipe and the maintaining of the creek banks for safety.

PRODUCTION PRACTICES

It is plainly evident in reviewing the progress in oil-field production and drilling practices that the proration or curtailment of oil production has continued in almost every oil field during the year. During this year, like last year, all producers of crude oil have been compelled to turn their attention to more fundamental and effective methods of attacking problems of economy on account of the low price received for the crude oil. Outward manifestations of progress are less obvious than usual, for the nature of the work accomplished and the improvements in progress have not been of the spectacular sort.

However, among some of the important improvements worthy of discussion are well-pumping equipment and operation, development of bottom-hole pressure gages, use of tapered tubing, and hoisting equipment.

Well-Pumping Equipment

Most of the progress made in this equipment has been confined to perfecting and modifying the existing equipment. Surface installations being generally adopted for heavy wells are steel rig fronts with substantial concrete foundations, rather than the

heavy timbers previously used. Each member is fabricated from structural steel sections, electrically welded or riveted, and supported by suitable concrete foundations. Anti-friction bearings are used where necessary to increase mechanical efficiency. Manufacturers of this type of equipment have responded loyally to the demands of the engineers of the oil industry by substantial improvements in design and material.

Sucker-rod manufacturers are continually improving this equipment for deep-pumping requirements. Special high-tensile alloy steels and careful heat treatment are now employed with marked results. Internal stresses in the metal resulting from the upsetting operation are removed, and the grain structure of the steel is improved by heat treatment. Combination or tapered sucker-rod strings, $^3/_4$ in. and $^7/_8$ in., are used successfully in pumping deep wells. The number of rods of each size for the string is calculated knowing the well load, and provides a more uniform distribution of stress in the rod string.

Improved field practice of thoroughly cleaning and tightening sucker-rod joints has practically eliminated joint failures, and under the most severe conditions sucker-rod service is considerably greater than that obtained several years ago.

Severe demands are made today on oil-well pumps and parts, especially in deep-well operation, and manufacturers are constantly improving or developing new designs of this equipment. Low daily allowable production has greatly furthered the adoption of the insert-type pump by many operators. Pumps of this type save considerable time in replacements or repairs, since the complete pump can be pulled from the well by pulling the sucker rods only. The diameter of the insert pump is limited by the size of tubing through which it must pass; therefore the pumping capacity is somewhat less than that of the common type pump, which is attached as a part of the tubing, and therefore possesses greater inside diameter and greater displacement. Oil-well pumps of small diameters, 1 in. and $1^1/_4$ in. i.d., are rapidly gaining favor for the pumping of wells of small daily production, 50 bbl or less. Small-diameter pumps reduce the load on sucker rods and other pumping equipment; also the maintenance expense and initial investment are considerably less than with pumps of larger diameter. In some cases daily production has been increased by use of the small-diameter pumps by increasing the pumping time, thereby maintaining a low fluid head and obtaining a constant flow of oil from the producing formation.

Vast improvements have been made in design and material of pump-valve cages, valves, and seats, in order to prevent excessive wear, thus materially reducing the number of repair jobs on many wells.

Most oil companies now keep detailed records of pump repairs, so that service obtained from all vital parts of the pump can be checked and improvements made where possible. Analyses of pump-repair costs involve material and labor charges as well as production losses arising from well shutdown time. It is therefore obvious that continuous service in operation with a minimum of shutdown time is very important from an economic standpoint.

Bottom-Hole Pressure Gages

Instruments for measuring the pressure at the bottom of the well bore are rapidly coming into general use in the major oil fields of the country. There are many types of both maximum indicating and recording gages available, some complicated and expensive, and others simple and inexpensive. Probably the simplest in design is that of using a heavy bronze tube equipped with a Schroeder tire valve in one end and an outlet valve in the other end for attaching an indicating test gage. This device, less the gage, is lowered to the bottom of the well and allowed to remain approximately 10 minutes in order to permit pressure inside and outside of the tube to equalize. It is then pulled to the top of the well and the test gage is attached, which indicates the pressure in the tube—the bottom-hole pressure, assuming no leakage has occurred.

The recording type of bottom-hole pressure gage is perhaps the most useful, since a continuous record of pressures may be obtained from the top to the bottom of the hole. There are several of this type available, and involve the use of the Bourdon tube or Sylphon bellows. The recording mechanism consists in most cases of a strip or circular chart, clock operated.

There is only one electric bottom-hole pressure gage known to the writer. This instrument is attached to the bottom of the tubing and gives a continuous reading at the surface by means of an electric cable connected between the gage and a recording apparatus at the well head. The angular deflection of a sealed helix in the instrument sets up a vertical motion, which is transmitted to the surface, of an armature moving in an inductance bridge coil. No field tests have been made with this instrument to date.

Practically all bottom-hole pressure gages are provided with maximum recording thermometers in order that corrections for temperature may be made in the pressure measurements. Petroleum engineers have determined that bottom-hole pressure measurements should be correct within 5 lb to be of most value.

Since the decline of oil and gas production is directly proportional to the decline in pressure in most oil-producing formations, bottom-hole pressure measurements are of considerable value in the study and analysis of flowing and pumping conditions, or production control.

Use of Tapered Tubing

When a well ceases to flow efficiently through 2-in. or larger tubing, conditions are frequently favorable for installing a small tapered string of tubing. The tapered string may consist of tubing as small as $^3/_8$ in. in diameter at bottom, followed by $^1/_2$, $^3/_4$, and 1 in. at the top. A thorough knowledge of the quantity of gas energy available is necessary in the design of the tapered string in order to obtain efficient flow.

The use of small tapered-tubing strings has been used successfully in extending the life of flowing wells, and is particularly adaptable to shallow wells of small production. Much closer study is now being given to the mechanics of flowing gas-oil fluids especially in regard to slippage, friction, and velocity, and it is expected that the use of tapered tubing for flowing small wells will be more widely used in the future.

Hoisting Equipment

When wells are pumped from central power, it is necessary to provide some type of hoisting equipment for removing sucker rods, tubing, or pump from the well in making repairs. Various makes of tractors equipped with winches have been in common use for many years for this purpose. During this year, however, several types of hoisting units mounted on skids or winches on trucks have been offered by manufacturers of this equipment.

Both types of units have been used successfully, and have effected great economies in the repair of wells because of the greater speed in moving from location to location, and thereby covering a larger territory in a shorter period of time than could be accomplished by tractors.

It is expected that during the next year some improvement in the financial status of the oil industry will result from the rigid curtailment of crude-oil production and possible increase in price. Such a condition should bring about expenditures for deeper drilling and development of undrilled acreage. With deeper drilling, the engineers of the oil industry will be confronted with the problem of providing equipment of proper design and material for accomplishing the work economically and efficiently.

PME-55-2

Heat-Transfer Design Data and Alignment Charts

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A search was made, during a research program on heat transfer, for the best available methods for the prediction of heat-transfer coefficients and for means of simplifying design calculations. The equations selected or developed on film coefficients are given in the paper for gases flowing in turbulent motion inside tubes, for gases flowing across tube banks, for free convection outside horizontal cylinders and at vertical surfaces, and for condensing vapors. Alignment charts are given for the solution of these equations. A simplified method of designing gas-heat exchangers was worked out on the basis of one of the equations for the heat-transfer coefficient, and a chart is also given for pressure-drop calculations made in designing heat-transfer equipment.

IN the course of a research program on heat transfer, a search was made for the best available methods for the prediction of heat-transfer coefficients and for means of simplifying design calculations. Equations developed or selected from the literature for film coefficients are presented in this paper for the following cases: Gases flowing in turbulent motion inside tubes; gases flowing across tube banks; free convection outside horizontal cylinders and at vertical surfaces; and condensing vapors. Alignment charts are given for the solution of these equations in which the properties of the fluids are included. A simplified method of designing gas-heat exchangers was worked out on the basis of one of the equations for the heat-transfer coefficient, and a chart representing this relationship is given. A chart is also given for pressure-drop calculations made in designing heat-transfer equipment.

While alignment charts are not relied upon ordinarily for extreme accuracy, the precision attainable is ample for engineering computations, and they afford an easily obtainable check on the order of magnitude of a given quantity. Most of the charts here presented are a combination of ordinary alignment charts with line-coordinate charts, which have been found convenient for the representation of data on any physical properties for which a straight-line function can be found by a choice of suitable coordinates (12).[5] The units employed are those customary in the chemical industry.

Table of Nomenclature

D = diam, in in.
M = mass velocity, thousands of lb per hr per sq ft of cross-section = $3.6\,G$
G = mass velocity, lb per sec per sq ft of cross-section
m = weight rate, thousands of lb per hr
P = pressure, in atm abs
Δp = pressure drop, lb per sq in.
ΔH = pressure drop, in. of water
f = friction factor in pressure-drop equation
L = length, in ft
Z = viscosity, in centipoises
w = density, lb per cu ft (on pressure-drop chart)
ρ = density, lb per cu ft (on heat-transfer chart)
c_p = specific heat at constant pressure, pcu[6] per lb per deg C = Btu per lb per deg F
h = heat-transfer coefficient, pcu per hr per sq ft per deg C = Btu per hr per sq ft per deg F
Δt = temperature difference, deg C
Δt_{mean} = logarithmic mean-temperature difference, deg C
t = temperature, deg C
N = number of tubes in parallel
k = thermal conductivity, pcu per hr per sq ft per (deg C per ft) = Btu per hr per sq ft per (deg F per ft)
β = coefficient of thermal expansion, per deg C
λ = latent heat of evaporation, pcu per lb = calories per gram = (1/1.8) × Btu per lb

Friction Loss in Pipes

The chart for pressure drop in pipes, Fig. 1, is based on the Fanning equation, which is in the units here employed:

$$\Delta P = \frac{0.0004\,fLM^2}{wD} \quad \ldots\ldots [1]$$

Values for the factor f have been assumed to follow a relation with Reynolds' number expressed by the equation

$$f = 0.034\,(Z/DM)^{0.22} \quad \ldots\ldots [2]$$

over the usual range of turbulent motion, i.e., for moderate velocities of other than highly viscous liquids or highly compressed gases. That this is approximately true can be seen when Equation [2] is plotted, as for instance, on the curve of Mc-

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[5] Numbers in parentheses apply to references at the end of the paper.

[6] The symbol pcu = pound-centigrade unit.

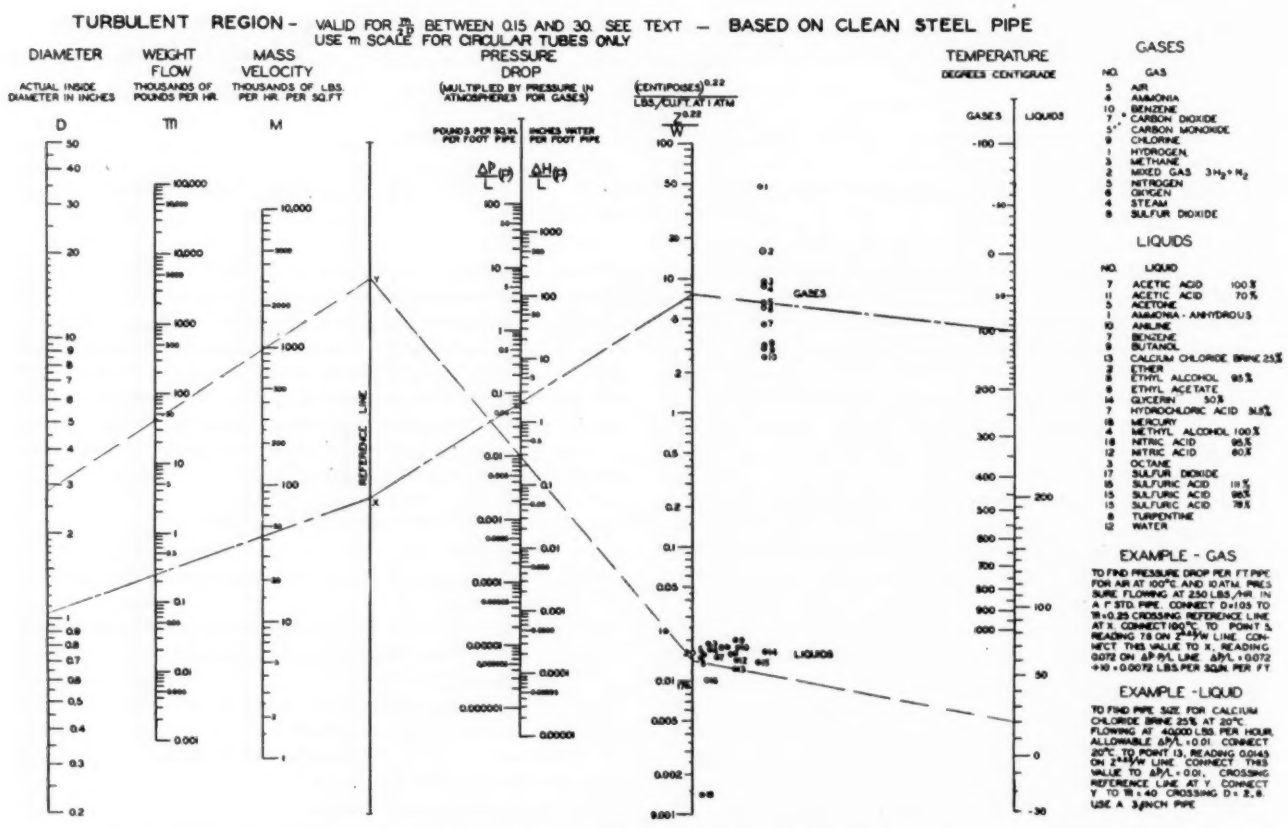


FIG. 1 FLUID FLOW, PIPE FLOW CHART (WEIGHT UNITS)

Adams and Sherwood (7). The chart does not therefore hold for the extreme turbulent region, above $DM/Z = 5500$, where, for commercial pipes, the friction factor appears to assume a constant value of about 0.0053, or for viscous flow, below $DM/Z = 28$. To determine whether the chart is applicable, values of the Reynolds number may be calculated. Alternatively, these extreme cases can be solved directly by the ordinary slide rule, since the equations do not involve fractional exponents:

For viscous flow:

$$\Delta P = \frac{0.0000186\ LZM}{wD^2} \quad \ldots\ldots\ldots\ldots [3]$$

For the extreme turbulent region:

$$\Delta P = \frac{0.0000212\ LM^2}{wD} \quad \ldots\ldots\ldots\ldots [4]$$

Of the results obtained by one or the other of these equations and by the chart, the less favorable should be taken; that is, the higher pressure drop, larger diameter, or lower capacity.

The friction-factor values are for commercial-steel or wrought-iron pipe in good condition. Allowance for roughness can be made, if desired, by adjusting the pressure drop in proportion to the expected friction factor.

It should be noted that for gases the value on the chart is the pressure drop (in lb per sq in. or in in. of water) multiplied by the absolute pressure (in atmospheres), so that to obtain the pressure drop from the scale reading, this product is to be divided by the absolute pressure.

For other than isothermal flow, the temperature to be used is the arithmetic mean temperature of the main body of the fluid. For the absolute pressure, the arithmetic mean of inlet and outlet pressures should be used.

Points can be added for substances not listed on the chart by connecting two values of $Z^{0.22}/w$ with corresponding temperatures and noting where the lines cross.

Notice also that the three left-hand scales are not independent, but may be used to give the third from any two of the values in the equation:

$$m = \frac{\pi}{4}\left(\frac{D}{12}\right)^2 M \quad \ldots\ldots\ldots\ldots [5]$$

This applies, of course, only to circular sections. The chart may be used for other sections by employing the M scale only and substituting for D the "equivalent diameter" (equal to four times the free cross-sectional area divided by the "wetted" perimeter).

HEAT TRANSFER, GASES FLOWING IN TUBES

The equation recommended (2) for predicting heat-transfer coefficients for gases in turbulent flow inside tubes is:

$$h = 13.3\ c_p Z^{0.2}(M^{0.8}/D^{0.2}) \quad \ldots\ldots\ldots\ldots [6]$$

At low velocities, the heat transfer is likely to be determined by free convection. Free convection may be controlling even above the critical velocity (especially at high pressures). In case of doubt for vertical tubes, the free-convection coefficient may be predicted by the chart of Fig. 6 for vertical surfaces and the result used when higher than the values from the chart representing Equation [6]. Equation [6] is for the convection coefficient only and does not include radiation from solid surfaces or gas radiation, which may be appreciable for CO_2, SO_2,

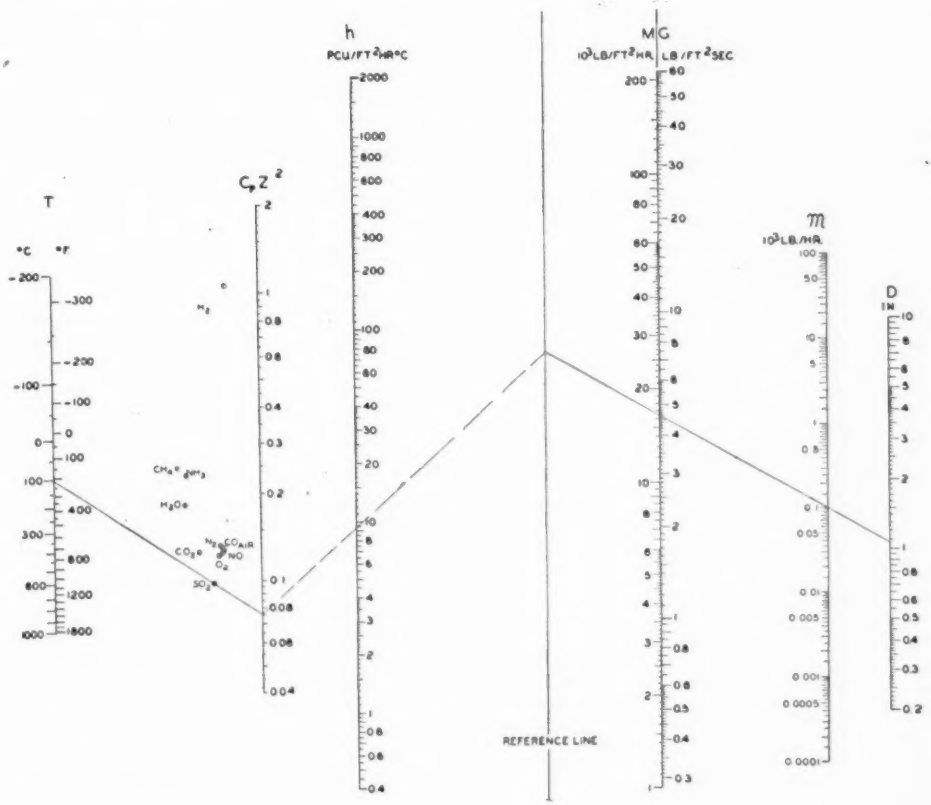


FIG. 2 GASES THROUGH TUBES
[Film coefficient, $h = 13.3\ c_p Z^{0.2}(M^{0.8}/D^{0.2})$. Example: 16,300 lb per sq ft per hr of SO_2 at 100 C average temperature flowing through a tube with 1.06 in. inside diameter. Align 1.06 on D with 16.3 on M to intersect reference line. The crossing at m gives the weight flowing per hour in the tube. Align 100 on T deg C with SO_2 to intersect $cZ^{0.2}$. Connect this with the reference-line intersection. The crossing on the h scale gives the film coefficient, 94 pcu per sq ft per hr deg C.]

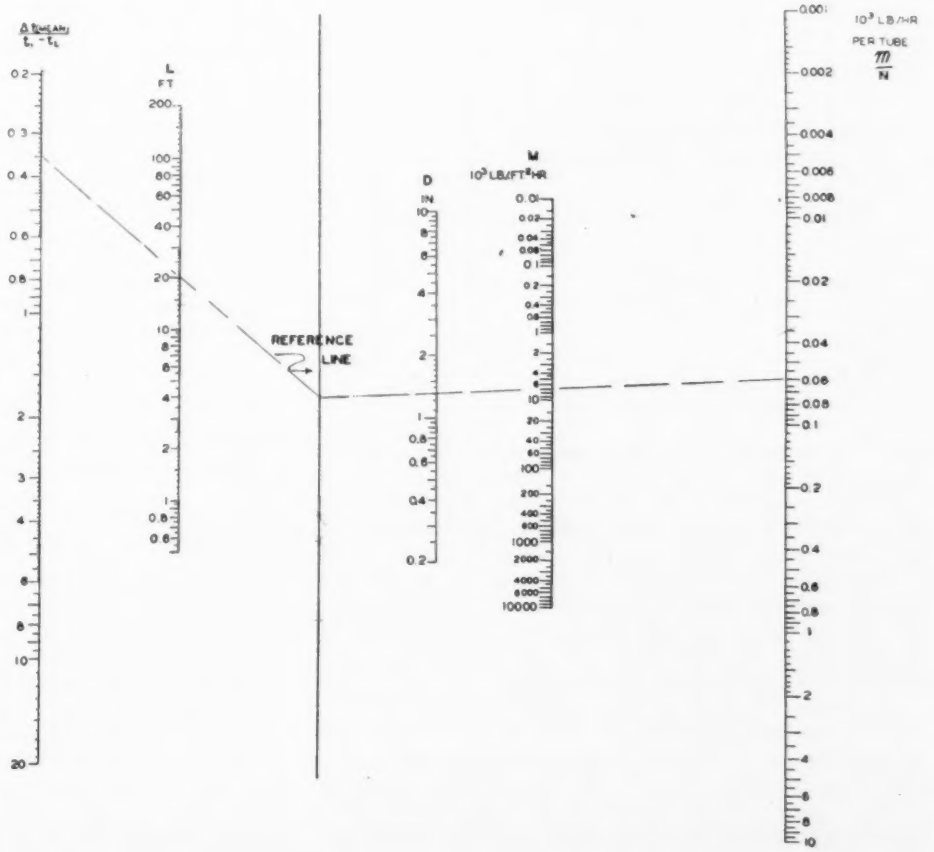


FIG. 3 GASES THROUGH TUBES
[$L(\Delta t\ \text{mean})/(t_1 - t_2) = 9.94\ (m/N)^{0.2} D^{0.8} = 3.51\ M^{0.2} D^{1.2}$, derived from $1000 m c_p (t_1 - t_2) = 5.94\ c_p (M^{0.8}/D^{0.2})\ A\ \Delta t$. Example: Required to heat 600 lb of gas per hr under the following precalculated conditions: Initial temperature 20 C, final 120 C; initial temperature difference to tube 40 C, final 30 C. Then, $\Delta t_{\text{mean}} = (40 - 30)/(\ln 40/30) = 34.9$ and $\Delta t_{\text{mean}}/(t_1 - t_2) = 0.349$. Assume 1.3 in. i.d. tubes; assume economical M about 7. Align 1.3 on D with 7 on M. This gives m/N about 0.06. With $0.06 \times 1000 = 60$ lb per hr per tube, then 10 tubes will be required. Align 0.06 on m/N with 1.3 on D to reference line, thence to 0.349 on $\Delta t_{\text{mean}}/(t_1 - t_2)$. This gives 20.2 on L. The tubes must be 20 ft long.]

NH_3, and H_2O and should be determined from curves of Hottel (5). No allowance has been made for an entrance effect on heat transfer, although an increase may be obtained by unusual entrance conditions.

Equation [6] has a theoretical basis, being derived from the Reynolds analogy equation (18) by substitution of practically the same expression as that given for the friction factor. It is in agreement with data of Nusselt (11) and of a number of other investigators. Data in the literature on the effect of temperature, aside from gas radiation, are very discordant, some being higher and others lower than the effect corresponding to the change in $c_pZ^{0.2}$ with temperature. This latter representation of the temperature effect has a theoretical basis and furthermore checks with that used in Weber's equation (19).

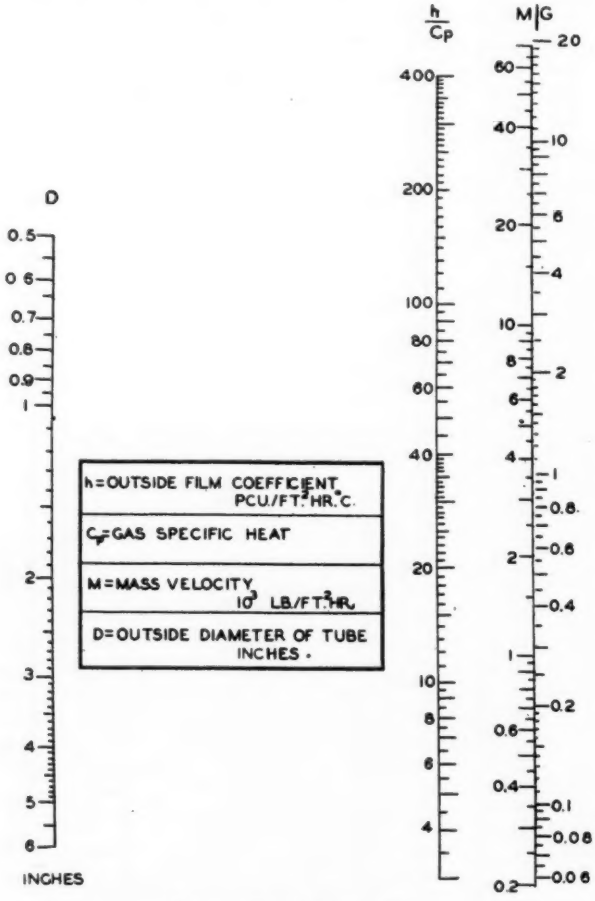


FIG. 4 GASES ACROSS TUBES

The use of the chart, Fig. 2, representing Equation [6], is explained by the example shown thereon. As on the pressure-drop chart, points can be added by joining two values of the function $cZ^{0.2}$ with their respective temperatures for the substance in question. Again as on the former chart, the M, m, and D scales are not independent and the m scale applies only for circular tubes. For other sections the M and D scales can be used, substituting equivalent diameter for D.

If a further simplified equation, originally obtained by F. C. Blake, of the Experimental Station, from some of the same data as was Equation [6], is used for the heat-transfer coefficient, a heat-balance equation can be set up from which the dimensions of an exchanger can be calculated directly in terms of the mean-temperature difference and the temperature rise, on the assumption that the gas film is the controlling resistance. The equation

$$h = 5.94\ c_p(M^{0.8}/D^{0.2}) \quad [7]$$

is equivalent to Equation [6] for air at 20 C ($Z = 0.018$), and should therefore be applicable to other gases consisting chiefly of nitrogen at moderate temperatures; it will be conservative at high temperatures. The heat transferred to or from the gas per hour may be expressed in either of two ways:

$$1000\ M\left(\frac{\pi D^2}{4 \times 144}\right) c_p\ (t_1 - t_2) = h\left(\frac{\pi DL}{12}\right) \Delta t_{mean} \quad [8]$$

Substituting for h by Equation [8], noting that Δt_{mean} refers to the temperature difference from wall to gas, and simplifying, we obtain:

$$3.51\ M^{0.2}\ D^{1.2} = L\ \frac{\Delta t_{mean}}{(t_1 - t_2)} \quad [9]$$

This relationship is represented by the chart, Fig. 3. The example given illustrates the method of making a calculation.

The main advantage of this chart is that it permits the rapid calculation of numerous alternative designs, without separate calculation of the cross-section and surface area of the exchanger at each step. In designing an exchanger, the value of $\Delta t_{mean}/(t_1 - t_2)$ is generally fixed in advance, and the number and dimensions of the tubes are required. The recommended procedure is then to start with a value of M somewhere near the optimum from the standpoint of cost of power for inducing the flow and fixed charges on the exchanger. Formulas for calculating the optimum velocity have been derived by Lewis, Ward, and Voss (6) and by Schack (16). It may be noted that the optimum M increases with the two-thirds power of the density of the gas being handled; for air at atmospheric pressure it ranges from $M = 5$ to $M = 12$. The designs as derived from the chart should be checked for pressure drop by Fig. 1.

An important consideration which can be derived from Equation [9] is that a given ratio of length to diameter will give approximately the same value of $\Delta t_{mean}/(t_1 - t_2)$, influenced only to a small extent by the mass velocity or the actual length.

The three scales D, M, and m/N on this chart are not independent, and apply only for flow inside circular tubes.

GASES FLOWING ACROSS TUBES

The following formula for the heat-transfer coefficient for gases flowing at right angles to tube banks was derived by F. C. Blake from data of Reiher (13) and of Carrier (1):

$$h = 16\ c_p\ M^{0.7}/D^{0.3} \quad [10]$$

The coefficient given applies for more than three rows of staggered tubes. For other arrangements, the following table gives factors by which the predicted coefficient is to be multiplied:

Spacing	Number of rows	Factor
Staggered	>3	1.0
Staggered	3	0.9
Staggered	2	0.8
In line	>3	0.75
In line	2–3	0.7
Single tube	–	0.75

The mass velocity is calculated on the minimum cross-section, or the free area between baffles at the closest approach of the tubes in an average row. The factor for a single tube has been adjusted to conform to the results of Drew and Ryan (4).

The data on which the equation and chart are based are from tests on flow directly at right angles to the axis of the tubes. It is not known what factors should be applied for a baffled exchanger with a moderate number of rows of tubes;

the gas coming up around the baffle is not always flowing at right angles to the tube, and some allowance should be made for this effect. Equipment manufacturers recommend factors as low as 0.5 to be applied to the calculated coefficient. Natural convection may, for cases of low velocity or high pressures, give rise to higher coefficients than predicted by this equation.

The alignment chart for this equation, Fig. 4, takes the simplest form, of three parallel scales. An example is scarcely necessary.

Free Convection

Coefficients for heat transfer by free or natural convection depend on the properties of the fluid and the temperature difference, and the alignment charts for the equations developed are particularly good examples of the advantages of the method of plotting here employed.

For heat losses from horizontal cylinders the equation:

$$h = 100 \sqrt[4]{\frac{\beta \rho^2 k^3 c_p \Delta t}{ZD}}. \quad [11]$$

agrees with data cited by Rice (14) and Nusselt (11) for tubes of ordinary diameters (not for fine wires).

It is of importance to notice, since h varies as the square root of the density, that gases at high pressures give high coefficients by natural convection, as shown by the example given on the chart, Fig. 5.

To illustrate the use of the chart for liquids, the following values may be used as a check: water, at 40 C, $\Delta t_m = 20$ C, $D = 1.4$ in.; $\Delta t_m/D = 1.43$; $h = 105$.

No allowance has been made for scale and dirt films, which may offer considerable resistance in stagnant or slowly moving liquids. No allowance, likewise, has been made for the effect on the coefficient of placing tubes in a vertical row as in a cooling coil in a tank of water; a somewhat lower coefficient will probably be obtained than predicted by this equation. Perhaps, as below, the sum of the diameters in the vertical row should be used instead of the diameter of a single tube.

The coefficient for heat transfer by natural convection at vertical surfaces may be represented by an equation somewhat similar in form, proposed by Colburn and Hougen (3):

$$h = 71 \sqrt[3]{\frac{\beta \rho^2 k^3 c_p \Delta t}{Z}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [12]$$

For air, this equation agrees with that of Rice (15).

A similar alignment chart, Fig. 6, is given for estimation of the coefficient for a number of common fluids. In addition to the example shown, the following may be used as a check (compare with the example in Fig. 5): ammonia at 200 C, $\Delta t_m = 100$, $P = 10$ atm; $P^2 \Delta t_m = 10{,}000$; $h = 5.2$. Note that at 1 atm the coefficient would be only 1.1.

No scale is shown on the reference line on these two charts, since it can be seen that the function is complicated. For adding other points, the equation is best solved for two temperatures, using the appropriate properties of the substance in question, and noting where the lines cross which connect the values of h so obtained with the respective temperatures.

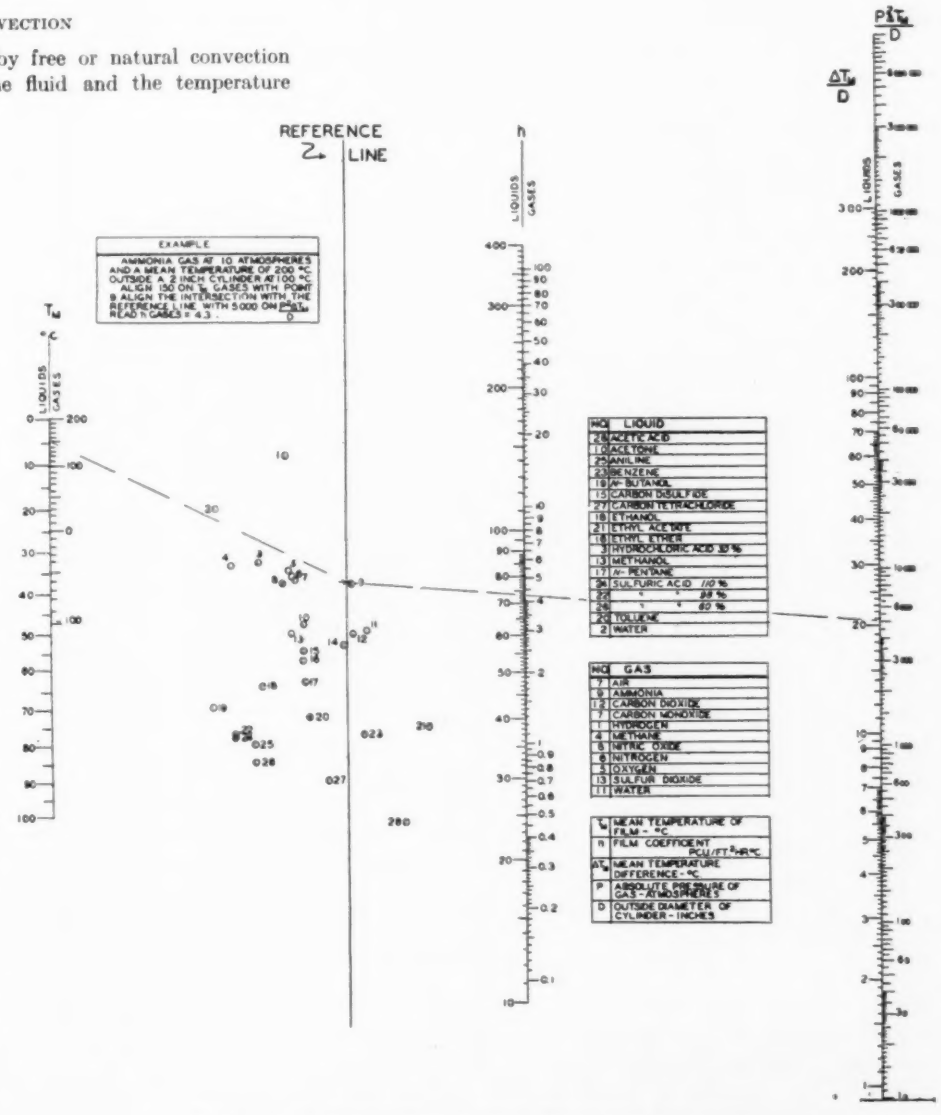


Fig. 5 Free Convection, Outside Horizontal Cylinders
[Film coefficient of heat transfer, $h = 100(\beta \rho^2 k^3 c_p/Z)^{1/4}(\Delta t/D)^{1/4}$.]

Neither of these equations includes radiation from the solid body or from polyatomic gases.

Condensing Vapors

Coefficients for the condensation or pure vapors are ordinarily high in relation to those for other resistances in heat exchangers. The following equations, due to Nusselt (10), give minimum values to be used as a guide. These equations are based on the

resistance to the passage of heat offered by the layer of condensate forming on the surface and flowing in viscous motion under the influence of gravity; they are, therefore, different for horizontal and for vertical surfaces.

For vertical surfaces:

$$h = 105 \sqrt[4]{\frac{\lambda \rho^2 k^3}{ZL\,\Delta t}} \quad \ldots\ldots\ldots\ldots\ldots [13]$$

For horizontal tubes:

$$h = 152 \sqrt[4]{\frac{\lambda \rho^2 k^3}{ZD\,\Delta t}} \quad \ldots\ldots\ldots\ldots\ldots [14]$$

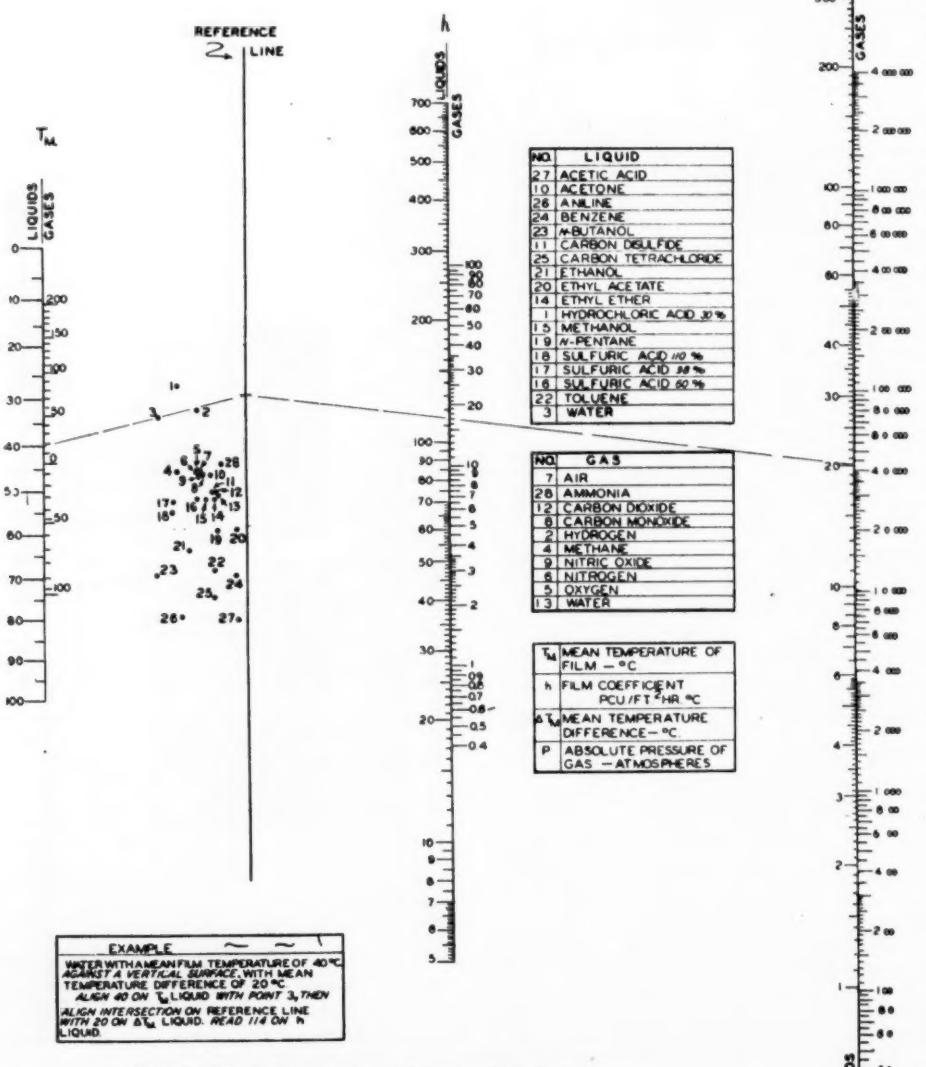


FIG. 6 FREE CONVECTION, VERTICAL SURFACES

[Film coefficient of heat transfer, $h = 71(\beta\rho^3k^2c_p/Z)^{1/3}(\Delta t)^{1/3}$.]

The value of Δt to be substituted in these equations will generally have to be estimated or obtained by trial and error, in consideration of the other resistances in series; it represents the temperature drop across the film, from saturated vapor to wall surface. The height of vertical surface, in feet, is represented by L; and the diameter of the tube, in inches, or the sum of the diameters of tubes in a vertical row, by D; the temperature called for is the average film temperature.

These equations are solved graphically on the chart, Fig. 7. An important conclusion obtained by inspection of the chart is that steam and ammonia give high coefficients, but most organic vapors give very much lower values.

Four properties are combined in the substance points on the chart. Points for other substances can be located by solving the equations for two temperatures where the properties are known.

There is evidence that these equations give conservative values for pure vapors moving at high velocity, or condensing on vertical surfaces of such height that turbulence is induced in the condensate film, though quantitative expressions for these effects are not available (8). Condensation of water on oily surfaces, forming drops instead of a continuous film, also gives high values (17).

The presence of even small percentages of non-condensing gas, however, reduces the coefficients markedly. With increasing concentrations of non-condensing gas, the resistance as calculated for the condensate layer soon becomes negligible, and the problem calls for solution on the basis of diffusion. For a theoretical treatment of this problem and empirical equations for air-steam mixtures, reference is made to the work of Colburn and Hougen (3).

ACKNOWLEDGMENT

The advice of Prof. W. H. McAdams in the selection of equations, the construction of the charts, and the preparation of this paper is gratefully acknowledged.

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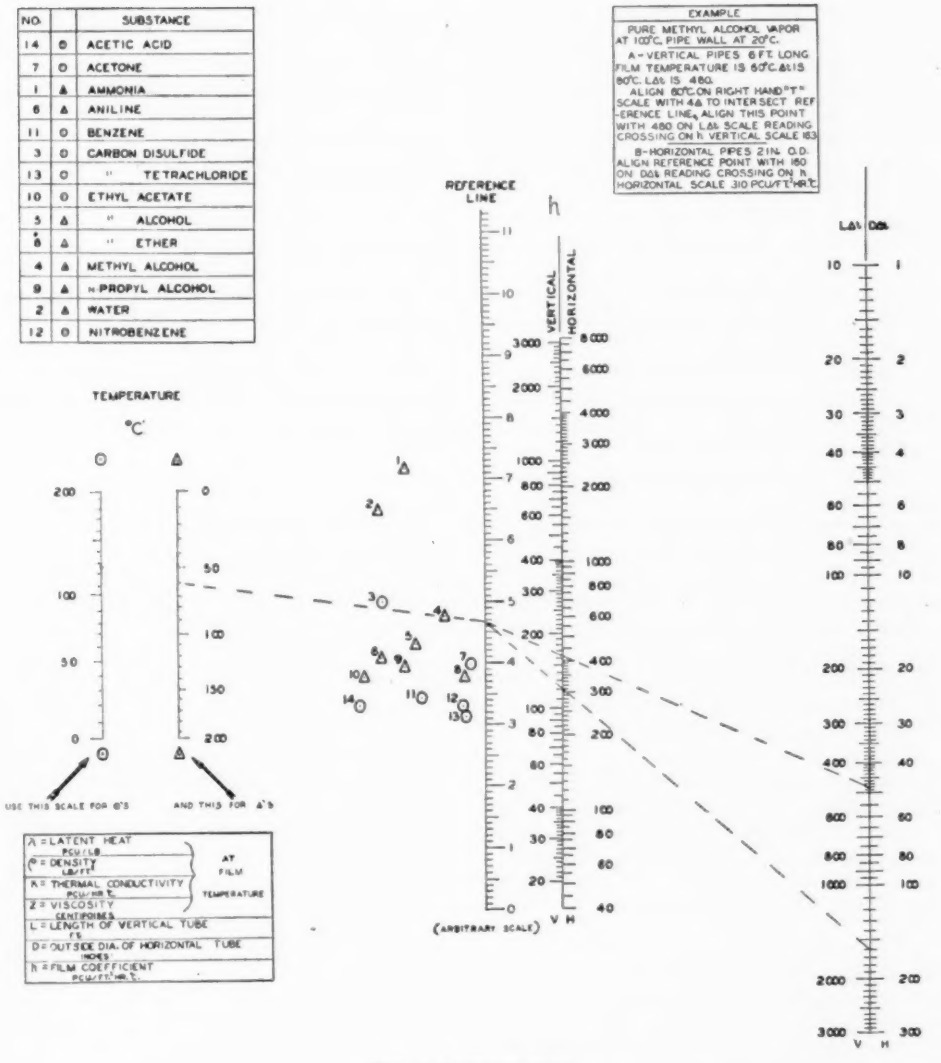


FIG. 7 CONDENSATION

[For vertical tubes, $h = 105(\lambda\rho^3k^3/ZL\Delta t)^{1/4}$. For horizontal tubes, $h = 152(\lambda\rho^3k^3/ZD\Delta t)^{1/4}$.]

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Discussion

W. W. KRAFT.[7] The authors have made a worth-while contribution to the design engineer in the charts. Work of this character has a double value. It reduces the labor involved in the calculation of optimum design of heat-transfer equipment and it simplifies the correlation of test data with sound theoretical equations. With such correlation, problems in design of heat-transfer equipment can be attacked with increasing confidence, and real advance in design should result.

W. S. PATTERSON.[8] The authors have prepared a very interesting paper and several simple and time-saving alignment charts. They refer to the equations and the charts as representing the "best available methods" for prediction of heat-transfer coefficients to be used in the design of commercial heat exchangers, but do not qualify this statement with data obtained on such apparatus. The conditions under which such equipment must operate and be tested are much different from those conditions under which most of the data were obtained from which the authors' equations and charts were developed. One of the biggest problems always facing the designer of commercial heat exchangers is to use the equations developed from data obtained in a scientific laboratory and at the same time to allow for the different conditions such as shape, size, length, material, and cleanliness of the heat transfer tubes or elements, combination of transverse and longitudinal fluid flow, combination of con-

[7] The Lummus Company, New York, N. Y.

[8] Air Heater Engineer, Combustion Engineering Corporation, New York, N. Y. Mem. A.S.M.E.

duction, radiation and convection heat transfer, presence of variable and perhaps unknown amounts of foreign material or vapors in heating or heated gases, and so on almost indefinitely.

Tests conducted by the writer's company on plate-type air preheaters with plates spaced $^1/_2$ in. and 1.0 in. apart showed that the friction loss between the plates does not vary with the mass velocity to the same power as does the pressure drop caused by entrance to and exit from the heater elements. An equation was developed, from these test results and application of the principle of dimensions, which agrees in form and gives practically the same results as the equation which would result from substitution of the authors' Equation [2] for f in their Equation [1], with the exception that the spacing between the plates was used directly in the equation instead of equivalent diameter. The end losses, however, vary directly as the square of the mass velocity and may amount to a large portion of the total pressure loss, depending on the length of the passages and the abruptness of the entrance to and exit from them. In the case of short tubes rolled into tube sheets or projecting through glands the end losses may be greater than the friction loss through the tube, which is the only portion that can be determined from the authors' equation or chart. Substituting Equation [2] for f in Equation [1], it will be found that the pressure drop due to friction varies inversely as the 1.22 power of the diameter of the tube. In an exchanger with $^1/_2$ in. internal diameter tubes or with plates spaced $^1/_2$ in. apart the accumulation of $^1/_{32}$ in. of scale or soot would therefore increase the friction pressure drop about 50 per cent. This example shows the importance of allowing for roughness and the accumulation of scale or other deposit.

The authors' Fig. 2 chart checks very closely with total transfer rates obtained on air heaters with 1-in. plate spacing, but gives values too high for the closer spacings. This is due to the fact that transfer rates on plate-type heaters as measured by the writer's company have been found to be independent of the spacing between the limits of $^1/_2$ in. and $1^1/_4$ in., which may be due to the fact that the radiation effect, though small, decreases with the spacing and tends to neutralize the effect of the spacing on the convection transfer rate.

In connection with the use of Equation [10] and the Fig. 4 chart the writer does not believe that such a method can be used for various tube spacings without applying some factor to take care of the variable ratio of tube diameter to tube spacing. This chart gives heat-transfer coefficients which are too high for economizers or boilers, in which equipment the tubes are generally not less than 2 in. in diameter and spaced not closer than 1.75 diameters, center to center.

Wherever the writer has mentioned air preheaters in the foregoing discussion, this refers to the power-plant type handling flue gas and air at temperatures not in the average case exceeding 800 F.

Authors' Closure

The authors appreciate Mr. Patterson's remarks with regard to allowance for accumulation of dirt, soot, scale, etc., on heating surfaces, and for other losses in pressure than that due to friction. These allowances will of course have to be made by the experienced engineer for the particular conditions encountered.

With regard to effect of the ratio of tube spacing to tube diameter, correlations of test data of several investigators, which we hope shortly to publish, afford a substantial check on the validity of Equation [10] for ratios of clearance between tubes to tube diameter of from about 1:6 to 4:1, a range of twenty-fourfold. We trust that Mr. Patterson will find opportunity to publish the original tests referred to on economizers, and also those on plate-type heaters with variable spacings.

PME-55-3

Semi-Luminous Flame Obtained With Heavy Domestic Fuel Oil in a Specially Designed Furnace

By P. L. MIKESKA,[1] SCHENECTADY, N. Y.

Some advantages of the semi-luminous flame for oil-burning furnaces are taken up in the early part of the paper. To assist in understanding some of the mechanics involved in obtaining the semi-luminous flame, the diagrammatic sketch, Fig. 2, labels a few of the happenings shown in Fig. 1. Fig. 2 shows four main branches, A, B, C, and D, coming off from the flame. Each of these branches has in it several blocks labeled with some of the factors involved. Branch A deals with oil atomization, primary air, and initial chemical reactions. Branch B deals with secondary combustion air, final chemical reactions, etc. Branch C explains the thermal setting for proper flame temperature. Branch D takes up the specially designed steel boiler made necessary to coordinate properly the many factors involved (such as proper size and correct proportion of the combustion chamber) to obtain the clean, quiet, efficient semi-luminous flame.



A FULLY LUMINOUS FLAME has dense gases or carbon particles in suspension, so there is danger of sooting wherever these gases or particles come into contact with relatively cool boiler surfaces; but such a flame gives off considerable heat by radiation. A non-luminous flame, on the other hand, permits the combustion reactions to be more nearly completed before the gases reach the combustion-chamber outlet; but this flame has very small capacity for giving off heat by radiation.

It was desired to incorporate in oil-burning furnaces the non-sooting characteristics of the non-luminous flame and the good radiation characteristics of the luminous flame. The semi-luminous flame, pictured in the section through the oil furnace (Fig. 1), does this. Since the flame has good radiation characteristics, less secondary heating surface is required and the boiler is lighter as well as less expensive.

An inspection of the flame will tell different stories to different people. Many will think of nothing more than oil, air, and the flame they see. Some, of course, think of the many problems that were involved in the development of such a flame. Then, too, there is the group that has considered the fundamental importance of the combustion efficiency of the sources that provide for our human comforts; but they, perhaps, have given little thought to the problems involved. To assist in understanding some of the mechanics involved in obtaining the semi-luminous flame, the diagrammatic sketch (Fig. 2) labels a few of the happenings shown in Fig. 1. Fig. 2 shows four main branches, *A*, *B*, *C*, and *D*, coming off from the flame. Each of these branches has in it several blocks labeled with some of the factors involved. Branch *A* deals with oil atomization, primary air, and initial chemical reactions. Branch *B* deals with secondary combustion air, final chemical reactions, etc. Branch *C* explains the thermal setting for proper flame temperature. Branch *D* takes up the specially designed steel boiler made necessary to coordinate properly the many factors involved (such as proper size and correct proportion of the combustion chamber) to obtain the clean, quiet, efficient semi-luminous flame.

Some Advantages of the Semi-Luminous Flame

Reference has been made to the facts that the semi-luminous flame is non-sooting and is of good radiation characteristics. With a luminous flame there is the probability of a combustion roar if all the combustion air is admitted through a nozzle along with the oil; non-luminous and semi-luminous flames, if properly handled, can be burned with quietness. This quiet flame is further possible because of the symmetry, which does not permit pockets in the combustion chamber where small explosions might take place, resulting in noisy as well as inefficient combustion.

Progressive combustion, symmetry of the whole arrangement, and proper coordination of the various design factors make possible not only a clean, quiet flame, but one that is highly efficient, for it becomes possible to control many of the different variables that enter into combustion, even the combustion air. Such things as draft variations and the like do not materially affect the operation. Some of the well-known requirements of proper combustion are:

1 Enough air should be admitted to assure sufficient oxygen for complete combustion.

2 The combustible should be admitted in as finely divided a state as practicable, and should be admitted at the proper time and place.

3 Complete combustion should be obtained with a minimum of excess air.

4 Air should be admitted at the proper time and in such a way that the oxygen of the air comes into intimate contact with the combustible.

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Contributed by the Utilization Committee of the Petroleum Division and presented at a meeting, New York, N. Y., March 15, 1933, of the Metropolitan Section of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

5 The gaseous mixture must be maintained at or above its ignition point until combustion is complete, by making use of the proper thermal setting.

6 Combustion space of adequate size and proper shape must be provided so that the gases can expand in the proper fashion during the combustion period.

An examination of these various points will reveal that there are many ways to burn oil. Knowing what had been done in a general way with oil combustion, effort was concentrated on trying hundreds of arrangements to burn oil. Finally the semi-luminous flame, employing the impact-expansion principle in two main stages, together with progressive combustion, was adopted.

Fuel oil must have its area enormously increased or be in a gaseous form in order to burn. This can be checked by throwing a lighted match into a small open vessel of the heavier domestic fuel oil; the oil will extinguish the match flame. If the same oil is broken into particles with a small atomizer, however, it can be ignited with a match.

Atomization, Primary Air, and Initial Chemical Reaction

Of the four main branches shown in Fig. 2, branch *A*, which deals with oil atomization, primary air, and initial chemical reaction, will be taken up first.

Motor Compressor. The oil-burner mechanism consists of a motor compressor and burner head, as shown in Fig. 3. A section of this mechanism is shown in Fig. 4. The whole unit is mounted on top of the boiler with a single nut, as shown in Fig. 1.

The driving power for the motor compressor, Fig. 5, is furnished by a quiet induction-type motor. The motor, with hardened and ground shaft, drives a multi-vane fan, the oil pump, and the air compressor without employing any intermediary gears or belts. The small eccentric pump near the lower end of the shaft is an oil pump; the larger eccentric pump just above it is the air compressor. The air compressor takes a small part of the air drawn in by the blower and compresses it for use in the nozzle; this will be taken up in more detail later. The lower part of the motor compressor forms a pressure chamber which contains both oil and air under the pressure necessary to force them through the burner head for atomization. Within this chamber is a float valve which automatically maintains the correct level of the oil in the chamber. Near the motor-compressor oil outlet to the nozzle is a valve for adjusting the rate of flow of oil to the nozzle. The oil pump was designed to produce sufficient suction for drawing in the oil from a storage tank. The pump is capable of producing sufficient negative pressure (suction) in the oil-supply line to raise the oil 15 ft, and the oil pump is self-priming.

Screen Anti-Siphon Valve and Surge Chamber. Because of the suction produced by the pump in the motor compressor, the oil from the tank is drawn through a screen anti-siphon valve, which is placed between the burner and the oil-storage tank. The screen valve is to prevent the siphoning back of the oil and to remove any dirt that may be in the oil. After passing the screen valve, the oil is drawn through the supply line to the motor compressor until it reaches a surge chamber located just ahead of the entrance of the oil line into the motor compressor. This chamber absorbs any shocks caused by pulsation that might come from variations in the flow of oil in the supply line, thus eliminating any noise from this source.

Some of the Main Parts and Functions of the Motor Compressor. Referring to Fig. 5, the oil after passing through the surge chamber enters the motor compressor unit and is conveyed from that point through a tube within the pressure chamber to the oil pump. Only a small part of the oil passing through the oil pump comes from the suction line, since a bypass valve is used on the intake side of the pump. This bypass valve, which is float controlled, regulates the proportion of oil taken from the pressure chamber and the supply line. The major part of the

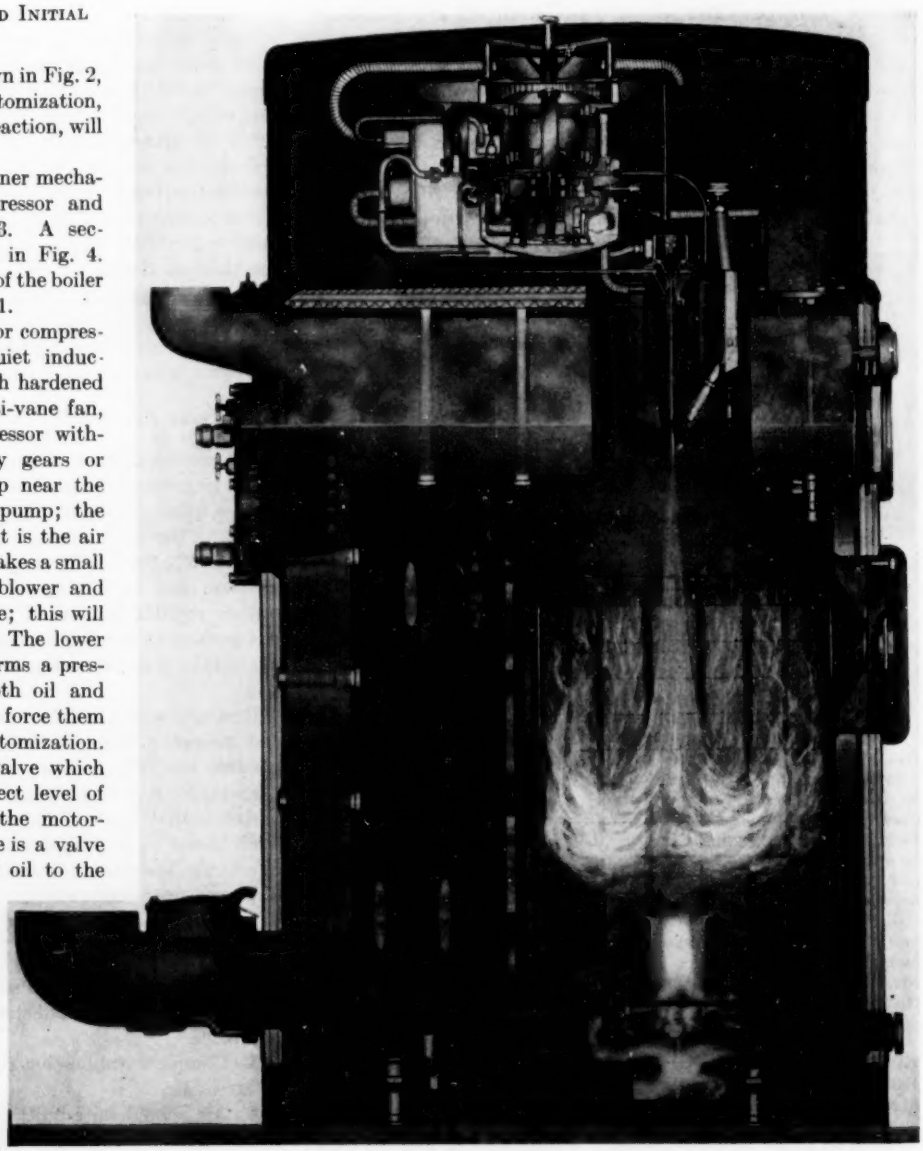


Fig. 1 Section Through Oil-Burning Heating Furnace, Showing Semi-Luminous Flame

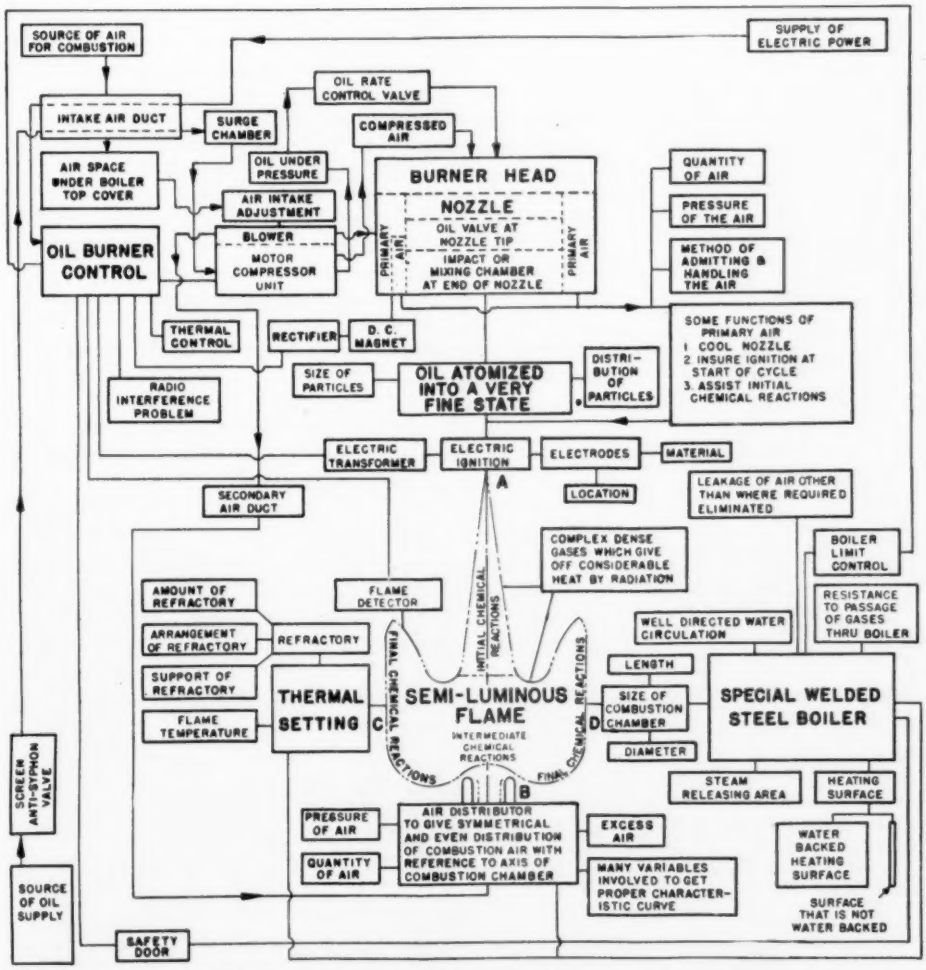


FIG. 2 DIAGRAMMATIC SKETCH OF THE SEMI-LUMINOUS FLAME
(Obtained with heavy domestic fuel oil in a specially designed furnace.)

oil is taken from within the screen of large area mounted below the oil pump in the pressure chamber. The oil is pumped out through an exhaust check valve to the main part of the pressure chamber outside the screen.

The check valve or exhaust valve is of the spring-loaded diaphragm type, which is simple in construction, reliable in operation, and designed to operate at 2 lb per sq in. gage. The two main functions of the check valve are to prevent air from being forced back to the oil pump, which would reduce the priming ability of the pump, and to prevent foaming of the oil in the pressure chamber when air is drawn from the supply line and discharged from the pump during priming. If the oil level in the pressure chamber is high enough to cause the float to open the bypass valve, no oil is taken from the supply line and the full capacity of the oil pump is used in circulating oil within the pressure chamber through the large area screen. During steady operation, the float maintains an oil level in the pressure chamber at a point where the oil passing through the pump is divided between that drawn from the supply line and from the pressure chamber, in such a way that the oil from the supply line equals the amount delivered to the nozzle.

The air compressor previously mentioned discharges into the space in the pressure chamber above the oil. Hence a uniform pressure exists throughout the pressure chamber, and it is by means of this pressure that the oil and compressed air are forced to the nozzle of the burner head.

The oil that feeds to the burner nozzle flows through a tube which extends below the level of the oil in the pressure chamber, as shown in Fig. 5. This tube leads into a valve by which the rate of flow of oil to burner nozzle is regulated. The oil then flows through a hole drilled in the main casting, and thence outward to the copper tube which leads to the burner nozzle.

Air Vent in Oil Line. In the vertical path of the oil just beyond the oil valve is a small vent hole. This vent leads to the chamber under the motor and serves to remove any included air from the oil after it passes the oil-rate adjusting valve. A small amount of oil also flows out through the vent, but this oil serves as a seal and lubricates the air compressor.

Lubrication of Motor Compressor. In fact, the fuel oil is used to lubricate all the parts of this mechanism automatically—it can be seen that oil is supplied to the bearings under pressure. Further examination of Fig. 5 will show that the shaft on which the rotating parts are mounted has a hole drilled up the center. Since the lower end of the shaft ex-

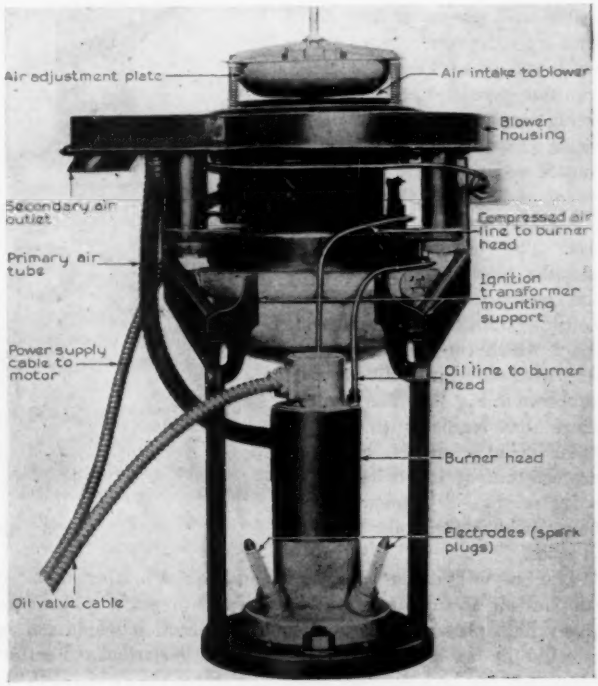


FIG. 3 OIL BURNER, FRONT VIEW

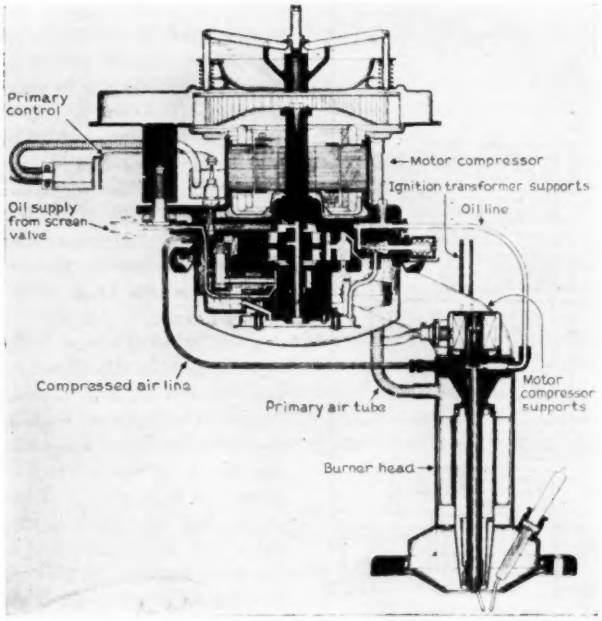


FIG. 4 OIL BURNER, SECTIONAL VIEW

tends below the oil level in the pressure chamber, oil is forced through this hole by the pressure in the chamber. This serves to lubricate under pressure all the bearings, and the air compressor through radial holes drilled in the shaft.

Intake Air Duct. The air used for combustion enters through louvers in the right-hand air duct at the side of the boiler and passes up the air duct into the space under the boiler top cover (Fig. 6). The resulting circulation of the entering air maintains a cool ambient temperature for the oil burner and the control shown on top of the boiler (Fig. 1).

Adjustment of Combustion Air. The combustion air enters the blower at the top of the motor-compressor blower housing through an annular space which can be varied in size by an air-adjustment plate, as shown in Fig. 3. The small shaft that regulates this annular opening to the blower projects out of the boiler top cover so that adjustment can be made with the boiler cover in place.

the unit by the natural draft of the chimney when the oil furnace is shut down. Thus the loss of stored heat in the boiler is very low during "off" periods. This, of course, is highly desirable where a boiler may be used periodically—as, for instance, in heating domestic hot water and the like.

As shown in Fig. 2, the blower supplies air through two separate paths. The smaller part of the two volumes, called primary air, blows through a flexible metal hose connecting the blower housing and the burner-head housing. Less than 10 per cent of the air used for combustion goes through this path.

The rest of the air supplied by the blower is conveyed through the secondary air duct to the air distributor located in the center line of the combustion chamber, as shown in Fig. 6. The details of the air distributor will be covered in the description of branch *B*.

The main functions of the primary air are to cool the nozzle, to supply enough oxygen so that the oil can be ignited, and to take part in the initial chemical reactions. More detailed information on these items appears later.

The air regulator (Fig. 3) at the blower intake controls both the primary and the secondary air in the correct proportion, regardless of what the oil rate may be set for in gallons per hour.

Air Supply for Air Compressor. In addition to the primary and secondary air which leaves the blower directly, there is a small amount taken from under the blower housing and drawn

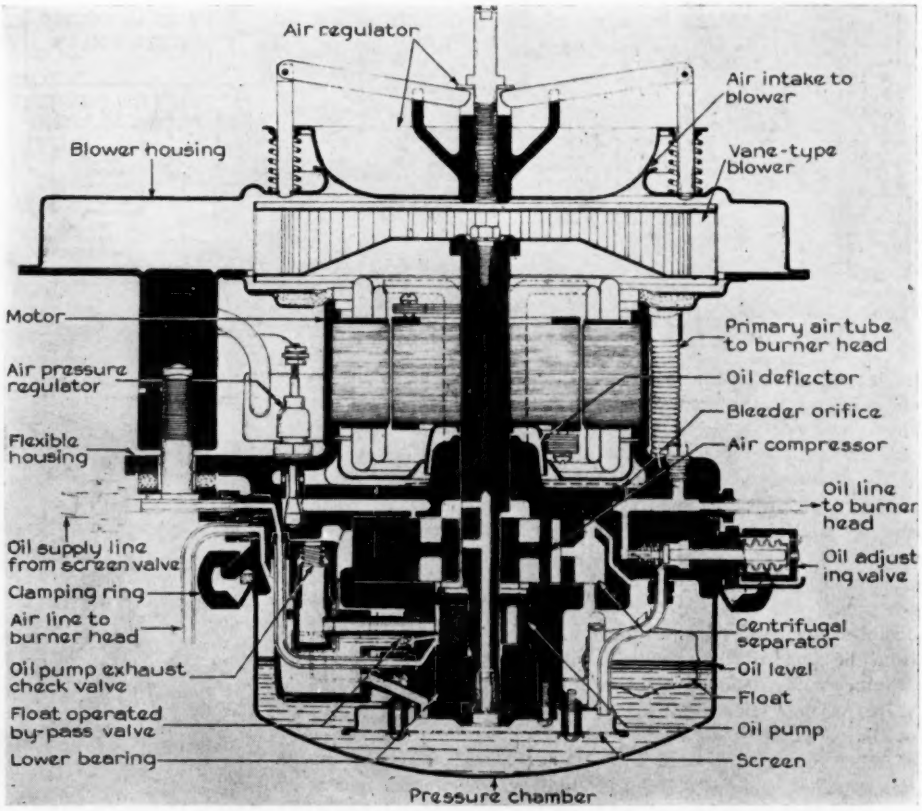


FIG. 5 SECTIONAL VIEW OF OIL-BURNER MOTOR COMPRESSOR

The blower is of the vane type, and is quiet in operation. It supplies air at from 1.5 to 1.75 in. water pressure. This relatively high pressure was used so that operation would not be affected by such things as slight draft variations. Further, because of the high pressure required to force the air through the system, only very small quantities of air can be drawn through

down through the motor stator and rotor to the air-compressor intake. The intake to the air compressor is located in the space immediately below the motor winding, and is so arranged that the oil passing through the upper motor bearing, as well as the oil which passes through the vent in the oil-feed line, is taken into the air compressor and delivered back into the pressure chamber

along with the air. The purpose of the vent, as previously mentioned, is to remove air from the oil line. The air compressor is of the rotary eccentric type (Fig. 5). A view of it on its shaft is shown in Fig. 7.

Centrifugal Oil and Air Separator. The air compressor discharges the mixture of oil and air into a centrifugal separator (see Fig. 5), which is located in the cylinder (housing) of the air-compressor eccentric pump. The high velocity of the air and oil leaving the air compressor sets up a rapid swirling motion of the discharge into the separator, effectively separating the oil from the air. The oil naturally flows out through a small opening in the bottom of the separator, and the air goes through an opening in the top of the separator. The main object in using the centrifugal separator is to prevent an accumulation of oil in the air passages through the nozzle, which might result in erratic operation of the burner. The discharge of air from the separator is directly into the pressure chamber. The compressed air in the pressure chamber is carried to the nozzle in the burner head (see Fig. 4) by means of a small copper tube.

Compressed Air Used for Atomization. The pressure of the compressed air varies with the oil rate. For a given oil rate the pressure is kept substantially constant by a simple pressure regulator. The compressed air line is taken from the top of the pressure chamber to the burner head by a small copper tube shown in Fig. 4. Before the nozzle valve in the burner head opens, 8 to 10 lb per sq in. gage is the pressure in the pressure chamber. After the nozzle valve opens, the compressed-air pressure is approximately 12 lb per sq in. gage for the 1¼ gal oil rate and 15 lb per sq in. gage (approximately) for the higher oil rate, around 2 gal per hour.

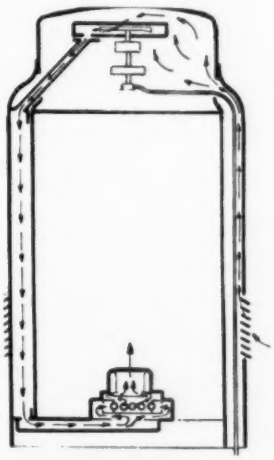


FIG. 6 DUCTS FOR CONVEYING AIR

Atomization of Oil in the Nozzle. The paths of the oil and air through the motor compressor have been traced through up to the nozzle. The purpose of atomization in the combustion of a hydrocarbon oil is to increase enormously the surface area of a given quantity of oil, thereby accelerating the change of the liquid into the gaseous stage. The radiant-heat energy from the flame playing upon the enormously increased surface of the small particles of oil later vaporizes it and changes it into a gasous stage.

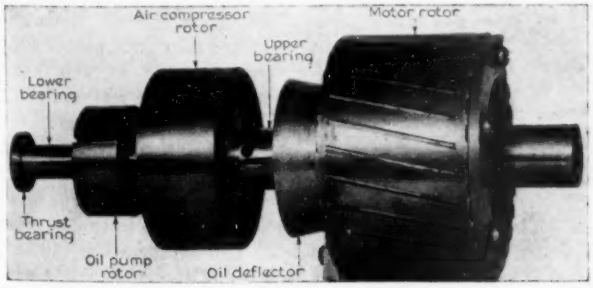


FIG. 7 AIR-COMPRESSOR OIL PUMP AND MOTOR ROTORS
(Mounted on the same shaft.)

Oil Valve at Nozzle Tip Operated by D-C Magnet. The burner head which houses the nozzle is shown in Fig. 8. Some of the things which will be noticed are the oil valve within the nozzle which is used for shutting off the oil when the burner is stopped. The oil valve shuts the oil off right at the nozzle tip, so that oil dribble is eliminated. Oil dribble, resulting in drops of oil adhering to the nozzle, will frequently (because of heat from the flame which causes the nozzle to operate at too high a temperature) break down and form a carbon deposit, which will upset the performance of the nozzle. An enlarged view of the end of the nozzle (Fig. 9) shows that, after the oil is shut off at the end of the oil orifice, compressed air continues to flow, because of the flywheel effect of the motor compressor. This compressed air blows off the small amount of oil that may have remained. When the burner is shut down, a spring keeps the oil valve closed. However, when the control (see wiring diagram Fig. 10) calls for burner operation, the magnet lifts the oil valve against the action of the spring. Hence, a power failure would immediately

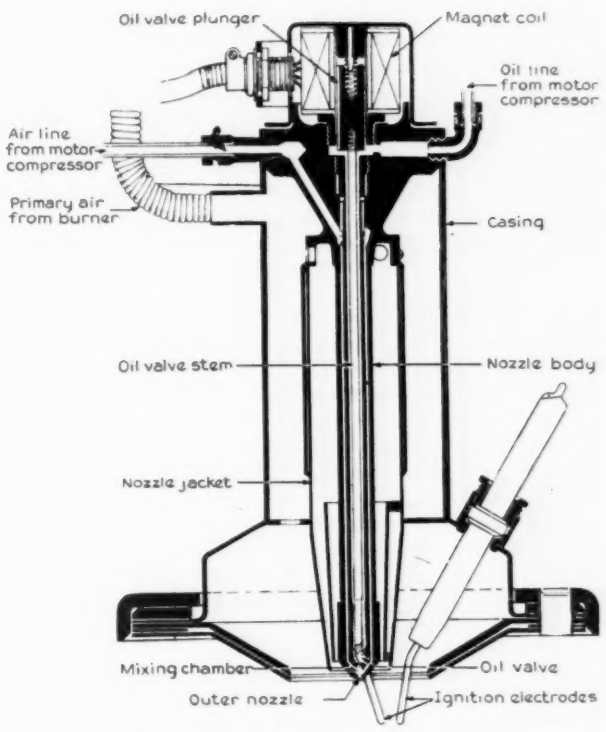


FIG. 8 SECTIONAL VIEW OF OIL-BURNER HEAD

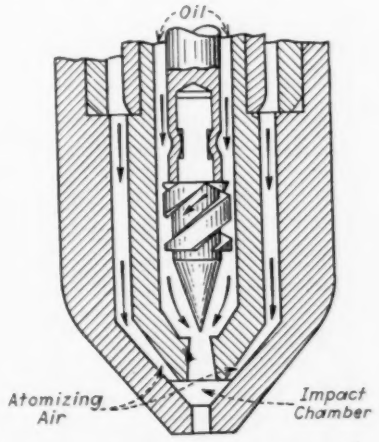


FIG. 9 ENLARGED VIEW OF LOWER PART OF NOZZLE

shut the burner off right at the nozzle tip and not permit any of the oil to dribble, which might cause carbonizing at the nozzle and "after fire" in the combustion chamber.

Impact Chamber Located Near the End of the Nozzle. The oil entering the nozzle passes down along the valve stem until it comes to the valve-stem guide, which has a spiral groove on it to give the oil a rotating motion. A much enlarged view of the end of the nozzle is shown in Fig. 9. The rotating stream of oil then passes out through the oil nozzle into a small impact or mixing chamber. Compressed air enters the mixing or impact chamber, as shown in Fig. 9. Thus the compressed air (the air is also at a high velocity) impacts or bombards the small stream of oil. This provides complete mixing of the oil and air, as well as breaking the oil into small particles.

The mixture of oil and air leaves the mixing chamber, discharging through the final nozzle orifice, which is approximately 50 times as large as the conventional small nozzle orifices of about 8 mils diameter.

Discharging of the compressed air in the manner described results in an exceedingly fine atomization.

Further, the air passage that supplies the compressed air to the impact chamber is so arranged that a whirling motion is given to the air as it mixes with the oil, so that centrifugal force resulting from the high speed of rotation of the mixture leaving the final orifice aids still further in breaking up the oil.

Oil Atomized Into a Very Fine State. Laboratory tests were made by observing these small oil particles by scattered light and measuring the rate of fall in still air. From the rate of fall the diameter of the drop was then calculated by means of Stokes's law. It was found that a drop of oil no larger than the head of a match that passed through this nozzle was broken up into millions of particles. A microphotograph of some of these particles coming from the nozzle is shown in Fig. 11.

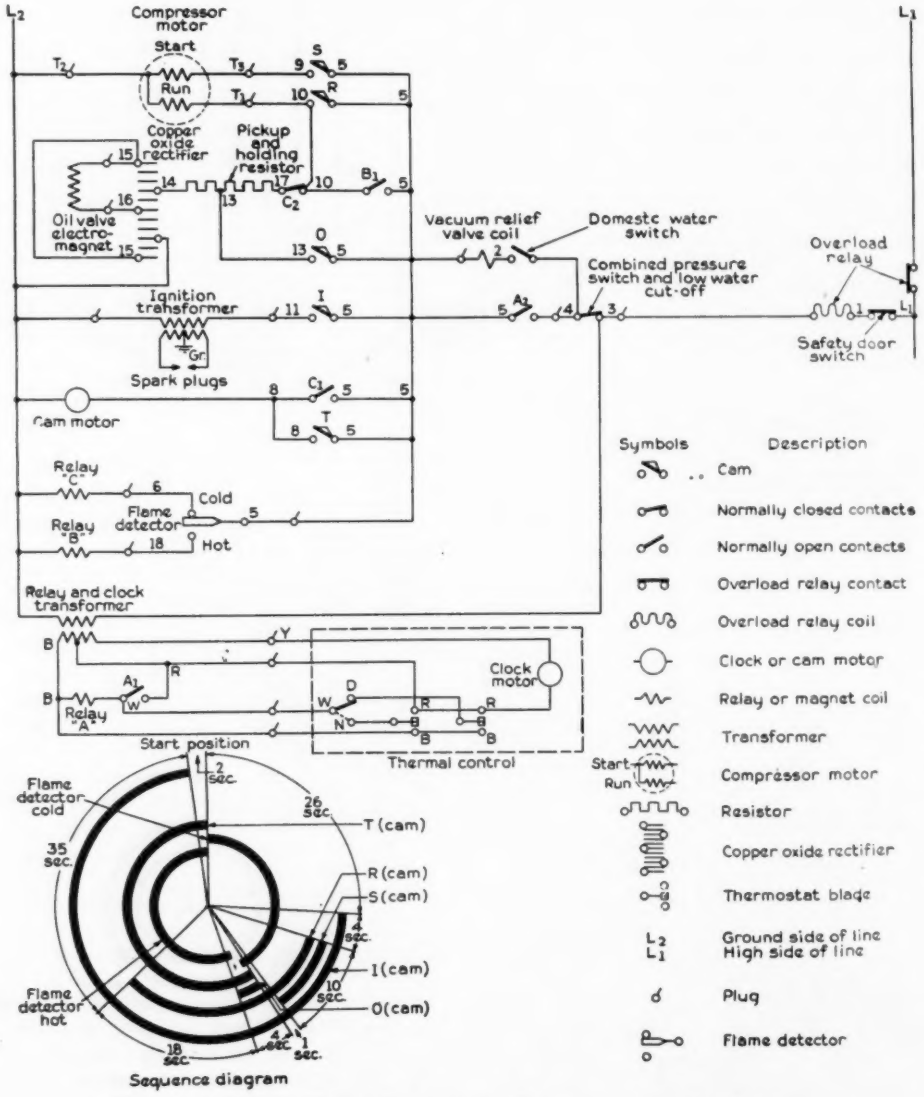


FIG. 10 SCHEMATIC WIRING DIAGRAM FOR CONTROLLING THE OIL FURNACE

Motion-Picture Study of Atomization During Development. At one stage of the development there were several designs of nozzles which were reasonably good. In order to be able to study the problem under little better conditions than being in a room filled with oil vapor and having to breathe it, a slow-motion-picture study was made of the nozzle performance without flame. Later it was possible to study these without the aforementioned handicap.

Electric Ignition of Oil and Initial Chemical Reactions. The finely atomized oil and air jets sweep past the ignition electrodes, which are placed just out of the path of the jets previously mentioned. The ignition spark is supplied by a 12,000-volt transformer which is grounded at the mid-point, and ignition is held on for 72 sec at the start of each cycle. The details of the wiring diagram of the control are given in Fig. 10. Ignition is made possible by the primary air which is blown downward around the nozzle, part of which passes inside of the nozzle jacket and the rest on the outside of the nozzle jacket.

Atomization, as mentioned previously, enormously increases the surface area of the oil. The heat produced by the spark from the electrodes is sufficiently intense and at a high enough temperature (1000 to 1500 F is necessary) to vaporize some of the small particles of atomized oil in the immediate vicinity of the spark and thereby initiate combustion. The moment that this has occurred, the few ignited particles of oil produce sufficient heat to vaporize the neighboring particles of oil and raise the temperature to a sufficiently high value to ignite them, which is possible because of the oxygen present in the primary air. Progressively, the flame burns downward slowly in what might be termed the ini-



FIG. 11 MICROPHOTOGRAPH OF SOME OIL PARTICLES COMING FROM OIL-BURNER NOZZLE

(These were sprayed on a ruled-glass slide, the squares being 0.004 in. on each side.)

tial chemical reaction, since the primary air contains only a very small percentage of the air necessary to burn the oil completely.

Cooling of Nozzle. Aside from making ignition positive, the primary air keeps the nozzle cool. Later this primary air takes part in the initial chemical reaction of combustion.

The design of the nozzle should be such that its maximum operating temperature is considerably lower than the temperature at which carbonizing takes place. The nozzle in question was so designed that during maximum load, as well as during shutdown (while the refractory is red hot), the temperature of the nozzle will at no time be anywhere near the point at which the oil breaks down and leaves a carbon deposit.

Method of Admitting Primary Air. The primary air is distributed symmetrically and evenly with reference to the axis of the combustion chamber and must be admitted correctly to get the proper initial chemical reaction to obtain the quiet, semi-luminous flame. Unsymmetrical admission of primary air would cause a deflection of the oil and air jets, which would produce inefficient combustion and a noisy flame. The primary air unites with the atomized oil and air jets and is ignited by the heat from the spark (previously described), combustion then starts, and a visible flame is produced.

SECONDARY COMBUSTION AIR FOR INTERMEDIATE AND FINAL CHEMICAL REACTIONS

Next in order is branch *B* of Fig. 2, which deals with secondary combustion air for the intermediate and final chemical reactions.

Secondary Air Duct. The secondary combustion air makes its way from the oil-burner blower to the bottom of the boiler, as shown in Fig. 6. Before the air enters the combustion chamber it has to pass through an air distributor, the location of which with reference to the boiler is shown in Fig. 1. Some of the necessary factors are the symmetrical and even distribution of air with reference to the axis of the combustion chamber.

Characteristic Curve of Air Distributor. Exploring with a pitot tube across the air distributor (refractory nozzle removed) gave the curve shown in Fig. 12. Exploring with the same tube at right angles to the first path gave readings such that only one curve had to be plotted—the other would be superimposed on the one shown. The curve demonstrates how the air distributor handles the air.

Design of Air Distributor. Such a curve is obtained by allowing the air to distribute in two directions as it enters the air-distributor housing, after which it passes through 12 holes into another annular chamber. A second time the air is made to pass through 12 holes placed out of phase with reference to the first set, so that the air strikes a metal strip and is deflected in two directions before entering the last set of holes. It then rises vertically through an orifice 4 in. in diameter and hits a circular plate. This plate deflects the air through an annular opening. The air then travels laterally through a number of vanes which give it a slight whirl, after which it passes out through the final orifice of the air distributor. The air then goes through a refractory nozzle before it is available for combustion purposes.

The foregoing may seem an intricate way of handling the air, but it is necessary in order to obtain the best performance.

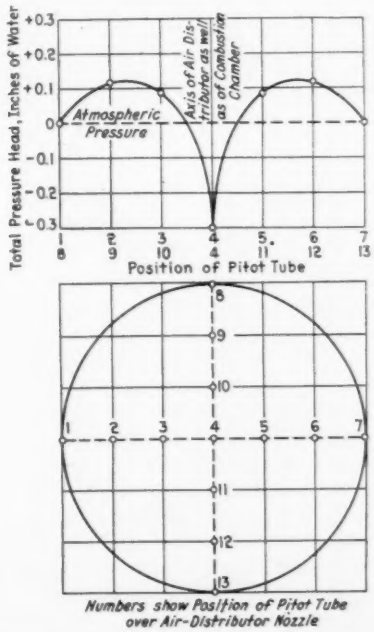


FIG. 12 CHARACTERISTIC SECONDARY-AIR PRESSURE CURVE

(For air distributor without refractory nozzle. This shows the symmetrical distribution of the air with reference to the axis of the combustion chamber.)

Aside from the problem of getting the air evenly distributed, there were many variables involved which complicated the problem, such as the angle and depth of the vanes and the diameter of the final orifice in the air distributor. Then placing the refractory nozzle brought in two more variables: the nozzle height and the shape of the end of the refractory nozzle.

The characteristic curve shown in Fig. 12 is necessary in order to obtain the semi-luminous flame at its best. All the aforementioned variables play their part in determining this curve.

With the refractory nozzle located on top of the air distributor, the characteristic curve is flattened out somewhat.

For purposes of discussion, three stages of combustion were selected, and these were designated as initial, intermediate and final chemical reactions. However, the mere symmetry of the whole thing will indicate that the combustion is progressive, and only for convenience during discussion were the three stages selected.

The nitrogen remaining from the air brought in to supply oxygen for combustion has many things to its discredit; however, there may be one thing to its credit—it possibly acts as a cushion which helps in securing a quiet flame.

Intermediate and Final Chemical Reactions. The secondary air impacts with the vaporized oil, air, etc. that had just previously

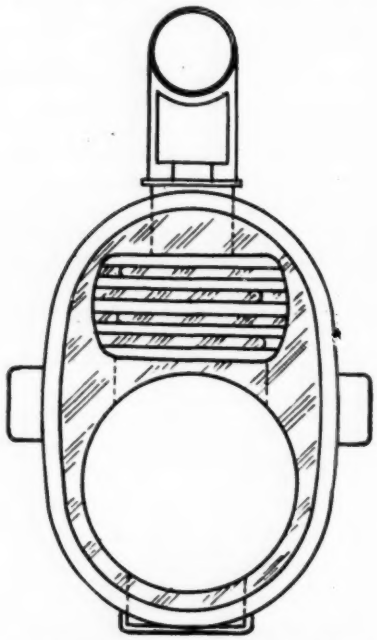


FIG. 13 COMBUSTION CHAMBER AND TUBE DRUM
(Surrounded by water on top, sides, and bottom.)

entered into its initial stages of combustion. This mixture now, at the proper time and place, meets the oxygen it needs to complete the combustion. The air supplying the oxygen as previously described is symmetrically and correctly distributed.

The exit of the combustion gases is out near the top of the combustion chamber; the primary air and atomized oil feed along the axis of combustion chamber at the top; the secondary air feeds from the bottom of the boiler, also symmetrically with reference to the axis of the combustion chamber. The products of combustion therefore rise nicely up the sides of the combustion chamber to exit out of the lateral pass. The mixture in rising up the combustion chamber wall obtains enough oxygen, well controlled, to complete the final chemical reactions. These chemical reactions are taking place under ideal thermal setting, which will be taken up in the description of branch *C* of Fig. 2.

THERMAL SETTING

The thermal setting of all this progressive combustion is an important point; hence, branch *C* logically follows.

Flame Temperature. For many reasons it is desirable to get as high a flame temperature as is practicable, since a larger amount of heat can be picked up in the combustion chamber because of the well-known Stefan-Boltzmann law. Aside from that, the higher flame temperature aids immensely in supporting the various chemical reactions and is of greatest importance to the final ones.

To carry out the final chemical reactions as they take place along the sides of the combustion chamber, such a temperature must be maintained that the rising oxygen can be practically all used and combustion made complete.

Refractory Bricks and Their Support. The proper flame temperature can be obtained by the use of the right amount of refractory. There is a practical limit to the amount of refractory necessary, which can best be determined by experiment. Refractory, especially in large pieces, has a tendency to crack or break when subjected to thermal abuse, so the refractory was made into small bricks and placed on supporting rods like beads on a string, as shown in Fig. 1. The expansions and contractions are small, and the bricks have ample freedom. These rods supporting the bricks are suspended from a refractory supporting ring welded to the upper end of the combustion chamber. The rods are of stainless iron with a high percentage of chrome. The refractory bricks in reality float as a sort of a sheet, leaving a space between the bricks and the water-backed combustion chamber wall. This general arrangement is highly desirable from a heat-transfer standpoint, since a large amount of the radiant heat can strike the water-backed walls from directly back of the strings of bricks. The refractory bricks very quickly become red hot and, in turn, radiate a great amount of heat to the water-backed combustion chamber. As a matter of fact, tests that have been made indicate that approximately 75 per cent of the heat is picked up in the combustion chamber. Of further interest is the fact that the combustion chamber has only 30.2 per cent of the total heating surface in the boiler.

This refractory arrangement makes it possible for the final combustion reaction to take place while the flame is rising up the side of the combustion chamber, thus giving at all times ample thermal support.

SPECIAL WELDED STEEL BOILER

The nature of the three branches, *A*, *B*, and *C* of Fig. 2, and the symmetry of the whole unit show why a special boiler, explained in branch *D*, is needed.

General. For a furnace that is to give quiet, complete, and highly efficient combustion, the size and shape of the combustion chamber are of great importance. The two main dimensions are the diameter and the length of the combustion chamber. Next in the order of importance is the matter of resistances to which the gases are subjected in their travel through the boiler.

The boiler with down draft was finally adopted because of its better adaptability to the many factors involved. The downdraft boiler is inherently less affected by draft variation than the other types.

In the construction of the boiler, use was made of the well-known and efficient counterflow principle as used in power-plant boilers; that is, hot gases from the combustion chamber pass into the top of the tube drum where the water is the hottest and then passes downward over the water tubes (adiators), becoming cooler as they come in contact with surfaces backed by the cooler return water at the bottom of the boiler.

The vertical section through the boiler is shown in Fig. 1, and a horizontal section through the boiler is shown in Fig. 13. The boiler proper is of arc-welded heavy steel plate, and upon completion is essentially one piece.

The amount of heating surface, the steam-releasing area, etc. are some of the simpler problems involved.

Boiler Output and the Amount of Heating Surface Used. The boiler (Fig. 14) has a nozzle output of 257,000 Btu per hour, or 7.68 bhp. The total heating surface of the boiler is 60 sq ft,

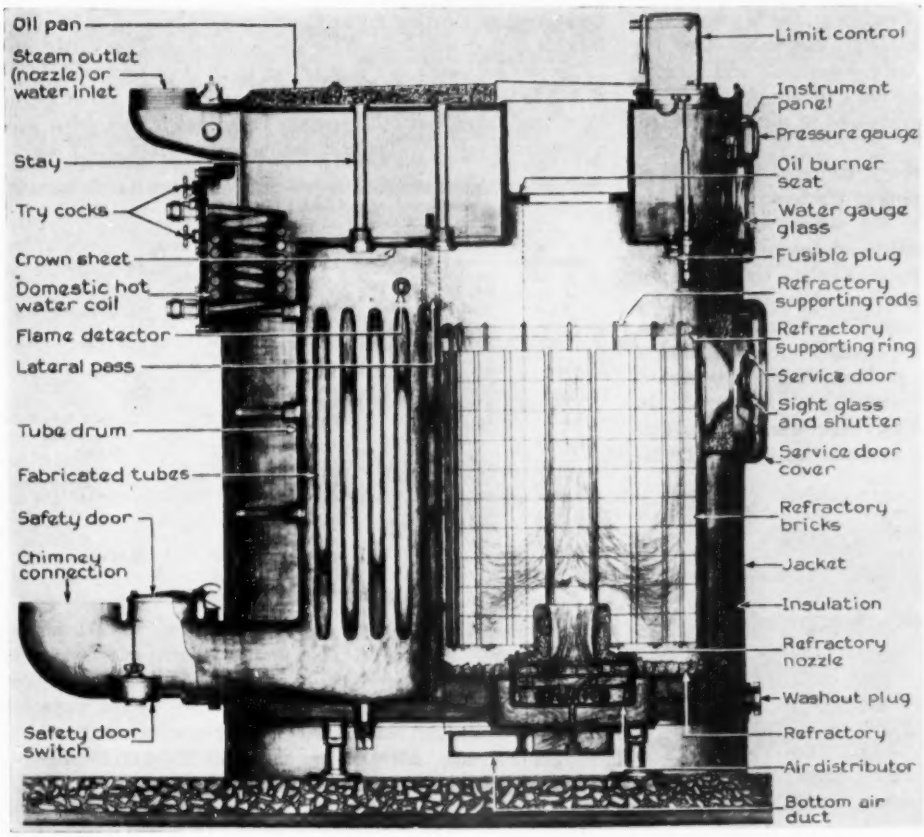


FIG. 14 SECTIONAL VIEW OF BOILER

of which 57.6 sq ft, or 96 per cent of the heating surfaces exposed to the combustion gases, is water backed. The volume of the combustion chamber (less volume occupied by refractory) is 5.2 cu ft. Hence this means a heat release of over 60,000 Btu per cubic foot of combustion space per hour.

Combustion Chamber and Tube Drum. The two main parts of the boiler are the combustion chamber and the tube drum. These are surrounded by an outer shell, together with the top and bottom of the boiler. The boiler was made elliptical (slightly modified) for convenience in handling in close quarters and for beauty. The width is such that it can be carried through an ordinary doorway.

A seat is provided for the burner head at the top of the combustion chamber, and the air distributor is at the bottom of the combustion chamber, as shown in Fig. 14. The refractory bricks are installed through a service door on the front of the boiler and are held within the combustion chamber by a refractory supporting ring which is welded to the upper end of the combustion chamber, as shown in Fig. 14.

The tube drum contains four fabricated tubes or "adiators" (see Fig. 1). Easy access is gained to the tubes for purposes of cleaning through a clean-out door opening. At the bottom of the tube drum is a rake-out door. The chimney connection with safety door (shown in Fig. 14) is attached to the back of the boiler at the bottom of the tube drum. If this door is open for any reason, it operates a switch which shuts down the burner. If a combination of failures should occur which would cause a "puff" or pressure to build up in the combustion passages, this door will open to relieve the pressure and at the same time shut off the power to the burner.

Boiler Accessories and Control. There are the usual little fixtures that go to complete a boiler, such as the fusible plug, water gage, steam gage, low-water cutoff (a part of the boiler-limit control which protects the crown sheet), etc.

It may be of interest to show how the low-water cutoff operates. The limit control which contains the low-water cutoff has a bulb containing turpentine, which is inserted into a well and which projects a little above the crown sheet. If the boiler-water level drops to a certain point above the crown sheets, the water quickly evaporates out of this well and causes the temperature of the bulb to rise. Expansion of the liquid in the tube causes a sylphon to expand and to break the main circuit. As soon as water is added to the boiler, the well is again filled with water, which cools the liquid in the bulb. This cooling causes the sylphon to contract and to make contact so that the burner can again operate. The arrangement of the low-water cutoff in the control circuit is shown in Fig. 10. The location of the limit control on top of the boiler is shown in Fig. 15. The water-gage glass, of the type used on some locomotives, permits flush mounting of the gage and adds to the appearance of the unit.

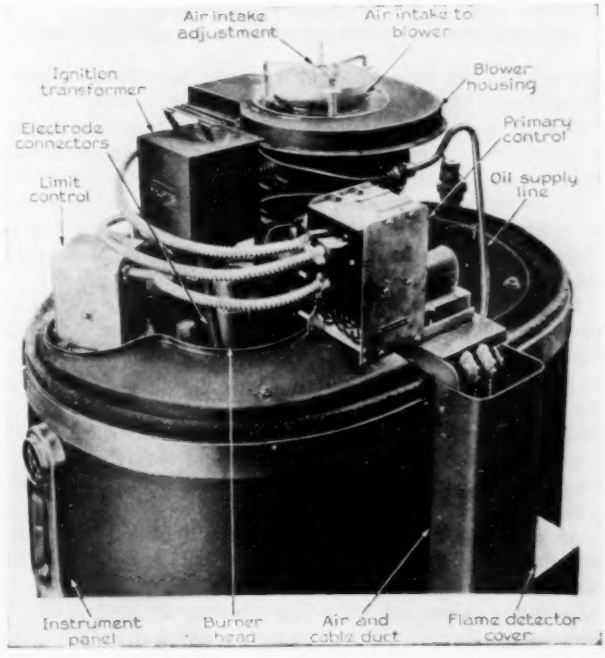


FIG. 15 OIL BURNER AND OIL-BURNER CONTROL ASSEMBLED ON BOILER

The limit control has three main functions: to keep the steam pressure between set limits; to operate the low-water cutoff for protecting the crown sheet, previously described; and to control the boiler water between 170 and 185 F for supplying heat to a domestic hot-water coil which is immersed in the boiler water, as shown in Fig. 1.

Flame Detector. The flame detector is mounted on the side of the boiler so as to permit the cleaning of the secondary heating surfaces without possible injury to the instrument. The element exposed to the products of combustion passes directly over one of the fabricated tubes, and is placed only a short distance beyond the zone where the final chemical reactions take place. The instrument is an extremely sensitive thermal-responsive device which operates on temperature changes in the combustion chamber. It detects the presence of flame during the ignition period, thus permitting continuous operation of the burner. It detects the absence of flame should this occur during operation, stops the flow of oil, and permits a new starting cycle to be inaugurated. The flame detector is so sensitive that it responds within two or three seconds, depending on the oil rate.

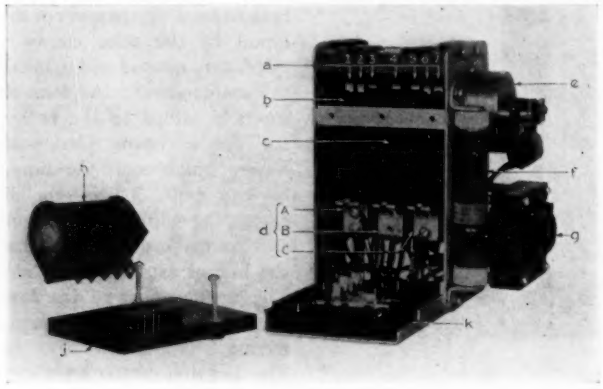


FIG. 16 PRIMARY CONTROL
(Showing oil-valve rectifier and rectifier plug board and cover.)

The flame detector is a nichrome tube within which is placed a quartz rod. The quartz has a negligible expansion with temperature changes; the nichrome has a decided expansion. The relative motion between the rod and the tube actuates contacts when the tube is subjected to heat from the combustion gases. The flame detector contacts control relays, as shown in the control circuit (Fig. 10).

If the oil does not ignite at the first try, the timing mechanism keeps on running after the flame detector has shut down the burner and repeats the cycle. Again, the condition may not be proper for combustion, and the flame detector will shut down the burner. The timing mechanism will continue running and repeating the cycle a number of times, but if combustion has not been established, the unit will shut down and no further action on the part of the burner is possible until a reset button is manually pressed.

Air and Cable Ducts. The intake-air and cable duct and the secondary-air ducts, the former on the right-hand side of the furnace and the latter on the left, are shown in Fig. 6. The intake-air and cable duct, aside from covering the electric and oil lines, has another function in that it conveys the air, which enters through louvers in the bottom of the duct, into the space under the boiler top cover. The cover, together with this duct, also acts as a sonic filter; that is, small mechanical noises originating under the cover are muffled.

Boiler Destruction Test Made for the A.S.M.E. Boiler Code Committee. The boiler was made to conform with the A.S.M.E. Low-Pressure Heating Boiler Construction Code. For some information that was wanted for the Boiler Code Committee, the boiler was subjected to a destruction test. At a hydrostatic test of 275 lb per sq in. gage, a small leak was formed in one of the fabricated tubes, but otherwise the boiler was in fine shape with no apparent injury. The test results are shown in Table 1.

TABLE 1 DESTRUCTION TEST OF OIL FURNACE BOILER (BOILER NO. 6)

Pressure, lb per sq in. gage	Deflection in mils				Space between two center adiators, in.	Remarks
	No. 1, top	No. 2, side	No. 3, side	No. 4, bottom		
25	4.5	0	0	0		
50	12	7	4	3		
75	19.5	14	9	6	61/64	
100	27.5	24.5	17	10	63/64	
125	33	35	25	12		
150	41	50	35	17	63/64	
175	54	64	48	20	61/64	Gasket failed on hot-water coil frame
0	10.5	21.5	15.5	5	63/64	
100	35	46	36	14	15/16	New gasket and cover
175	53.5	67.5	52	21	31/32	
200	65	81	63	25	15/16	
225	81	98	75	31	59/64	
250	100	113	85.5	33	57/64	
275	Weld failed in edge of adiator nearest combustion chamber					

OIL-BURNER CONTROL

The first application of the semi-luminous flame was in a domestic oil furnace. Hence, it is only proper to bring in a few words about the general control used. At several points, references have been made to various control parts such as the flame detector and boiler limit control, with further reference to the wiring diagram (Fig. 10). Mention was also made of the ignition spark, together with the manner in which the oil valve at the nozzle tip is operated.

Primary Control. By referring to Fig. 15 it will be seen how the primary control (Fig. 16) is mounted with reference to the oil burner and boiler. The primary control unit controls the sequence of operation of the various parts composing the complete oil burner. In this particular case, a thermal control (a wall-type clock thermostat with day and night control) and the boiler limit control take care of starting and stopping the oil burner, opening and closing the circuit to the oil valve and ignition transformer at the proper point in the starting cycle. The primary control also responds to the signals given by the flame detector, which indicates whether the flame is on or off. The primary control has a motor which causes the camshaft to make one revolution in 100 sec and which operates the contacts in the time sequence as shown in the wiring diagram (Fig. 10).

The cams and cam contacts control the following circuits: Referring to Fig. 16 and reading from left to right, they are: (1) Oil burner motor-starting winding. (2) Snap contact for motor-starting and running windings. (3) Oil-burner motor-running winding. (4) Ignition transformer. (5) Camshaft motor. (6) Snap contacts for cam motor and oil valve. (7) Oil valve. Also reading from left to right, the relays control the following circuits: Relay A: Left contact, A-1 (normally open), holding contact for thermal control circuit; right contacts, A-2 (normally open), main line contacts. Relay B: Right contact, B-1 (normally open), motor-running winding and oil-valve contacts. Relay C: Left contact, C-2 (normally closed), oil-valve contacts; right contact, C-1 (normally open), camshaft motor contacts.

Plug receptacles are used in the primary control for the copper-oxide rectifier, flame detector, thermal control leads, and for the oil-valve cable. The plug-in type arrangement is characteristic of the whole control. For instance, referring to Fig. 15, the ignition transformer can be lifted from its position merely by pulling out one plug and lifting directly by the case. The burner and

control, shown in Fig. 15, can be removed from the top of the boiler and placed back in position in less than 10 min, assuming that the burner was operating at the start of this test and is again running at the end of this test.

As mentioned under the head of atomization in branch *A*, Fig. 2, sometimes the flame does not ignite the first time. Hence the control was worked out for the "repeat start" system of control. During the five starts for which the control is mechanically adjusted, the flame must ignite or else the control must be reset manually.

The very sensitive flame detector makes it possible to reduce the oil-flow period during the starting operation, and use of the "repeat start" system of control eliminates the danger of ignition "puff."

PME-55-4

Some Problems in the Lubrication of Vertical Journal Bearings

By A. I. PONOMAREFF[1] and E. D. HOWE,[2] BERKELEY, CALIF.



A. I. Ponomareff

In deep-well pumping a motor at the surface drives a pump runner at the water level through a long vertical shaft supported at the top by a thrust bearing and guided at intervals of its length by journal bearings. For effective design the power absorbed by the journal bearings should be known approximately. The preponderance of horizontal shafts in various kinds of machinery has resulted in an extensive literature on their lubrication, but there is little that bears on the lubrication of such vertical shafts as are used in pumps, hydraulic turbines, a few steam turbines, cream separators, and centrifuges. The friction loss in these is small, but in the deep-well pump the loss may be as great as 5 per cent of the power input. The paper relates the details of an investigation to establish some of the basic facts underlying the lubrication of vertical journal bearings such as are used in deep-well pumps.



E. D. Howe

THE CONVENTIONAL deep-well pump installation consists of a motor at the ground elevation driving a pump runner at the water level (often several hundred feet below the ground surface). The power is transmitted from the motor to the pump through a long vertical shaft supported at the upper end by a thrust bearing and guided at intervals of from 3 to 10 ft by journal bearings. The power absorbed by the vertical journal bearings must be known approximately for effective design. However, a survey of the current engineering literature revealed that no data were available for this computation. The reason for the lack of information in the current literature probably lies in the preponderance of horizontal shafts in various kinds of machinery. Vertical shafts are used in comparatively few machines, some of which are pumps, hydraulic turbines, a few steam turbines, cream separators, and centrifuges. In all of these except the deep-well pump the loss in the vertical guide bearings is a very small part of the total bearing friction loss, as the major part of the bearing loss occurs in the thrust bearing. Since in the deep-well pump the total journal-bearing loss may be as great as 5 per cent of the power input, the design of these bearings becomes important. The following investigation was therefore made for the purpose of establishing some of the more basic facts underlying the lubrication of vertical journal bearings such as are used in deep-well pumps.

The experiments described were performed upon a vertical-bearing model designed by the authors and built by the Byron Jackson Company of Berkeley, Calif. The experimental work was done in the Mechanical Engineering Laboratories of the University of California at Berkeley. The authors gratefully acknowledge the many helpful suggestions given by Mr. A. Hollander, of the Byron Jackson Company, and the mechanical assistance given by the hydraulic laboratory of the university.

The theory for the lubrication of bearings for horizontal shafts is not applicable to the lubrication of vertical bearings because there exists in the former case a radial load which is lacking in the latter case. The theory of the lubrication of the vertical bearing is therefore developed independently from the equations of fluid motion, which apply only to those cases in which the fluid films are complete. The equations of motion[3] are:

$$\rho \frac{Du}{Dt} = \rho X - \frac{\partial p}{\partial x} + \frac{1}{3}\mu \frac{\partial \phi}{\partial x} + \mu \nabla^2 u$$

$$\rho \frac{Dv}{Dt} = \rho Y - \frac{\partial p}{\partial y} + \frac{1}{3}\mu \frac{\partial \phi}{\partial y} + \mu \nabla^2 v$$

$$\rho \frac{Dw}{Dt} = \rho Z - \frac{\partial p}{\partial z} + \frac{1}{3}\mu \frac{\partial \phi}{\partial z} + \mu \Delta^2 w$$

Let x = distance measured along the journal in the direction of motion, but referred to a point fixed in space (see Fig. 1)

y = distance measured perpendicular to the journal surface

z = distance measured axially from the end of the bearing sleeve

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] "The Mechanical Properties of Fluids," a collective work, 1924, D. van Nostrand Company, New York, p. 92.

$y_{max} = h$ = width of clearance space
u, v, w = the velocity components parallel to the axes at the point (x, y, z)
X, Y, Z = the external force components per unit mass
ρ = the mass density
μ = the coefficient of viscosity
p = the pressure at (x, y, z)
$\frac{D}{Dt} = \frac{\partial}{\partial t} + u\frac{\partial}{\partial x} + v\frac{\partial}{\partial y} + w\frac{\partial}{\partial z}$
$\phi = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}$
$\nabla^2 = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$
N = speed of rotation in rpm
F = force.

The following assumptions are made:

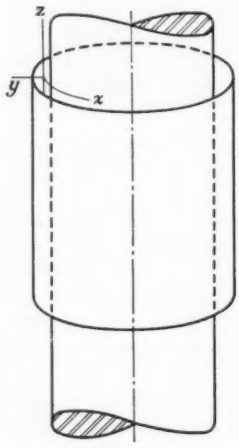


FIG. 1

(1) Reynolds' number $vl\rho/\mu$, computed from the journal velocity, the radial clearance, and the oil viscosity is so small that the flow is free of eddies.

(2) The forces due to weight and inertia of the fluid are so small that they may be neglected in comparison with the viscous forces.

(3) Since the fluid is a liquid under low pressures, all terms depending upon compressibility may be neglected.

(4) Because of the thinness of the oil film as compared with its length πd, the curvature of the film may be neglected.

(5) The axial flow is so small that it may be neglected and the problem may be treated as one of flow in two dimensions.

(6) The radial load is zero at all times.

The consequences of the above assumptions are:

(*a*) From (2) $\frac{Du}{Dt} \to 0$, $\frac{Dv}{Dt} \to 0$, $\frac{Dw}{Dt} \to 0$, $X, Y, Z \to 0$

(*b*) From (3) $\phi = 0$

(*c*) From (5) $w \to 0$

(*d*) From (6) $\frac{\partial p}{\partial x} = 0$, $\frac{\partial p}{\partial y} = 0$

Two cases arise in practice when the journal and sleeve surface elements are parallel: the first one is that in which the journal and bearing sleeve are concentric, and the second one that in which they are eccentric.

Case 1 (*Journal and Bearing Sleeve Concentric*). Since the shaft is concentric, $v = 0$, and there is no acceleration in the x-direction; so that $\frac{\partial^2 u}{\partial x^2} = 0$. Therefore Euler's equations reduce to:

$$\frac{\partial^2 u}{\partial y^2} = 0 \quad \text{or} \quad u = C_1 y + C_2$$

The constants can be evaluated from the boundary conditions, which are: when $y = 0$, $u = V$, and when $y = h$, $u = 0$. Inserting these values, $C_1 = -V/h$ and $C_2 = V$.

Hence $u = V(1 - (y/h))$.

From the definition of the coefficient of viscosity:

$$F_x = -\mu \frac{\partial u}{\partial y} \cdot A = \mu \frac{V}{h} A = \text{force tangential to shaft.}$$

The torque is:

$$F_x \cdot r = \mu \frac{V}{h} \frac{d}{2} \pi dl$$

or

$$T = F_x \cdot r = \mu \frac{V}{h} \frac{d}{2} \pi dl$$

where $A = \pi dl$

V may be expressed as $V = (\pi dN/60)$, so that:

$$T = \mu \frac{\pi^2 d^3 lN}{120\, h}$$

The power necessary to produce this torque is:

$$L_c = \frac{2\pi TN}{33{,}000} = \frac{\mu \pi^3 d^3 lN^2}{1{,}980{,}000\, h} \text{ (hp)} \ldots\ldots\ldots\ldots [1]$$

d = diameter of journal in feet
l = length of sleeve in feet
N = speed in rpm
h = clearance (radial) in feet
μ = viscosity in lb × sec per sq ft

or

$$L_c = 6.12 \times 10^{-10} \frac{\eta \pi^3 D^3 LN^2}{H} \ldots\ldots\ldots\ldots [1a]$$

where η = coefficient of viscosity in poises
D = diameter in inches
L = sleeve length in inches
N = speed in rpm
H = radial clearance in thousandths of inches.

That this equation is similar to that for the power loss in a pipe line may be seen by rearranging the factors:

$$L_c = K\eta D \left(\frac{L}{H}\right) V^2 = K'\eta D V^2 \ldots\ldots\ldots\ldots [a]$$

where K is a pure number.

The power is thus proportional to the product of shaft diameter and the square of the linear speed.

The power loss due to pipe friction is

$$L_f = \frac{Qjh_L}{550} = \frac{Avj}{550} \times f \frac{L}{D} \frac{v^2}{2g}$$

for viscous flow $f = \frac{64}{\frac{vd\rho}{\mu}}$, so that:

$$L_f = \frac{64}{550} A\mu \frac{L}{D^2} v^2$$

$$L_f = K_1 \mu l v^2 \ldots\ldots\ldots\ldots [b]$$

In both of these Equations [*a*] and [*b*], the power loss is proportional to the product of a length and the square of the speed, thus illustrating the analogy between pipe friction and bearing friction.

Case 2 (*Journal and Bearing Sleeve Eccentric*). In this case the change of speed of the particle is rather small, so that all of the assumptions for case 1 are valid. The effect of the eccentricity is to render the tangential force dependent upon the angle through the relation:

$$dF = \mu \frac{V}{h} dA$$

in which $h = f(\theta)$; θ = angle of radius vector; $dA = lrd\theta$, so that $dF = \mu lr \frac{V}{h} d\theta$.

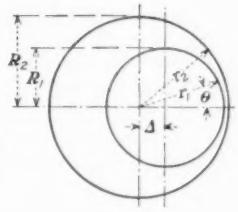


FIG. 2

To find $h = f(\theta)$, consider the figure shown. The equations of the two circles (see Fig. 2) in cylindrical coordinates are:

large circle $r_2 = R_2$

small circle $r_1^2 - 2r_1 \Delta \cos\theta + \Delta^2 = R_1^2$

Solving the latter equation for r_1, gives:

$$r_1 = \frac{2\Delta\cos\theta \pm \sqrt{4\Delta^2\cos^2\theta + 4(R_1^2 - \Delta^2)}}{2}$$

Since the two radii are nearly equal, the clearance h is given by:

$$h = r_2 - r_1 = R_2 - [\Delta\cos\theta \pm \sqrt{\Delta^2\cos^2\theta + (R_1^2 - \Delta^2)}]$$

Hence:

$$dF = \frac{\mu l R_1 V d\theta}{R_2 - [\Delta\cos\theta \pm \sqrt{\Delta^2\cos^2\theta + (R_1^2 - \Delta^2)}]}$$

or

$$F = \mu l V R_1 \cdot 2 \int_0^{\pi} \frac{d\theta}{R_2 - [\Delta\cos\theta \pm \sqrt{\Delta^2\cos^2\theta + (R_1^2 - \Delta^2)}]}$$

To evaluate the integral, note that the radical may be reduced, since $\Delta^2(1 - \cos^2\theta)$ is negligible in comparison with R_1^2. The integral therefore becomes:

$$2\int_0^{\pi} \frac{d\theta}{R_2 - R_1 - \Delta\cos\theta}$$

which can be evaluated by Equation [234], page 46, Hudson's "The Engineering Manual," as:

$$= \frac{4}{\sqrt{(R_2 - R_1)^2 - \Delta^2}} \tan^{-1}\left(\frac{\sqrt{R_2 - R_1 - \Delta}}{\sqrt{R_2 - R_1 + \Delta}} \tan\frac{\theta}{2}\right)\Bigg]_0^{\pi}$$

$$= \frac{2\pi}{\sqrt{(R_2 - R_1)^2 - \Delta^2}}$$

Inserting this value in the foregoing equation gives:

$$F = \mu l V R_1 \frac{2\pi}{\sqrt{(R_2 - R_1)^2 - \Delta^2}}$$

The power necessary in this case is:

$$L_c = \frac{2\pi N R_1}{33{,}000} \times \frac{\pi d_1 N}{60} \cdot R_1 \mu l \frac{2\pi}{\sqrt{(R_2 - R_1)^2 - \Delta^2}} \quad \ldots\ldots [2]$$

$$= \frac{\mu\pi^2 d_1^3 N^2 l}{1{,}980{,}000\, h_c} \times \frac{1}{\sqrt{1 - k^2}} = \frac{L_c}{\sqrt{1 - k^2}}$$

d = diameter of journal in feet
h_c = difference in radii in feet
Δ = eccentricity in feet

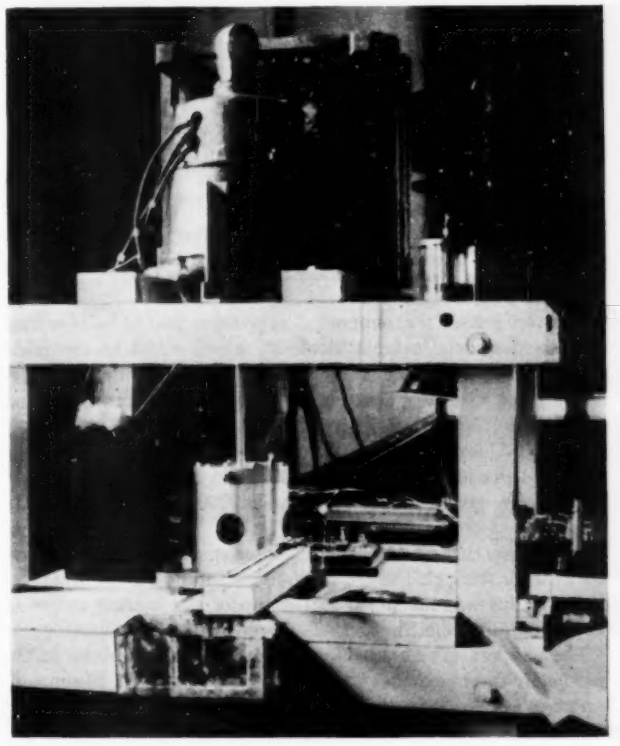
$$k = \frac{\Delta}{R_2 - R_1}$$

Other symbols as before.

DESCRIPTION OF THE APPARATUS

To verify experimentally the theoretical relations established by Equations [1] and [2], the test apparatus was designed to permit (1) the setting of a given journal centrally in the bearing sleeve and (2) the movement of the journal laterally in the bearing sleeve to obtain a desired amount of eccentricity. The apparatus station used in the tests is shown in Fig. 3, and a cross-section of the test bearing showing the arrangement of the principal working parts in Fig. 4.

In Fig. 4 the brass bearing sleeve E, 2.4415 in. inside diameter by $5^1/_2$ in. long, is shown supported centrally by the two ball bearings B, which are in turn pressed into the steel housing P. The sleeve is free to rotate in either direction, being limited in angular motion only by the lever L, which strikes the sides of the


FIG. 3 APPARATUS USED IN THE TESTS

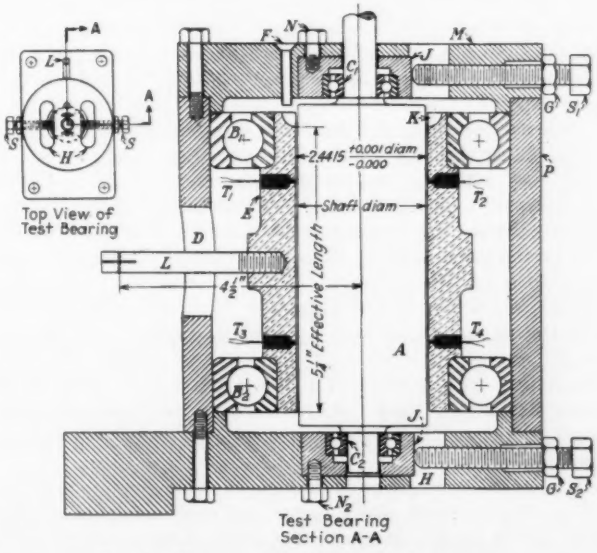


FIG. 4 WORKING PARTS OF TEST APPARATUS

hole D. In order to insure parallelism of the axes of the sleeve and the journal, the steel housing was bored and faced in a single lathe set-up, and the housing cover and base M were provided with spigot joints, which were machined in the same set-ups used for the boring operations upon them. Four slots in the cover and base of the housing were so placed that, without removing either cover or base, the clearances could be measured with an ordinary feeler gage.

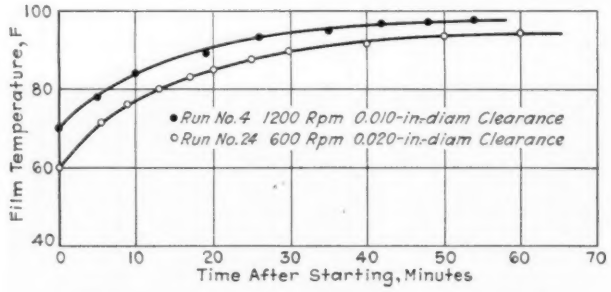


FIG. 5 ESTABLISHMENT OF EQUILIBRIUM CONDITIONS
(Change of film temperature with time. Vertical bearing tests, bearing $2^{7}/_{16}$ in. in diameter by $5^{1}/_{4}$ in. long.)

The test journal was supported by another pair of ball bearings C, pressed into adjustable blocks J, which could be moved in guides by adjusting screws S. Thus the journal could be placed centrally or with any desired eccentricity in the bearing sleeve by means of the adjusting screws, and then locked in position by bolts N and lock nuts G. Change in average clearance was made by using three journals, 2.4405, 2.4400, and 2.4395 in. in diameter, respectively, giving 0.010, 0.015, and 0.020 in. clearance on diameter.

The lubricating oil was supplied from an oil cup located above the bearing through the $^{3}/_{16}$-in.-in-diam tubing F. A groove of $^{1}/_{8}$-in. radius was machined in the top of the bearing sleeve to insure uniform distribution of oil around the journal.

The temperature of the oil was measured at four places on the bearing by thermocouples T. No. 30 copper and No. 30 constantan wires were used for thermocouples. The method of attaching the thermocouples was as follows: The wires were first arranged in pairs, wrapped together with thread, coated with bakelite varnish and then baked. Holes $^{1}/_{16}$ in. were then drilled through the sleeve and counterbored with a diameter of about $^{1}/_{4}$ in., as shown in the sketch. The insulation was then scraped from the ends of the wires, and the wires were placed in the center of the hole projecting beyond the bearing surface far enough so that their exposed portions extended about $^{1}/_{32}$ in. back into the hole. Woods metal, melting at about 200 F, was then poured around the wires, filling the hole. The wires and excess metal were then filed flush with the surface. The thermocouple junction was thus formed by the Woods metal immediately adjacent to the bearing surface. The temperature of the junction should therefore be the temperature of the bearing surface, which approaches that of the oil. One advantage of this construction was that the surface was not warped by heating as would have been the case had the thermocouples been soldered in place. Each thermocouple was calibrated after attachment to the bearing sleeve by comparison with a standard thermometer over a temperature range of from 60 to 150 F. A Brown Instrument Company potentiometer was used for measuring the voltage generated at the hot junction.

The journal was connected through a flexible coupling to a 1-hp 110-v direct-current shunt motor mounted vertically on the table above the test bearing. The speed of the motor could be varied from 300 to 1350 rpm by means of rheostats. (For the tests at 3600 rpm a $^{1}/_{3}$-hp alternating-current motor was used.) The speed of the motor was measured by means of a Horn type tachometer directly connected to the end of the motor shaft.

To measure the rotative effort on the main bearing sleeve produced by the rotating journal a lever arm L was projected through a 2-in. hole in the bearing housing. The torque was measured by a spring scale attached to the lever arm at a radius of $4^{1}/_{2}$ in.

EXPERIMENTAL PROCEDURE

The procedure followed in the final tests was adopted after a number of trial runs. The oil cup was adjusted to supply oil at a rate sufficiently great to keep the groove on top of the bearing sleeve filled, thereby assuring a uniform flow into the clearance space. The motor was then started and constant speed was maintained until the torque and film temperature became constant. More than one hour was required to establish thermal equilibrium, as may be seen from Fig. 5, a curve showing the change of oil-film temperature with time. To obtain a wide range of viscosities three different oils were used in the tests. The curves in Fig. 6 show the variation with temperature of the oil viscosities.

Some difficulty was experienced in maintaining a complete film with the apparatus as described, so that a second series of tests was made with the bearing sleeve completely immersed in oil. For this purpose the four holes, through one of which the torque arm extended, were closed by means of plates. Leakage was allowed to occur around the torque arm, thus providing a continuous flow through the apparatus. The procedure followed in this series was the same as in the series with the sleeve not immersed.

After thermal equilibrium of the oil film and the bearing had been established, the torque, speed, and oil-film temperature were measured. The theoretical power required to drive the journal

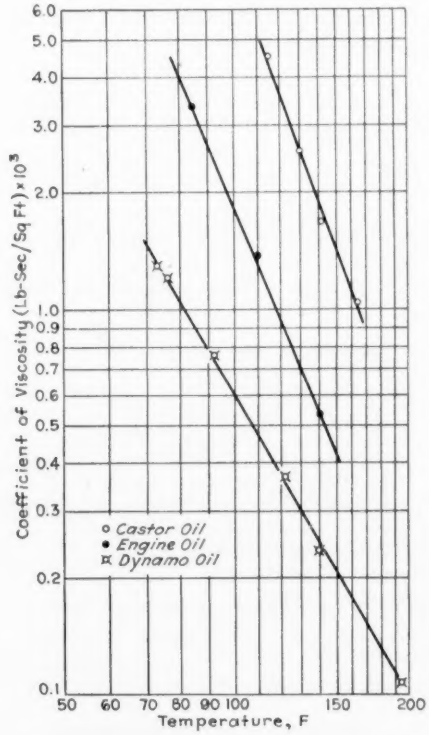


FIG. 6 VISCOSITIES OF OILS USED IN TESTS
(Data obtained with Saybolt viscometer. Vertical bearing tests, bearing $2^{7}/_{16}$ in. in diameter by $5^{1}/_{4}$ in. long.)

was calculated from Equations [1] and [2], and the actual power from:

$$L_a = \frac{2\pi PLN}{33{,}000} \text{ hp} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

where P = pull on spring scale in pounds
L = length of lever arm in feet
N = speed in revolutions per minute.

Results Obtained

The results of the tests made on the 2.4405- and 2.4395-in.-in-diam. journals located centrally in the bearing sleeve with diametral clearances of 0.010 and 0.020 in., respectively, are shown graphically in Fig. 7. The solid lines give the theoretical power required to drive the journals as computed from Equation [1], using a viscosity of 1.00×10^{-3} (lb-sec/sq ft) (47.9 centipoises). The experimental points were obtained with various viscosities ranging from 0.3×10^{-3} to 6.9×10^{-3} (lb-sec/sq ft). (The film temperatures ranged from 70 F to 160 F.) The measured power was then divided by (the viscosity $\times$ 1000) in order to reduce all values to a common basis. The resulting points were then plotted in Fig. 7. (Note that for viscosities greater than 1.00×10^{-3} it was necessary to immerse the bearing sleeve to insure a complete film.) The plotted experimental values are seen to agree very well with the theoretical curve, the maximum deviation being less than 8 per cent, in agreement with the computed probable error. (See the Appendix.)

The numerous runs made by the authors seem to indicate conclusively that for oils having, under the operating conditions which obtained, viscosities less than approximately 0.7×10^{-3} (lb-sec/sq ft), a positive hydrostatic head on the oil in the clearance is necessary to maintain a complete film. Without such a head the oil is apparently wiped off the journal, thus reducing the effective area and consequently the amount of power

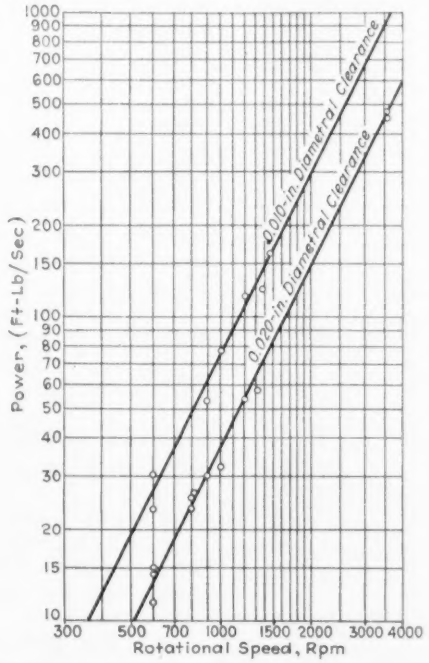


Fig. 7 Comparison of Theoretical and Measured Power

[Journal and sleeve concentric. Solid lines: Power = $\frac{\mu\pi^3 d^3 l^3 n^2}{3600\ h}$ (μ = 100 $\times 10^{-3}$). Plotted points: Power = $\frac{2\pi PLN}{60\mu \times 10^{-3}}$. Vertical bearing tests, bearing $2^7/_{16}$ in. in diameter by $5^1/_4$ in. long.]

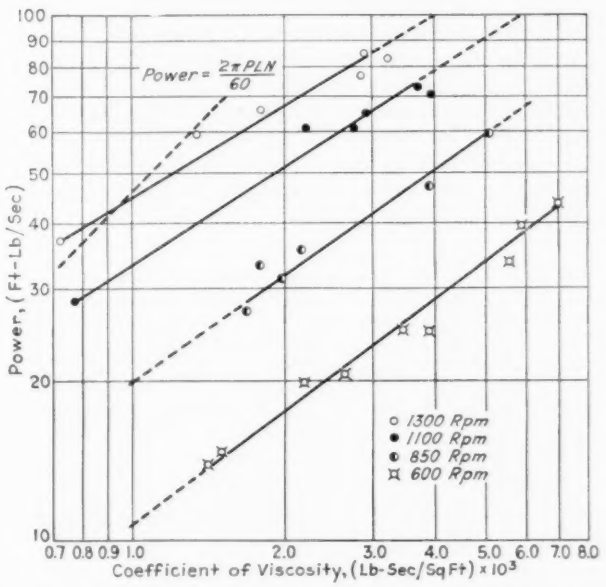


Fig. 8 Effect of Oil Viscosity Upon the Power Required for Lubrication

(Bearing sleeve open to the air. Vertical bearing tests, bearing $2^7/_{16}$ in. in diameter by $5^1/_4$ in. long; 0.020 in.-in-diameter clearance; 0.002-in. eccentricity.)

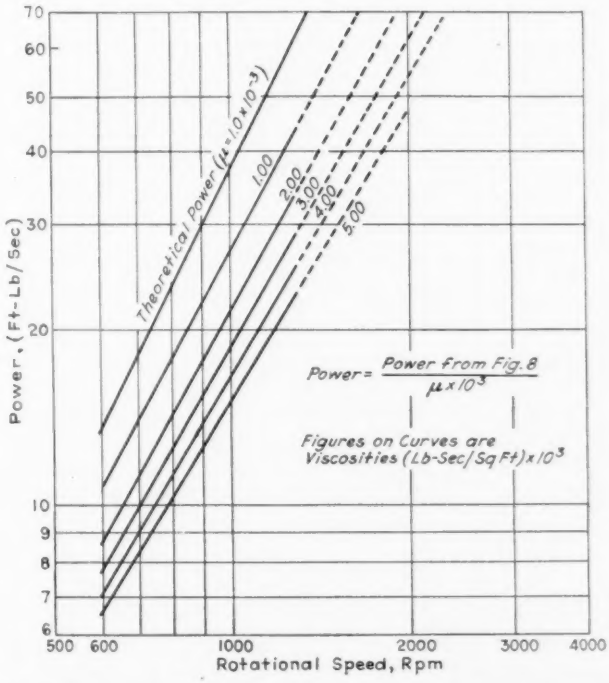


Fig. 9 Effect of Oil Viscosity Upon the Power Required for Lubrication

(Cross-plotted from Fig. 8. Vertical bearing tests, bearing $2^7/_{16}$ in. in diameter by $5^1/_4$ in. long; 0.020-in.-in-diameter clearance; 0.002-in. eccentricity.)

required to drive the shaft. Carrying this to the extreme, one might be led to the erroneous conclusion that the most desirable state would be one in which no film existed, thus reducing the power to a minimum. The fallacy of this will be apparent if it is borne in mind that the purpose of the oil film is to preserve the alignment of the shaft with respect to the housing. Absence of an oil film in a deep-well pump bearing would probably re-

sult in an enormous increase in power required due to the solid friction during rubbing of the journal over the bearing surface.

With the oil cup adjusted to feed 3 drops per minute, as in the actual deep-well pump installations, the reduction of the active oil-film area might produce a decrease of power of as much as 50 per cent. This effect was noticed most definitely when running with cold oil; that is, when the viscosity was large. The reduction of power necessary to drive the journal due to the reduction of the effective oil-film area is shown in Figs. 8 and 9.

The solid curves in Fig. 8 are drawn through the experimental values and show the variation of power required with viscosity at several different speeds. According to Equation [1] these curves should have slopes the same as that of the dotted curve. The reduction in slope is from 1.00 (theoretical) to about 0.60 (actual) for 1100 rpm. This change in slope suggests that possibly the degree of completeness of the oil film is dependent upon the viscosity.

The curves in Fig. 9 are cross-plotted from those in Fig. 8, after reducing all values to a viscosity of 1.00×10^{-3} (lb-sec/sq ft). This curve indicates that for large viscosities (cold oil) a complete film is not maintained, and the power required to drive a journal under such conditions is considerably smaller than computed from Equation [1]. As the viscosity is decreased, it becomes easier to maintain a complete oil film, so that the theory given is more nearly applicable.

The tests made to ascertain the effect of eccentricity of the journal show that for a small eccentricity the effect is negligible, as indicated by Equation [2]. For instance, an eccentricity of 0.003 in. when the average radial clearance is 0.010 in. would produce an increase in power of only 5 per cent, an amount less than the error in the work. For greater eccentricities the measured power was less than the theoretical, from which it is assumed that the film was incomplete due to the reduced minimum clearance.

The results indicate rather definitely that the power actually required by a vertical bearing can be computed directly from Equation [1], when conditions are such that a complete film is maintained.

Under these conditions the power may be obtained from Fig. 10, which was drawn from computations using Equation [1].

The tests previously described have been concerned with the absorption of power by vertical bearings under a small range of conditions. In addition to the factors studied there are others which should be investigated. Some of these factors are:

(*a*) The effect of grooves running the full length of the sleeve

(*b*) The effect of grooves running part way down the sleeve

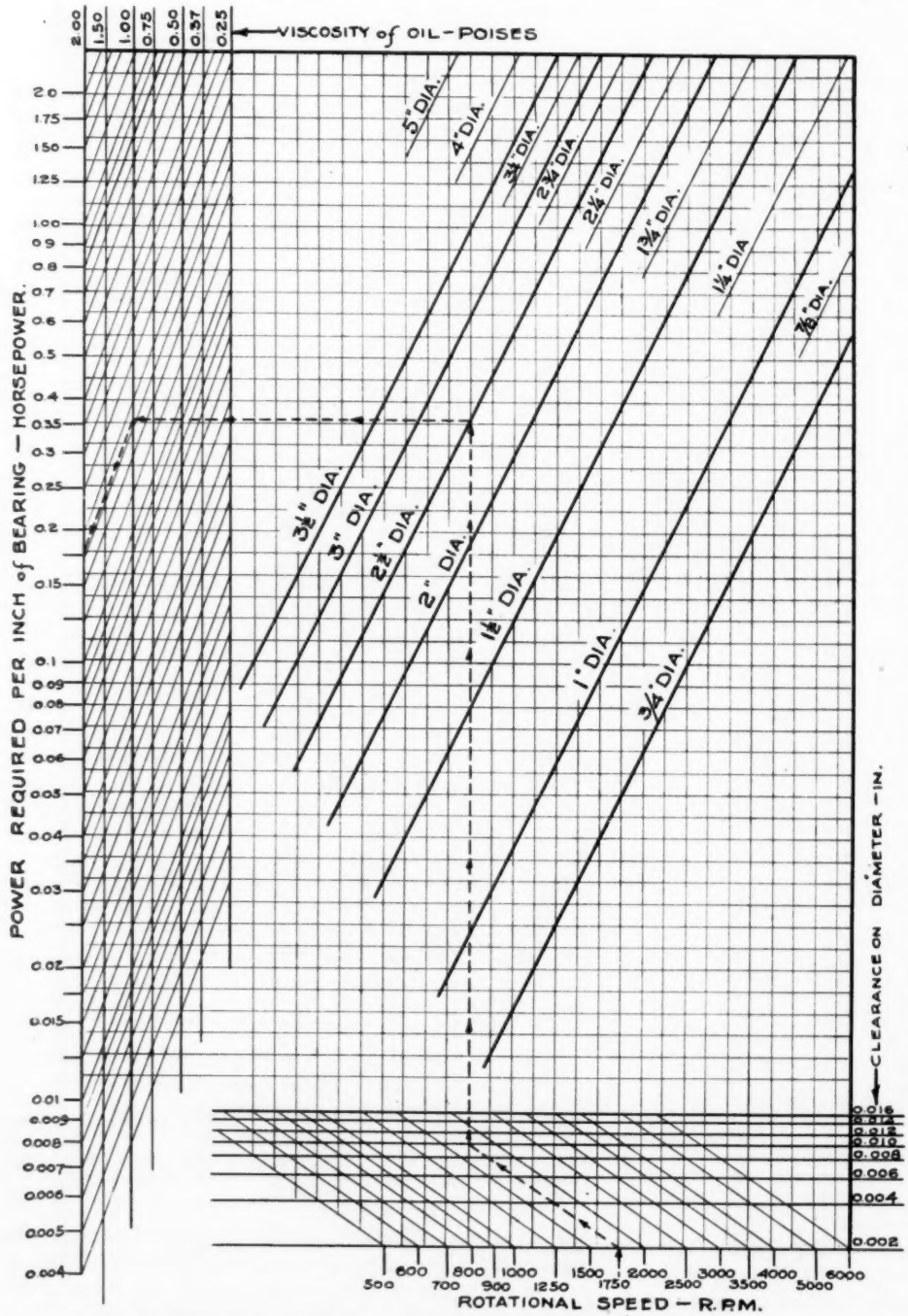


FIG. 10 POWER REQUIRED BY A VERTICAL JOURNAL BEARING PER INCH OF LENGTH

(Based on the equation: Horsepower $= 6.12 \times 10^{-10} \frac{\eta \pi^3 L D^3 N^3}{H}$; where η = viscosity in poises, D = diameter in inches, L = length in inches, N = rpm, H = radial clearance in thousandths of an inch.)

(c) The effect of greater pressures on the oil.

The result of perfect workmanship in the construction of a deep-well pump would be a pump in which the journals and bearing sleeves were concentric and parallel. This is the aim of the manufacturer and is the condition to which the results derived in this work apply directly. Fig. 10 makes it possible for one to obtain immediately the power absorbed by a vertical bearing, assuming that a complete film exists. If the film is incomplete, Fig. 9 gives an approximate idea of the deviation from the case of the complete film.

That these results are dependable is evidenced by the agreement between the actual and theoretical values as shown in Fig. 7. It is to be hoped that these results may point the way to the more effective design of vertical journal bearings in practice and to further research on a problem vital to one of the industries.

Appendix

SAMPLE CALCULATIONS FOR RUN 19-4

1 Theoretical power from Equation [1]:

$$L_c = \frac{\mu\pi^3 d^3 l N^2}{3600\, h}$$

$$= \frac{1.15 \times 10^{-3}}{3600}\,\pi^2\left(\frac{2.44}{12}\right)^3 \times \frac{5.25}{12} \times \frac{12}{0.010} \times 800^2 = 27.9 \text{ ft-lb per sec.}$$

2 Actual power:

$$L_a = \frac{2\pi LPN}{60}$$

where P = pull on scale
L = length of lever arm
N = speed in rpm

$$L_a = \frac{2\pi \times 14.0 \times 4.50 \times 800}{60 \times 12 \times 16}$$

27.5 ft-lb per sec.

3 Actual power based upon viscosity 1.00×10^{-3} lb-sec per sq ft where

$$L_a' = L_a \times \frac{\mu_{st}}{\mu_a}$$

μ_{st} = standard viscosity = 1.00×10^{-3}
μ_a = observed viscosity

$$L_a' = 27.5 \times \frac{1.00 \times 10^{3}}{1.15 \times 10^{-3}} = 23.9 \text{ ft-lb per sec.}$$

4 Accuracy of the results:

The errors involved in the determination of the measured and computed power are of two kinds: accidental errors of observation and systematic and constant errors due to the instruments and methods used.

Systematic and constant errors are evidenced by a greater departure from the theoretical value than the accidental errors would indicate. In the work described, the instruments and methods used were, with one exception, standard. The exception was that of the film temperature measurement, which was accomplished with the thermocouples already described. The results, as will be shown, fall within the computed accidental error, which is an indication that the systematic and constant errors must be of small magnitude.

The measurements were made by means of the instruments named in the table at the foot of the page.

The accidental errors of the results can be computed from the relation:[4]

$$M_{La}^2 = \left(\frac{\partial L_a}{\partial p} \cdot M_p\right)^2 + \left(\frac{\partial L_a}{\partial L} \cdot M_L\right)^2 + \left(\frac{\partial L_a}{\partial N} \cdot M_N\right)^2$$

where M_{La} = mean square error in L_a
M_p = mean square error in p, etc.
$\frac{\partial L_a}{\partial p}, \frac{\partial L_a}{\partial L}, \frac{\partial L_a}{\partial N}$ = partial derivatives.

To determine the mean square errors of the individual measurements, note these data as given for run 19-4:

Time	Speed, rpm	Film temperature—Upper	Lower	Mean	Pull, oz	Viscosity, lb-sec/sq ft
4:15 pm	800	116.5	115.3	116.0	13.8	.....
4:20 pm	800	116.5	115.8	116.1	13.8	.....
4:25 pm	800	116.5	115.8	116.1	14.0	.....
4:30 pm	800	116.5	115.8	116.1	14.0	.....
Average	800	116.5	...	116.1	13.9	1.07×10

Since the data show no appreciable variation from the mean, it is assumed that the mean square error of each quantity is equal to the sluggishness of the instrument. The partial derivatives are obtained from the equations used in calculations. For example:

$$\frac{\partial L_a}{\partial p} = \frac{2\pi LN}{60} = \frac{2\pi \times 4.50 \times 800}{60 \times 12 \times 16} = 1.96$$

Evaluating the other partial derivatives and substituting, the mean square error in the actual power is:

$$M_{La}^2 = \overline{1.96^2} \times \overline{0.3^2} + \overline{6.06^2} \times \overline{0.02^2} + \overline{0.0364^2} \times \overline{3^2}$$
$$= 0.35 + 0.015 + 0.012 = 0.377$$

or $M_{La} = 0.6$ and $L_a = 2.75 \pm 0.61$ ft-lb per sec.

A similar computation for the theoretical power gives:

$L_c = (27.9 \pm 1.9)$
$= (28 \pm 2)$ ft-lb per sec.

In this case the greatest error lies in the determination of h, the mean radial clearance, which is assumed measurable to the nearest 0.0005 in. due to the effects of thermal expansion.

Examination of the curves in Fig. 7 shows that the deviation of the experimental points from the curves is in all cases less than 10 per cent, which compares favorably with the 7 per cent error as heretofore computed.

Quantity	Instrument: Name	Least count	Calibration: Sluggishness	Standard	Conditions
N = speed	Tachometer	10 rpm	3 rpm	Stop-watch rev. counter	Direct-connected vert. shaft
t = film temp.	Thermocouples, potentiometer	0.01 mv	0.5 F	B. of S. thermometer	Bearing shell immersed in oil
P = force	Spring balance	0.5 oz	0.3 oz	Dead weights	Horizontal position
h = clearance	Feeler gage	0.0005 in.			
d = diameter	Micrometer calipers	0.0005 in.			
Viscosity	Saybolt viscometer	0.2 sec, 1 F			
Lever arm	Scale	1.64 in.			

Discussion

BURT L. NEWKIRK.[5] The authors have studied the losses due to fluid friction of a journal constrained to rotate centrally or with a definite eccentricity in a vertical sleeve. In the course of experimental studies published[6] some time ago it was noted that

[4] A. F. Palmer, "The Theory of Measurements," 1912, McGraw-Hill Book Company, p. 100.

[5] Engineer, Research Laboratory, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

[6] "Shaft Whipping Due to Oil Action in Journal Bearings," by B. L. Newkirk and H. D. Taylor, *Gen. Elec. Rev.*, August, 1925, pp. 559–568.

TABLE 1 SUMMARIZED DATA AND RESULTS

Series 1, Journal and Sleeve Concentric

Run No.	Radial clearance, in.	Speed, rpm	Average pull on torque arm, oz	Average film temp, F	Abs. viscosity, lb-sec/sq ft $\times 10^6$	Computed power, ft-lb per sec.	Measured power, ft-lb per sec.	Measured power based on $\mu = 1.00 \times 10^{-6}$	
9-1	0.010	600	26.0	86.5	3.07	41.8	35.3	11.5	Sleeve open to air
19-5	0.010	600	12.5	110.0	1.30	17.7	18.4	14.2	Sleeve immersed in oil
19-8	0.010	600	11.6	113.5	1.15	15.6	17.1	14.9	" " " "
19-4	0.010	800	13.9	116.0	1.07	25.9	27.2	25.4	" " " "
19-6	0.010	800	16.5	107.6	1.37	33.2	32.3	23.6	" " " "
19-7	0.010	815	12.9	119.5	0.97	24.4	25.8	26.6	" " " "
20-1	0.010	900	14.8	116.0	1.08	32.8	32.7	30.2	" " " "
19-2	0.010	1000	16.5	111.0	1.25	47.2	40.4	32.3	" " " "
20-2	0.010	1200	14.2	128.0	0.77	42.0	41.8	53.6	" " " "
10-3	0.010	1335	20.5	113.5	1.17	79.5	67.4	57.6	Sleeve open to air
21-2	0.010	3530	18.0	161.5	0.33	155.0	156.0	47.3	Sleeve immersed in oil
21-1	0.010	3530	19.5	155.5	0.37	174.0	169.0	45.7	" " " "
23-4	0.005	600	5.4	136.0	0.260	7.1	7.9	30.4	Sleeve immersed
22-4	0.005	600	15.0	120.0	0.95	25.7	22.1	23.3	" "
22-3	0.005	900	22.0	121.0	0.92	56.0	48.7	52.8	" "
23-3	0.005	1000	9.4	131.0	0.292	22.0	23.0	78.8	" "
23-1	0.005	1200	12.1	129.5	0.303	32.9	35.5	117.0	" "
22-2	0.005	1360	30.0	126.0	0.82	110.0	101.0	123.0	" "
23-2	0.005	1450	13.3	131.3	0.292	46.3	47.3	162.0	" "

Series 2, Journal and Sleeve Eccentric

(Maximum and minimum clearance, 0.012" × 0.008". Bearing shell open to the atmosphere)

Run No.	Speed, rpm	Average pull on torque arm, oz	Average film temp, F	Abs. viscosity, lb-sec/sq ft $\times 10^6$	Computed power, ft-lb per sec	Measured power, ft-lb per sec
12-1	1300	11.6	129.5	0.73	..	37.2
17-2	1300	18.5	151.0	1.35	..	49.2
17-4	1300	20.5	140.5	1.82	..	65.6
17-3	1300	24.0	127.0	2.87	..	76.8
16-1	1300	26.5	126.5	2.90	..	84.9
17-1	1300	26.0	123.5	3.22	..	83.2
12-2	1100	10.6	127.5	0.78	..	28.6
15-5	1100	17.5	157.5	1.17	..	47.2
15-4	1100	19.0	149.5	1.38	..	51.3
15-3	1100	22.5	134.5	2.22	..	60.7
17-5	1100	22.5	128.0	2.75	..	60.7
15-2	1100	24.0	126.5	2.92	..	64.8
15-1	1100	27.0	119.5	3.70	..	72.8
14-1	1100	26.0	118.0	3.90	..	70.2
17-7	850	13.0	142.5	1.70	..	27.2
13-1	855	16.0	141.0	1.80	..	33.4
16-5	850	15.0	138.0	1.97	..	31.3
16-3	850	17.0	135.3	2.17	..	35.5
17-6	850	22.5	118.0	3.92	..	47.1
16-4	850	28.5	110.5	5.07	..	59.5
12-5	600	9.5	107.0	1.41	20.7	14.0
12-3	600	10.0	105.0	1.50	22.0	14.7
12-4	600	13.5	95.0	2.20	32.4	19.9
17-9	600	14.0	129.0	2.65	39.0	20.6
16-7	600	17.0	121.5	3.45	50.7	25.0
16-6	600	17.0	118.0	3.90	57.4	25.0
17-8	600	23.0	107.0	5.58	87.1	33.8
15-6	600	27.0	105.5	5.90	86.8	39.7
16-8	600	29.5	100.0	7.00	103.0	43.4

oil films in journal bearings have, under some circumstances, a tendency to cause the journal to whirl in its bearing. Thus a vertical shaft running in a sleeve bearing so that it is free to move within its clearance may not run centrally or with any fixed eccentricity, but it may move over to a very eccentric position and then travel around in the clearance at a rate equal to half the rotation speed. This tendency may be overcome[7] by any force of moderate magnitude tending to hold the journal to one side of its clearance. It is possible that the bearings of the long shafts of deep-well pumps are sufficiently out of line to supply such a radial force for each bearing and so prevent the whirling, but it is well to keep in mind this whirling tendency of the oil film in any discussion of vertical shafting carrying no radial loading. It seems likely that in bearings having the large clearances used in the experimental apparatus of the authors (0.010 to 0.020 in.) the journals would run in very eccentric positions or whirl.

M. D. HERSEY.[8] The formula for friction of a concentric full journal bearing has been known for many years. A formula for eccentric bearings was published in the Journal of the Washington Academy of Sciences, November 19, 1914, which is exactly equivalent to that of the paper, but only regarded as an approximation. The useful features of the paper are those relating to the apparatus itself, and its application to the eccentric case, and might be more thoroughly investigated when possible.

C. G. WILLIAMS.[9] It is regrettable that there is so little information on vertical bearings. The writer has had experience with deep-well pumps north of Antoneda, Colo., which is irrigated from wells driven from 350 to 950 ft. The pumps were operated by electric motors, and in the deepest wells the frictional resistance in the vertical guide bearings and the thrust bearings amounted to 15 per cent of the input of power when the installation was new. The pumps were multiple-stage centrifugal pumps in 12- to 14-in. tubing. The flow of water from the well was through a 6- or 8-in. pipe. The drive shaft occupied the center of the pipe, and at every joint, of from 16 to 18 ft, a guide bearing was located. In the original installations these guide bearings were of a soft bronze, and in some replacements of soft stick brass, such as for bushings on railroads, the repair work being generally done in the local railroad shop.

These bronze or brass bushings lasted not more than one season of approximately 8 months. The cause of the cutting out of the bearing and part of the cause for the cutting of the shaft was the presence of a layer of fine white quartz sand, such as would pass through 100-mesh screen. The thrust bearing was at the bottom of the motor shaft, which is the poorest location. The housing contained over 2 in. of metal all around the bearing. This allowed the bearing to be made 2 in. larger in diameter. A thrust bearing, somewhat like the Kingsbury thrust bearing, was installed. Rubber bushings or bearings were installed at all points along the shaft. There was a large supply of old feedwater-pump rubber valves 1 in. thick, three of these making a bearing. A fluted punch cut out the center of the rubber disk to allow free water flow through the bearing around the shaft, thus lubricating the bearing. The repairs were made in 1912 in one week's time, and when the writer was in that country in 1922, he found that every pump so repaired was running in good condition, except one pump where the runner had been cut out by the quartz sand. In 1923 the same type of bearing was installed in a multiple-stage centrifugal pump operating in a 24-in. tubing at a depth of 225 ft.

The power losses on the pumps in Colorado were cut from the figures given above to less than 0.25 per cent, while the pump in Arizona, which had had a power loss of 25 per cent in a 200-hp motor, had the losses reduced to 0.15 per cent after the installation of new thrust bearings and soft rubber guide bearings.

Because of the torsional effect of the power on the long shaft, there is a tendency for all of the wear to come on one side of the shaft, rather than on the entire circumference. This stress has the effect of causing the bearing and shaft to assume the features of a stationary shaft and moving bearing that we sometimes have in horizontal-bearing combinations. This effect will, naturally, have an effect on lubrication of the bearing quite different from

[7] Except where the rotor speed is more than twice the critical speed.

[8] Vacuum Oil Company, Inc., Paulsboro, N. J.

[9] Consulting Mechanical Engineer, Rockford, Ill.

that in machine tools; however, where this effect is present, more care must be applied to the manner in which the lubricant is delivered to the bearing.

ALEXEY J. STEPANOFF.[10] The knowledge of the power lost in bearing friction of deep-well pumps is of much more importance than that of any other type of centrifugal pump. Short units can be guaranteed within close margin, since they are tested in the shop under the same conditions as they are operated in the field. Deep-well pumps are sold on the basis of pump over-all efficiency covering hydraulic losses (as a rule the velocity head at discharge is considered a loss) and bearing losses of the column sometimes several hundred feet long, while the manufacturers' facilities usually provide for testing deep-well pumps with short columns only. Byron Jackson Company has built a tower for testing deep-well pumps with columns up to 135 ft long.

The predetermination of the field performance of deep-well pumps involves the following steps: (1) Add to the given static head hydraulic losses in the column; (2) for the head thus obtained and capacity specified the proper pump is selected from the factory test data and "shop bhp" is calculated; (3) add to the shop bhp the power absorbed by the column shaft for field bhp. The power consumed by the thrust bearing is usually provided for in the factory test; however, if the factory tests are run with ball thrust bearings and a Kingsbury bearing is used, the power to drive this bearing should be added to the shop bhp; (4) by dividing the water hp based on the field head by the field bhp from (3), the field pump efficiency is obtained; (5) with the motor efficiency, the plant efficiency can be calculated.

It is seen that the knowledge of the power lost in the column-shaft friction is necessary, not only for effective design, but for every-day routine work. The column bearings of deep-well pumps vary in size and operate at different speed—some of them on oil, others on water. Thus the effect of clearance, speed, diameter, and viscosity of lubricant should be known. The laws of lubrication of vertical bearings were not always understood, and experimental data were scarce and haphazard.

The work accomplished by the authors is the first and necessary step toward the solution of this problem. By the study of an isolated bearing the authors were able to trace the effects of the eccentricity, incompleteness of film, and temperature and thus prepare the future investigator of the long shafts with many bearings for the difficulties he may encounter in the work of this nature.

The writer wishes to call attention to several points not sufficiently stressed. The Euler's equation in its final form applied to the flow similar to that found in the vertical bearings

$$u = V(1 - y/h)$$

means that the velocity of the liquid contained between two surfaces, one moving and the other stationary, is proportional to its distance from the stationary wall and that velocity distribution is a linear function of that distance. Thus the average velocity of the film is equal to one-half that of the moving wall and is independent of the clearance. This is true as long as the flow is viscous. An interesting application of this is found in centrifugal-pump practice. Since the impeller of a pump revolves in a stationary case, the pressure in the clearance between the impeller shroud and the case wall is less than the pressure in the volute proper by the amount of the centrifugal head created within this clearance. The latter depends upon the velocity of rotation of water in that space. From the foregoing it is seen that the average velocity in the clearance should be equal to one-half of the impeller speed if the flow is in the viscous region. The measurements of the pressure distribution in that space by the writer and calculation of the average velocity show that this is approximately true in practice. Displacement of the impeller from its central position did not show material difference in the pressure distribution in the pump case as long as the discharge from the impeller did not hit the side walls of the volute, which is indicated by the vibration of the rotating element.

It is possible to combine the authors' Equations [1], [1*a*], and [2] into a single equation by substituting into Equation [2] for k its value in terms of clearance and eccentricity:

$$P = K \frac{\mu l N^2 d^3}{\sqrt{(c^2 - e^2)}}$$

The writer purposely uses a notation slightly different from that of the authors, as follows:

P = power in hp; this symbol is used rather than L_e because in the paper L stands for length; P is the A.S.M.E. symbol for power

c = clearance radial, used instead of h because on the same page h_L is used for head lost, which is more common notation.

e = eccentricity; preferable to Δ, as that has an established meaning of "increment" used in connection with some other letter; e for eccentricity is also A.S.M.E. symbol

$K = 1.9 \times 10^{-8}$ if μ is in poises, d and l in inches, and c and e in thousandths of an inch

$K = 1.565 \times 10^{-5}$ if μ is in lb-sec per sq ft and d, l, c, and e are in feet.

The advantages of this form of equation are that (1) the necessity of dual notation is eliminated, the system of units affecting the value of the constant only, and (2) it covers both cases with shaft concentric and eccentric—in the first case $e = 0$.

From the inspection of the equation it is seen that the power absorbed by the bearing is proportional to the absolute viscosity, length of the bearing, and square of the speed and cube of the diameter, and inversely proportional to the radial clearance for concentric bearing. For eccentric bearing $\sqrt{(c^2 - e^2)}$ stands for the radial clearance, which is a side of a right triangle constructed on c as a hypotenuse and e as the other side.

The increase in power with the decrease of radial clearance will be easily visualized if one would keep in mind a model consisting of two rubber blocks of the same size but of different height or thickness. Now, if the bases of both blocks are fixed and forces are applied to the upper surfaces, it is readily seen that to produce the same displacement of the upper surfaces the thin block will require a greater force.

The decrease of power to drive a bearing on account of film incompleteness is very interesting. Similar results were observed with horizontal shaft but to much greater degree. With thin film lubrication the actual decrease in power was observed with an increase of viscosity of lubricant. This is explained by the fact that with low oil viscosity the intermeshing of asperities on the two surfaces takes place (that is how the wear of bearings and shafts occurs), while with increased viscosity the bearing surfaces are separated by a film of oil. It seems that with the apparatus used by the authors the possibility of partial solid friction is excluded.

It should be noticed that Equation [1] is limited in its use by the assumptions upon which it is based. Thus at high velocities or large clearances the inertia forces come into play, and the flow changes from viscous to turbulent. One particular case is of interest—that is when the clearance is equal to infinity or when the shaft is immersed into liquid. The Equation [1] gives zero for friction loss, which is not true in practice.

The writer had some difficulty in using Equations (1) and

[10] Designing Engineer, Byron Jackson Company, Berkeley, Calif. Mem. A.S.M.E.

(1*a*) for calculation of a few examples. The results obtained with the formula (1*a*) are 60 times greater than those using Equation (1). Upon checking, it was found that the constant in Equation (1*a*) is in error. Instead of 3.67×10^{-8}, this constant should be 6.12×10^{-10}. [NOTE: The authors have made this correction in Equation (1*a*) of the paper, and also have inserted a new Fig. 10, to correspond to a correct Equation (1*a*).]

It may be well to mention here the conversion factors for the absolute and kinematic viscosity from metric to English units for convenience, as those are missing in standard reference books:

1 To convert the absolute viscosity μ in poises (dynes-sec/cm²) to metric gravitational (kg-sec/Mt^2), divide by 98.1.

2 To convert μ in poises to English absolute units (poundal-sec/ft²), divide by 14.88.

3 To convert μ in poises into English gravitational (lb-sec/ft²), divide by 478.

4 To change the kinematic viscosity from metric (cm²/sec) to English (ft²/sec), divide by 929.

It has been found on many occasions, especially with very deep oil wells, that the field-pump efficiency is appreciably better than expected from the factory test. In the case of deep oil wells, this is explained by the pumping effect of the gas contained in oil (gas lift); in the case of water deep-well pumps with oil-shaft lubrication, this may be caused by the leaky inner tubing and shaft running on water instead of oil. It takes less power to revolve a shaft totally immersed in water than a shaft enclosed in a tubing and running on oil.

The authors have thrown light on the problem of lubrication of vertical bearings under idealized conditions. In deep-well pump practice with very long shafts and a great number of bearings, deviations from these ideal conditions occur, such as incompleteness of film, eccentricity of shaft, side load due to shaft misalignment and crooked wells, and leaky joints in tubing resulting in water lubrication of some bearings instead of oil, so that it is natural to expect that the calculated shaft friction loss may differ from the actual. Compared with limited and scattered test data with long shafts, it seems that the friction horsepower, as calculated by Equation [1], is somewhat too high. The incompleteness of the oil film is probably the explanation for this. Systematic tests with long shafts under actual operating conditions can indicate the way to correct the theoretical formula, probably by attaching to it a single experimental coefficient.

AUTHORS' CLOSURE

Dr. Newkirk's discussion indicates one of the directions which further experimentation might take. The cause of "shaft whirling" would probably not be the condition of oil supply in the case of deep-well pumps. The mechanical processes involved in producing a long shaft and bearing housing assembled with threaded joints do not allow perfect construction, but probably result in shafts and bearing supports which are slightly bent. The shaft in such a case would probably not run with a fixed eccentricity, but would tend to "whirl" due entirely to mechanical imperfections. The authors would be glad to receive more detailed suggestions as to methods for determining the effect of "shaft whirling" upon the power required for lubrication.

The equation to which Mr. Hersey refers is:

$$f = \frac{\pi^2 \frac{D}{c} \cdot \frac{\mu N}{p}}{\sqrt{\frac{x}{c}\left(2 - \frac{x}{c}\right)}}$$

in which

f = coefficient of bearing friction
D = shaft diameter
c = mean radial clearance
x = minimum film thickness
μ = coefficient of viscosity
N = speed of rotation
p = bearing pressure.

As noted in the paper, the radial load p is zero. Inserting this value of p in the foregoing equation, one obtains $f = \infty$. Inasmuch as the assumptions upon which the equation is based are not stated, one is not entitled to eliminate p algebraically by combination with other equations of the same article. It therefore seems that the original presumption of no previous theoretical treatment is justified.

Mr. Williams cites some of his personal experiences which seem to discredit the lubrication of deep-well pump shafts with oil as a sound practice. He seems to imply that water-lubricated rubber bearings are the best for such service. The pumps to which he refers, built 23 years ago, were of the open-column type in which the bearings and shaft were exposed to the grit in the water. Such construction is no longer used in this country except for special cases in which the presence of oil in the water is objectionable. The accepted practice is to inclose the shaft and bearings in a tube which serves the three purposes of supporting the bearings, distributing the lubricating oil, and protecting the shaft and bearings from grit. Thus the testimony of practice is that the oil-lubricated bearing is preferred to the water-lubricated type. Furthermore, the experiments of Professor Langille, of the University of California, indicate that although the rubber bearing is very durable when exposed to much grit, the shaft in such cases is subjected to extreme wear. Unfortunately Mr. Williams' figures in regard to the friction losses in bearings combine the vertical-guide-bearing loss with the thrust-bearing loss. As the figures quoted are for pumps more than 20 years old, we obtained for comparison the following figures on more modern pumps from the Byron Jackson Company, one of the larger manufacturers of deep-well pumps:

Case 1, 40-hp unit with 250-ft setting, 450-ft total head, 1750 rpm speed. In this case the thrust was about 7000 lb, the thrust-bearing loss 3 hp with Kingsbury thrust and 1 hp with ball-bearing thrust. The estimated guide-bearing loss is $1^1/_2$ hp, making the total loss in two cases $2^1/_2$ and $4^1/_2$ hp. This represents $6^1/_4$ and $11^1/_8$ per cent of the power input.

Case 2, 250-hp unit, 700-ft setting at 1200 rpm. The thrust in this case would be about 14,000 lb, the thrust-bearing losses 5 hp, and the guide-bearing loss 10 hp. The total loss in this case is 15 hp, or about 6 per cent of the input.

From these figures it would seem that the loss quoted by Mr. Williams is too small to account for even the thrust-bearing loss, and, further, these figures indicate the order of importance of the whole investigation. Mr. Williams' other remark concerning the wear on one side of the shaft is very interesting. Probably such behavior is due to a bent shaft, since obviously pure torsion would not deflect the shaft.

Mr. Stepanoff points out certain features of the paper which appear to be very important to one in the pump-manufacturing business. In the interests of brevity the original paper was devoted entirely to the essentials of the experimental problem. However, from Mr. Stepanoff's remarks one can not avoid being impressed with the fact that the subject of shaft lubrication is a vital one to the designer of such machinery.

Two errors in the paper as originally printed have been called to attention. The first one is the value of the constant in Equation (1*a*), which should be 6.12×10^{-10} in place of 3.67×10^{-8}. The second relates to Fig. 10, which has been redrawn to eliminate the errors that appeared in the original.

The questions raised in the discussion should lead to further investigation in the field of deep-well pump lubrication.

RP-55-1

Performance of Cutting Fluids in Drilling Various Metals

Progress Report No. 4 of the Sub-committee on Cutting Fluids[1] of the A.S.M.E. Special Research Committee on the Cutting of Metals

By O. W. BOSTON,[2] ANN ARBOR, MICH., and C. J. OXFORD,[3] DETROIT, MICH.

This paper gives the results of a series of drilling tests on nine different metals using eleven different cutting fluids. Modern twist drills having 30-deg helices, ranging from 1/2 in. to 1 1/2 in. in diameter, were used. The materials consisted of a cast aluminum alloy, free-cutting brass, cast iron, malleable cast iron, S.A.E. 1020 steel, S.A.E. 1035 steel, carbon tool steel, S.A.E. 3150 steel, and S.A.E. 1112 steel. The eleven cutting fluids consisted of dry cutting, borax water, two soluble oils, lard oil, two straight mineral oils, two compounded oils, and two sulphurized oils.

For each material a formula for torque and thrust was developed as a function of variable diameter and variable feed. Constants for these formulas for dry cutting and each of the eleven cutting fluids are determined. The influence of the different cutting fluids on each metal is shown. With the net output power for drilling dry expressed as 100 per cent, the savings made are as follows:

- **33.3 per cent for S.A.E. 1020 steel when using a sulphurized oil**
- **31.7 per cent for aluminum alloy when using a 90 per cent mineral–10 per cent lard oil**
- **26.0 per cent for S.A.E. 1035 steel when using a sulphurized lard-mineral oil**
- **23.0 per cent for S.A.E. 3150 steel when using a sulphurized mineral oil**
- **21.0 per cent for carbon tool steel when using sulphurized mineral oil**
- **17.2 per cent for cast iron when using a mineral oil containing 5 per cent oleic acid**
- **10.5 per cent for malleable cast iron when using any kind of an oil, such as lard, mineral, compounded, or sulphurized**
- **6.8 per cent for S.A.E. 1112 steel when using a sulphurized lard-mineral oil.**

No appreciable difference is made whether free-cutting brass is cut dry or with any kind of a cutting fluid.

In the development of the formulas for torque and thrust, it was found that the same results of the exponents for drill diameter and feed would be obtained regardless of the diameter of the drill used in the variable feed tests and regardless of the feed used in the variable drill diameter tests. The values of these exponents are not influenced by the type of cutting fluid used. The torque and thrust lines as a function of feed or diameter as the variable are merely displaced vertically, if affected at all. The constants of the torque and thrust formulas, however, indicate the cutting characteristics of the cutting fluids.

The torque and thrust formulas for all kinds of steels as covered in the paper are found to be identical except for different values of the constant. Other metals, however, produce formulas apparently characteristic of that metal. In other words, different formulas as summarized in Table 4 are developed for the aluminum alloy, the free-cutting brass, cast iron, malleable iron, and steel.



O. W. Boston



C. J. Oxford

THIS REPORT covers an extension of Progress Report No. 3, "Performance of Cutting Fluids," presented at the Hartford, Conn., meeting of the Society in June, 1931. In that report an attempt was made to determine the characteristics of each of eleven cutting fluids, including dry cutting, when drilling and planing an annealed chrome-nickel steel corresponding to S.A.E. 3150.

The present paper is confined to drilling tests using the same eleven cutting fluids, in which eight additional metals have been cut. These metals were selected to represent various classes,

[1] The personnel of the Sub-committee consists of: C. J. Oxford, Chairman; O. W. Boston; F. C. Spencer, *ex-officio*, Chairman of the Special Research Committee on the Cutting of Metals.

[2] Professor, College of Engineering, University of Michigan. Mem. A.S.M.E. Orlan W. Boston was graduated from the Univerity of Michigan Engineering College in 1913, received a Master's degree in 1917, and the degree of Mechanical Engineer in 1926. After graduation he was engaged at the university as instructor in engineering mechanics and mechanical engineering for four years. In October, 1917, he was commissioned in the U. S. Navy and assigned to duty in the Bureau of Ordnance on design and manufacture of submarine mines used in the North Sea blockade. From 1919 to 1921 he was engaged in industrial engineering work for the Cleveland Tractor Company, in Cleveland, as assistant to the vice-president and works manager. In the fall of 1921, he returned to the University of Michigan, where he is professor of shop practice and director of the department of engineering shop. He is author of many papers dealing with the subject of metal cutting and is serving as an A.S.M.E. representative on the Specifications for Tool Steel Committee of the American Society for Testing Materials.

[3] Chief Engineer, National Twist Drill & Tool Company. Assoc. Mem. A.S.M.E. Carl J. Oxford was born in Norway and came to the United States in 1909. After two years in the engineering department at the University of Michigan, he served as tool draftsman with the Packard Motor Car Company, Dodge Brothers, and Nordyke & Marmon until 1916, when he became designing engineer for the Wilt Twist Drill Company. The following year he became associated with the National Twist Drill & Tool Company.

Contributed by the Special Research Committee on the Cutting of Metals and presented at the session of the Machine Shop Practice Division at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

and consist of a cast aluminum alloy, a free-cutting brass, cast iron, malleable cast iron, a free-cutting steel, and low-, medium-, and high-carbon steels.

The tests were confined to power determinations in which drill torque and thrust, as measured on a dynamometer, and input power, as measured on a wattmeter, were determined. Drills ranging from 1/2 in. in diameter to 1 1/4 in. in diameter were used. They were operated at a peripheral speed of 60 fpm and at feeds corresponding to commercial practice for steel for the respective diameters. Formulas representing the torque and thrust are developed for all materials as a function of drill diameter and feed.

The exponents for drill diameter and feed are determined for each of the materials drilled, as well as the constants for each cutting fluid used on each material.

The drill press and the equipment used in the tests are shown in Fig. 1.

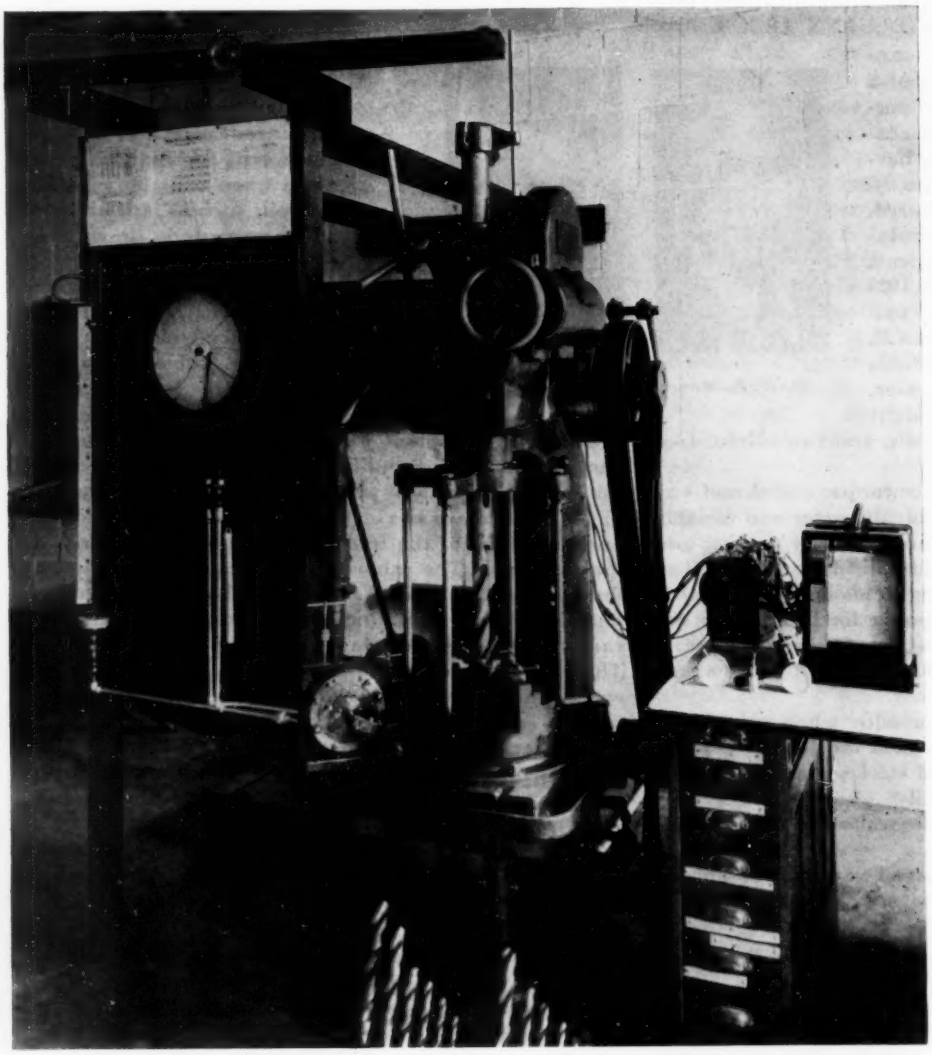


FIG. 1 DRILL PRESS AND EQUIPMENT USED IN TESTS

THE CUTTING FLUIDS

The cutting fluids used in these tests are the same as those used in the previous report. They consist of eleven cutting fluids selected to represent the various classes used commercially. They were described in detail in Report No. 3, but are summarized in Table 1, in which the number, composition, and various properties of each cutting fluid are given.

THE MATERIALS CUT

In Report No. 3, an annealed chrome-nickel steel, corresponding to S.A.E. 3150, was used exclusively. In these tests a limited variety of ferrous and non-ferrous metals were selected to give as good a picture as possible of the performance of the various cutting fluids used. These materials, together with their bar designations, heat treatments, chemical analyses, hardness numbers, and, when available, their physical properties, are listed in Table 2. Photomicrographs at 100 magnifications of all materials used are shown in Figs. 2 to 10, inclusive. These photomicrographs were made from a surface 1/2 in. beneath the outside surface.

The aluminum alloy, corresponding to S.A.E. 33, was selected as that drilled most commonly in this class. It was procured from the University of Michigan foundry in chilled cast bars 1 1/2 in. wide by 2 in. deep and 14 in. long. The bars were dense and uniform, showing no blowholes or other imperfections. The photomicrograph is shown in Fig. 2.

The brass was of the leaded free-cutting variety commonly used for screw-machine work. It was obtained in bars 1 3/4 in. square, extruded to 1 13/16 in. square, and then drawn to finished size. It was sawed to convenient lengths of 14 in. The photomicrograph of this free-cutting brass is shown in Fig. 3.

The soft cast iron tested contained about 1 per cent nickel and 2.62 per cent silicon. It was cast in the University of Michigan foundry in bars 2 in. by 4 in. in section and 14 in. in length. The photomicrograph is shown in Fig. 4.

The malleable cast iron was furnished gratis by the Cadillac Malleable Iron Company in the form of bars 2 in. by 4 in. in section by 14 in. long. The hard iron contained 2.46 per cent total carbon and 1 per cent silicon, and represented a type in common use. The bars were annealed in the front thermocouple pot of the company's No. 2 oven. The annealing cycle was as follows: 40 1/4 hr to heat up to 1550 F, 48 hr at 1550–1625 F, 15 hr cooling to 1500 F, 12 hr cooling to 1400 F, 19 hr cooling to 1300 F, and 18 1/4 hr cooling to 1200 F. To cool from the annealing temperature to 1200 F required 64 1/4 hr. The analysis of the hard

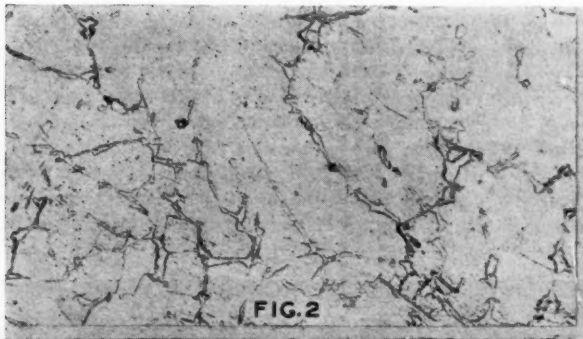
TABLE 1 PROPERTIES OF CUTTING FLUIDS USED

No.	Composition	Base of mineral oil	Per cent compounding by analysis	Viscosity at deg F: 100	130	210	A.P.I.[4] gravity at 60 deg F	d specific gravity	Temperature, deg F: Pour	Flash	Fire	N.P.A.[5] color	C Specific heat[2]: 100 deg F	210 deg F	430 deg F	k Thermal conductivity[3]: 100 deg F	210 deg F	430 deg F
1	Dry cutting	.....	.....	...	...	..	..	....	.....	.....	...	....	....	....	....	....	....	....
2	1½ per cent by weight of borax in water	.....	1.19 borax	...	...	..	..	....	.....	.....	...	....	....	....	....	....	....	....
3	1 part soluble oil in 50 parts water	.....	2.3 soluble oil	31	...	..	..	....	.....	.....	...	....	....	....	....	....	....	....
4	1 part soluble oil in 10 parts water	.....	8.37 soluble oil	32	...	..	..	....	.....	.....	...	....	....	....	....	....	....	....
5	No. 1 lard oil	.....	12.23–13.6 F.F.A.[1] 86.4 fat	200	110	..	23.7	0.9117	45	445	...	4	0.4510	0.5149	0.6438	0.8738	0.8448	0.7852
6	Light mineral oil	Mixed	.....	111	69	40.5	23.5	0.9129	0 to −30	320–305	375	2–2¼, 2½–3	0.4500	0.5142	0.6430	0.8724	0.8440	0.7845
7	Heavy mineral oil	Mixed	.....	209	104	45.5	23.0	0.9159	0 to −15	340–350	390	2¾–3, 3	0.4490	0.5137	0.6417	0.8695	0.8400	0.7818
8	10 per cent No. 1 lard oil in mineral oil	Paraffin	10.12 lard oil	101	67	41	29.4	0.8794	25	355	...	3	0.4630	0.5290	0.6612	0.9052	0.8750	0.8145
9	5 per cent oleic acid in mineral oil	Paraffin	4.90 oleic acid	97	65	40.5	29.2	0.8805	25	295	...	3.5	0.4624	0.5284	0.6610	0.9048	0.8748	0.8138
10	Sulphurized mineral oil	Asphaltic	4.38 sulphur	105	67	40	21.6	0.9242	0 to −25	285	...	Black	0.4459	0.5096	0.6374	0.8620	0.8344	0.7752
11	Sulphurized lard oil blended with 5 parts mineral oil	Mixed	2.34 sulphur 4.43 F.F.A. sulphurized 13.94 fat sulphurized	102	67	41.5	25.4	0.9018	25 to 20	355	...	Black	0.4542	0.5190	0.6490	0.8837	0.8548	0.7942

[1] F.F.A. is free fatty acids.
[2] Specific heat C computed from formula $C = (t + 670)(2.10 - d)/2030$, in which t = temperature of fluid deg F, and d = specific gravity of oil at 60 deg F.
[3] Thermal conductivity k computed from formula $k = (0.813/d)\,[1 - 0.0003(t - 32)]$, in which d = specific gravity of the oil at 60 deg F, t = temperature of fluid in deg F; k is in units of Btu per hr per sq ft per in. per deg F.
[4] American Petroleum Institute.
[5] National Petroleum Association.

TABLE 2 THE MATERIALS DRILLED

Bars No.	Bars Section	Material	Heat treatment	Analyses	Hardness numbers: Rockwell 100 kg, 1/16 in. ball	Hardness numbers: Brinell	Physical properties: Yield point, lb per sq in.	Physical properties: Ultimate strength, lb per sq in.	Physical properties: Elongation per cent in 2 in.
AF	1½ × 2	Aluminum alloy S.A.E. 33	Chill cast	91.0 Al, 8.0 Cu, 1.0 impurities	37.5	70	....	.....	..
BF	1¾ × 1¾	Leaded free-cutting brass	Extruded and drawn	61.75 Cu, 35.0 Zn, 3.21 Pb, 0.04 Fe	66.0	100	34,700	51,800	43.5
CF	1¾ × 4	Cast iron	Cast	3.44 C, 2.62 Si, 0.986 Ni, 0.478 Mn	83.5	179	....	.....	..
DF	1¾ × 4	Malleable cast iron	40¼ hr to 1550 F, 48 hr at 1550–1625 F, 64½ hr to cool to 1200 F	Analysis of hard iron: 2.46 C, 0.26 Mn, 0.163 P, 0.071 S, 1.00 Si	71.0	137	....	.....	..
KF	2 × 6	S.A.E. 3150 steel	Annealed	0.50 C, 0.68 Mn, 0.022 P, 0.023 S, 0.62 Cr, 1.16 Ni	86.0	196	66,140	114,200	26.0
EF	1¾ × 4	S.A.E. 1020 steel	Annealed	0.22 C, 0.52 Mn, 0.011 P, 0.026 S	65.0	131	....	.....	..
FF	1¾ × 4	S.A.E. 1035 steel	Annealed	0.38 C, 0.53 Mn, 0.017 P, 0.031 S, 0.18 Si	75.0	156	....	.....	..
GF	4 × 6	Carbon tool steel	Annealed	0.97 C, 0.31 Mn, 0.025 P, 0.027 S, 0.10 Si	85.0	152	36,500	77,350	30.0
MF	1½ × 1½	S.A.E. 1112 steel	Cold drawn	0.09–0.13 C, 0.7–0.9 Mn, 0.085–0.120 S, 0.08–0.40 P	89.5	217	....	.....	..


FIG. 2

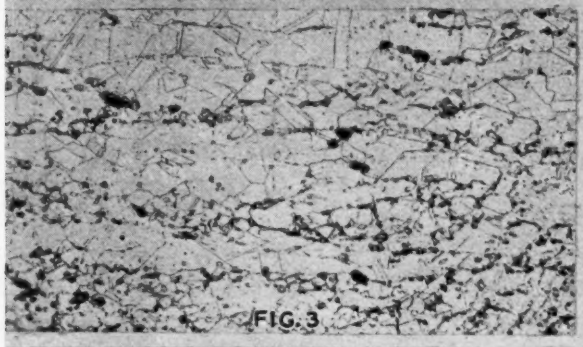
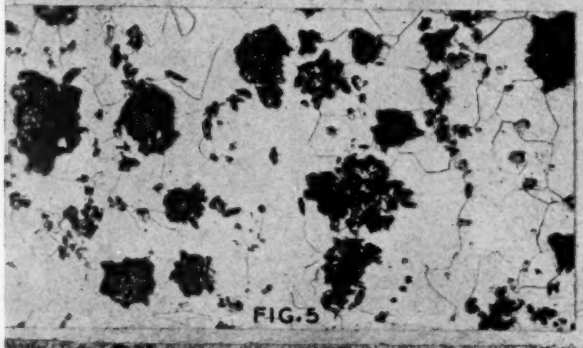
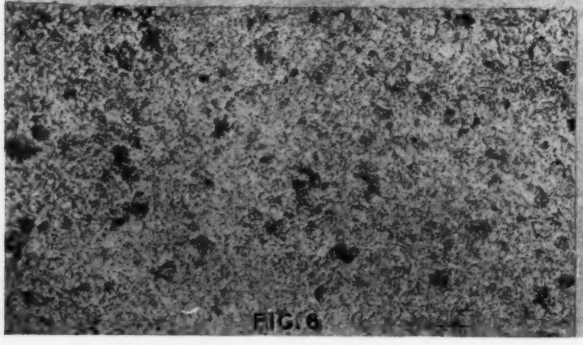
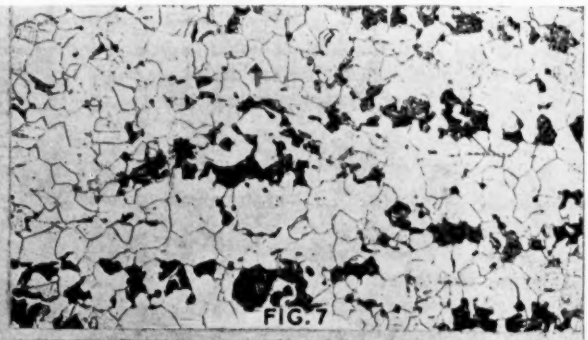
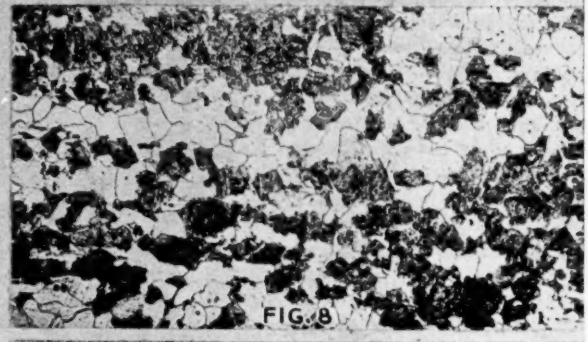

FIG. 3

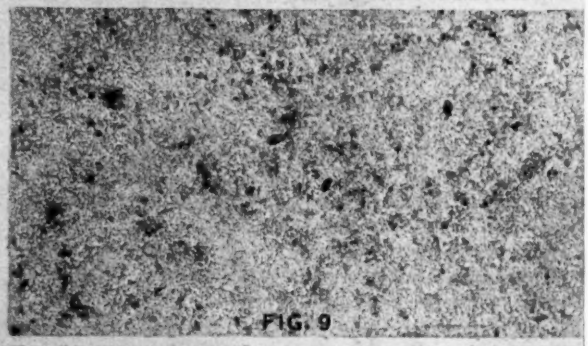

FIG. 4

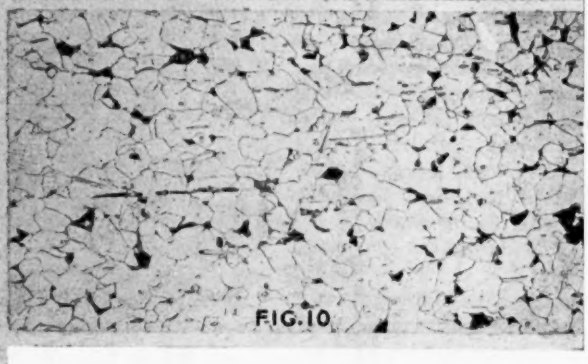

FIG. 5


FIG. 6


FIG. 7


FIG. 8


FIG. 9


FIG. 10

FIGS. 2 TO 10 PHOTOMICROGRAPHS OF ALL MATERIALS USED (× 100)

(Fig. 2, cast aluminum alloy, S.A.E. 33, specimen AF, etched with 5% HF; $CuAl_2$ network in Al matrix. Fig. 3, free-cutting cold-drawn brass, specimen BF, etched with NH_4OH and H_2O_2; alpha plus beta. Fig. 4, cast iron, specimen CF, etched with 4% nital; normal pearlitic and graphitic structure. Fig. 5, malleable cast iron, specimen DF, etched with 4% nital; nodular graphite and ferrite with numerous small inclusions. Fig. 6, alloy steel, S.A.E. 3150, specimen KF, etched with 4% nital; fine-grained, fine pearlite. Fig. 7, carbon steel, S.A.E. 1020, specimen EF, etched with 4% nital; ferrite and pearlite, medium grain size. Fig. 8, carbon steel, S.A.E. 1035, specimen FF, etched with 4% nital; ferrite and pearlite, medium grain size. Fig. 9, carbon tool steel (0.97 per cent carbon), specimen GF, etched with 4% nital; well-spheroidized structure. Fig. 10, free-cutting steel, S.A.E. 1112, specimen MF, etched with 4% nital; ferrite and pearlite, medium grain size.)

iron, that is, before annealing, is given in Table 2. A photomicrograph of the malleable cast iron is given in Fig. 5.

The annealed chrome-nickel steel, S.A.E. 3150, used in this test is the same as that used in the test for Progress Report No. 3. Certain data from that report are included in this report for purposes of comparison and to make the information contained in this report complete. These bars were furnished in sections 2 in. by 6 in., but for convenience in drilling were cut to 14 in. lengths. The photomicrograph of the S.A.E. 3150 steel is given in Fig. 6.

The low-carbon steels, S.A.E. 1020 and S.A.E. 1035, were furnished as long, annealed, smooth forgings, having a 2 in. by 4 in. cross-section. The carbon contents were 0.22 per cent and 0.38 per cent, respectively. They were selected as being representative of steels commonly used in production work. The carbon tool steel tested had a carbon content of 0.97 per cent and was in an annealed condition. Photomicrographs of the three carbon steels are shown in Figs. 7, 8, and 9, respectively.

A free-cutting high-sulphur steel, corresponding to S.A.E. 1112, also was included in these tests, inasmuch as this type of steel is commonly used in connection with cutting fluids in screw-machine work. It was obtained in the form of cold-drawn bars $1^1/_2$ in. square. A photomicrograph of the S.A.E. 1112 steel is shown in Fig. 10.

THE DRILL PRESS AND THE DYNAMOMETER

The testing equipment used for determining the influence of the cutting fluids in drilling is the same as that described in Progress Report No. 3. The drill press was a No. 2 manufacturing-type Colburn press, driven through a short belt by a 5-hp, 220-v, 3-phase, 60-cycle motor. A gear pump was provided for raising the coolant from the tank at the side of the column to the work and drill. The cutting fluid was supplied in as large quantities as possible and dumped at a low velocity upon the drill just above the work. No drill guide bushing was used.

The testing equipment consisted of a drill dynamometer with gages and recording wattmeter. The drill dynamometer was such that the torque and thrust as developed at the drill point were recorded graphically. The chart of the recording gage was made to rotate so as to register the torque and thrust as a function of the travel of the drill. The method of calibrating the torque and thrust scales is described in detail in an earlier paper entitled, "Power Required to Drill Cast Iron and Steel."[4] Actual cutting speeds were determined during each test, using a speed counter. The wattmeter recorded the tare input power developed by the machine running with the speed and feed gears engaged, but before the drill was cutting, and the gross power input while cutting. The drill point was brought to within $^3/_8$ in. of the surface of the work to be drilled, after which the feed clutch was engaged. The drill press was started and the tare power was recorded on the wattmeter chart while the drill was fed the $^3/_8$ in. to the work. As the drill penetrated the work, the total power also was recorded on the wattmeter chart. The tare power deducted from the gross power gives the net power input. This corresponds to the power output at the drill point, except for the additional frictional load on the gears and bearings of the machine due to the cutting load. This frictional power loss is least for light loads and high speeds and greatest for high loads and low speeds. The efficiency, as determined by dividing the net power output, computed from the torque and thrust by the net power input, determined by the wattmeter, is about 90 per cent for the $^1/_2$-in. drill rotating at 440 rpm and a feed of 0.009 in. per revolution, but is only about 75 per cent for the $1^1/_4$-in. drill at 174 rpm and at a feed of 0.015 in. per revolution.

[4] Trans., A.S.M.E., 1930, MSP-52-2.

THE DRILLS USED

The drills used in these experiments were selected as being representative of the drills corresponding with standard specifications for taper-shank, high-speed-steel twist drills. They were selected in sizes $^1/_2$ in., $^3/_4$ in., 1 in., $1^1/_4$ in., and $1^1/_2$ in. in diameter, and correspond to the "R" drills described in Progress Report No. 3 and a previous paper.[4] The various angles and dimensions of these drills as actually used in the experiments are shown in Table 3. The actual revolutions per minute used for a set-up of approximately 60 fpm linear speed and the feed in inches are shown for each drill size.

RESULTS OF THE DRILLING TESTS

Since the object of this investigation was to determine and compare the characteristics of the various cutting fluids on each of the materials selected, it was thought best to standardize as much as possible on the tests. Therefore, each drill size used was run at a peripheral speed as near as possible to 60 fpm and at a commercial feed as indicated in Table 3.

TABLE 3

Drill Diam.	No.	Web thickness, in.	Helix angle, deg	Point angle, deg	Chisel-edge angle, deg	Clearance at periphery, deg	Actual rpm	Fpm	Feed, in.
$^1/_2$	R	0.081	29	121	129	5–20	449	58.8	0.009
$^3/_4$	R	0.104	31	121	134	5	300	59.0	0.012
1	R	0.141	31	121	136	5	231	60.5	0.013
$1^1/_4$	R	0.180	30	121	138	5	174	57.1	0.015
$1^1/_2$	R	0.213	31–20	121	138	5	151	59.4	0.015

When drilling aluminum and free-cutting brass, however, in order to get values of torque and thrust large enough to measure without an appreciable error, the feeds actually used for each drill size were double those shown in Table 3.

All drills, $^1/_2$ in. to $1^1/_4$ in. in diameter, inclusive, were used in Progress Report No. 3 on the S.A.E. 3150 steel, bar KF, and in this paper on the carbon tool steel, bar GF, which was the first material tested. The results from both of these materials were so uniformly accurate that it was decided, in order to save time, to eliminate the $^3/_4$-in.-diam drill from the tests on the rest of the materials tested. The experimental data presented for cast iron, S.A.E. 1020 steel, and S.A.E. 1035 steel were determined from the $^1/_2$-in., 1-in., and $1^1/_4$-in.-diam drills at their respective speeds and feeds. From these tests, however, it was concluded that the low values of the torque on the $^1/_2$-in. drill made it difficult to read the values with satisfactory accuracy, so that on the rest of the tests, the $^3/_4$-in., 1-in., and $1^1/_4$-in. drills were used, with what the authors believe to be complete satisfaction. The experimental results obtained from each of these materials are discussed separately below and are then finally compared.

Tests on Aluminum

It was found that the formulas for torque and thrust when drilling the S.A.E. 3150 steel, as previously reported, did not hold for aluminum. It was thought desirable, therefore, to determine formulas for aluminum. In Fig. 11 are shown the results of a series of tests to determine correct formulas for aluminum. The variable-feed tests were run with the $1^1/_4$-in.-diam drill, using feeds of 0.015 in., 0.021 in., 0.030 in., and 0.043 in. The values of feed are greater than those generally used on steel in order that the torque and thrust may be sufficiently large to be read with accuracy. The values of feed as used were selected to give approximately equal increments between plotted points on the log-log curve in Fig. 11.

When cutting dry, the torque values, as determined from

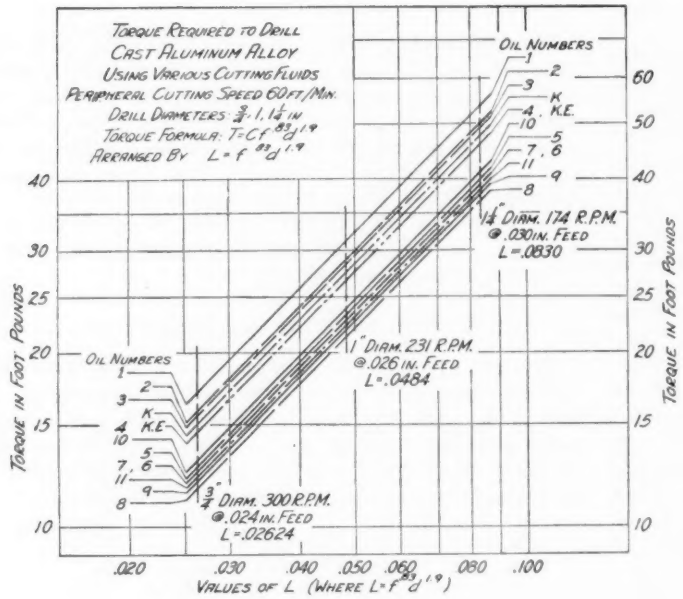


Fig. 12

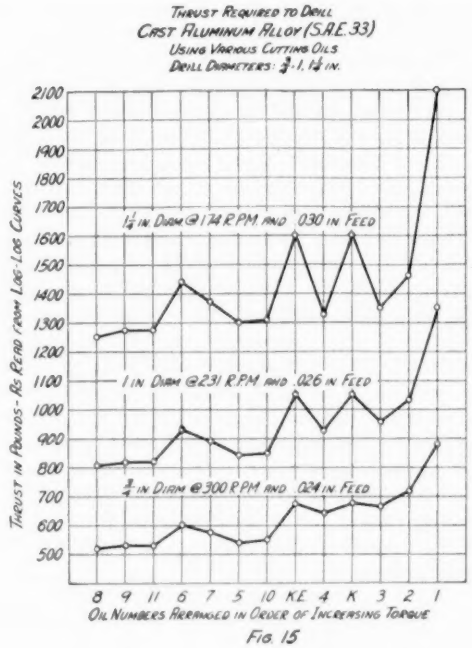


Fig. 15

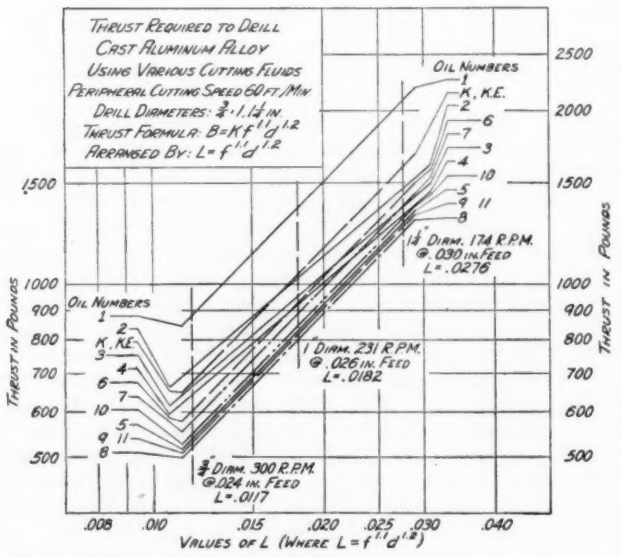


Fig. 13

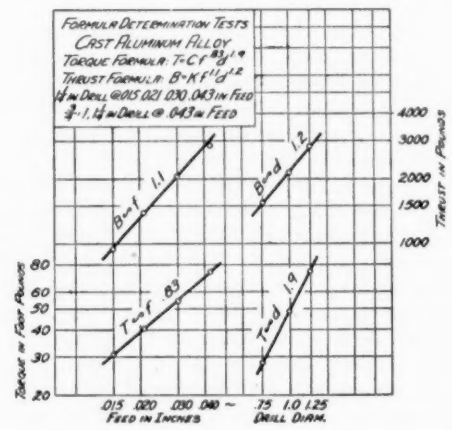


Fig. 11

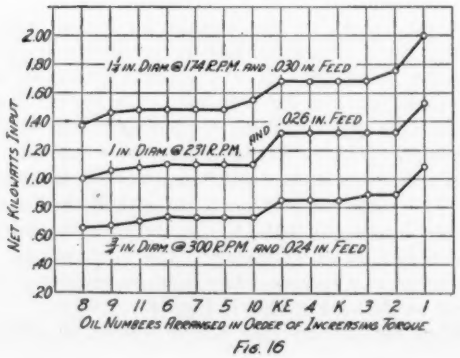


Fig. 16

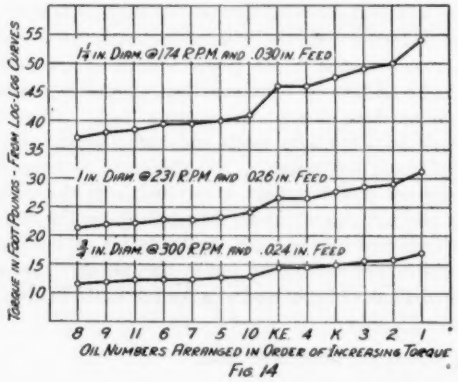


Fig. 14

these tests as a function of the feed, fall very satisfactorily on the straight line indicated, which has a slope with the horizontal of 0.83. The thrust values also indicate a straight line when plotted over the variable-feed values. The slope of this line is 1.1.

For the variable-diameter tests, a feed of 0.043 in. was selected, and both torque and thrust values as plotted over the variable diameter on the log-log paper in Fig. 11 give satisfactory straight lines. The slope of the torque line, as a function of the diameter, is 1.9 and that of the thrust, as a function of the diameter, is 1.2. The torque and thrust equations for aluminum indicated are then as follows:

$$T = C f^{0.83} d^{1.9}$$
$$B = K f^{1.1} d^{1.2}$$

in which T is the torque in foot-pounds developed by the drill; C is a constant depending upon the material drilled, the cutting fluid, and the drills; f is the feed of the drill in inches per revolution; d is the diameter of the drill in inches; B is the thrust in pounds developed by the drill; and K is the constant for thrust corresponding to C in the torque formula.

Tests for the determination of the torque and thrust formulas have been run so often on such a wide variety of materials that the formulas as shown, based on one drill diameter for variable feeds and one feed for variable drill diameters, would maintain had other constant feeds and diameters been selected. The curves shown in Fig. 11 would simply have been raised or lowered with the same slope.

After running the tests to determine formulas, tests were then run to compare, under different cutting conditions, the various cutting fluids. Three separate tests were run for each cutting condition—that is, combination of drill diameter and cutting fluid. If the results did not agree closely, additional check tests were run. Many more, however, were run when cutting dry, inasmuch as the original keenness of the cutting edge was removed by dry cutting, so that uniformity in torque and thrust values could maintain as far as the condition of the cutting point of the drill was concerned. The average of the experimental values of torque, thrust, and power input for the $^{3}/_{4}$-in.-diam drill operating at 0.024 in. feed per revolution, the 1-in.-diam drill operating at 0.026 in. per revolution, and the $1^{1}/_{4}$-in.-diam drill operating at 0.030 in. per revolution are shown plotted in Figs. 12 to 16, inclusive. Results are shown for each of the eleven cutting fluids, as well as one or two in addition.

As in Progress Report No. 3, in order that the torque formulas should give straight lines when plotted on log-log paper for any diameter and feed, the abscissa was selected as a variable L, in which L equals $f^{0.83} d^{1.9}$, as taken from the formula for torque already determined. The averages of the experimental values of torque for the different drills and feeds are shown plotted for each cutting fluid in Fig. 12. Lines drawn through the torque values for a given cutting fluid are straight, but those for one cutting fluid may be above or below those for another, indicating that the only influence of the cutting fluid is to displace the lines vertically and not to change their slope, inasmuch as all of them are parallel.

It is apparent that cutting fluids have a marked influence on the power required to drill aluminum. There appear to be three distinct groups of curves: that for drying cutting (No. 1), the water solution (No. 2), soluble oils (Nos. 3 and 4), and kerosene (K), and the oils.

Pure kerosene (K) and the kerosene-soluble oil mixture (KE), consisting of one part kerosene to four parts of No. 3 soluble oil, were introduced as additional cutting fluids in drilling aluminum, inasmuch as they are used extensively commercially.

From Fig. 12 it is seen that cutting dry requires the greatest power, expressed as torque. It also is seen that, of the water and kerosene group, borax water (No. 2) produces the greatest torque, the 1-to-50 soluble oil (No. 3) produces the next greatest torque, pure kerosene (K) a slightly lower torque, and the 1-to-10 soluble oil (No. 4) and the kerosene-soluble oil (KE) mixture the lowest torque of the second group. The oils, as the third group, produce the lowest torque values, and several tendencies may be noted from their order as indicated. The sulphurized mineral oil (No. 10) requires more power than either of the plain mineral oils (Nos. 6 and 7). Also, the sulphurized lard-mineral oil (No. 11) produces higher torque values than the mineral-lard oil (No. 8). It is interesting, also, to note that the oleic acid-mineral oil (No. 9) produces a torque next to the lowest of all, whereas in cutting S.A.E. 3150 steel, presented in Progress Report No. 3, it produces the highest torque. It appears that sulphur in connection with an oil does more harm than good in a cutting fluid used on aluminum. Also, it appears that the mineral-lard oil (No. 8) and the mineral-oleic acid (No. 9) are better than the straight mineral oils (Nos. 6 and 7) and the lard oil (No. 5).

Values of torque for the various combinations of drill diameter and feed have been taken from the straight-line curves of Fig. 12 for each of the cutting fluids tested. Using the equation for torque as developed for aluminum, the values of the constant C in the torque formula have been computed. On several different occasions the authors have determined drilling formulas for other aluminum alloys. It has been found that no two of these are exactly alike. By changing the composition or the method of producing—that is, whether they are cast, chill cast, or cold drawn—the exponents of the feed and drill diameters are changed. It should therefore be kept in mind that the formulas and constants for the various cutting fluids as presented for this aluminum alloy apply with accuracy only to this one material. Insufficient data are available to show the effect of change in composition, etc. on the characteristics of cutting fluids.

In Fig. 13 are shown plotted on log-log paper the averages of experimental values of thrust over the values of L as abscissa, in which L is equal to $f^{1.1} d^{1.2}$, taken from the equation for thrust as developed for this aluminum. Straight lines are obtained for the thrust as a function of the value of L for each of the thirteen cutting fluids. Two sets of parallel lines, one set having less inclination than the other, are shown. Oils Nos. 2, 3, and 4, which consist of the water compounds, have the less slope. The thrust lines for all of the other cutting fluids, including dry cutting, are parallel, but at a greater inclination. From this fact alone it would appear that, where the value of L is high enough, reaching a value of approximately 0.045, the water solutions and soluble oils would give results comparable to the best cutting oils for thrust only. For this reason the thrust formula as developed for this aluminum does not hold exactly for cutting fluids Nos. 2, 3, and 4, but does hold for dry cutting, under which condition it was determined, as well as for all of the oils, including the kerosene-soluble oil mixture. No explanation of this irregularity can be made at this time.

The thrust values for dry cutting (No. 1) are seen to be considerably higher than those of any other cutting fluids. Kerosene and the kerosene-soluble oil mixture are next in order. In general, the magnitude of the thrust places the oils in much the same order as for torque in Fig. 12, with oils Nos. 8 and 9 at the bottom. The fatty oil (No. 5) and the sulphurized oils (Nos. 10 and 11) also are among the lowest on the thrust scale.

Values of thrust for the various cutting conditions for each of the cutting fluids have been taken from the curves of Fig. 13, and from the equation developed for thrust for aluminum, values for the constant K have been computed.

In arriving at values of C and K as given in Table 4, three

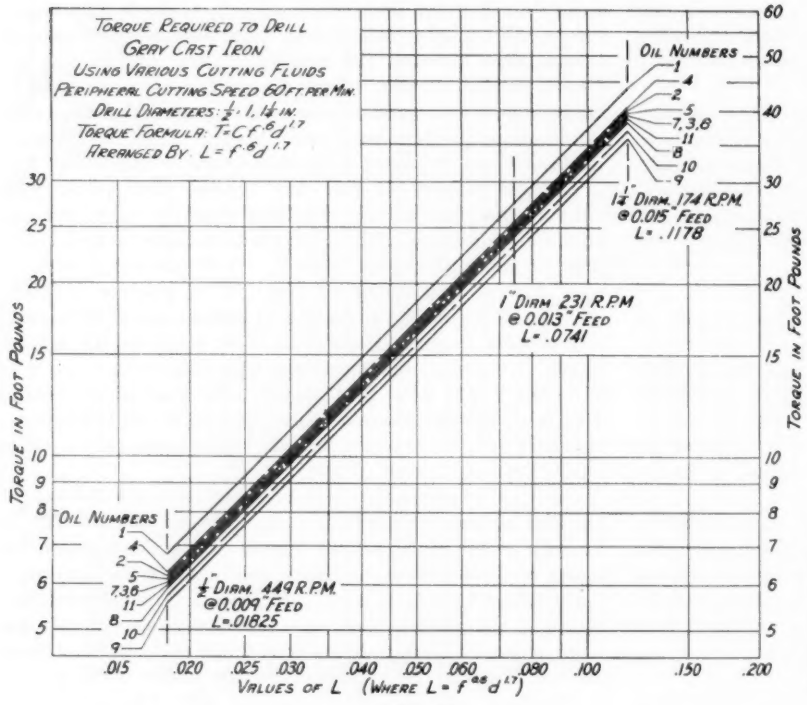


Fig. 20

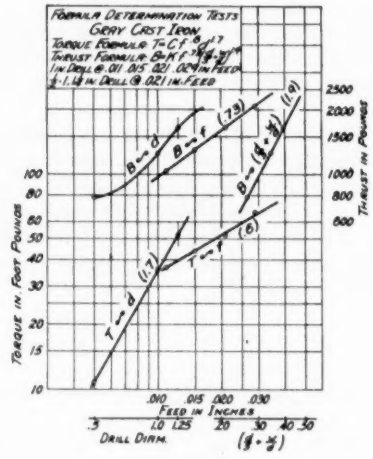


Fig. 19

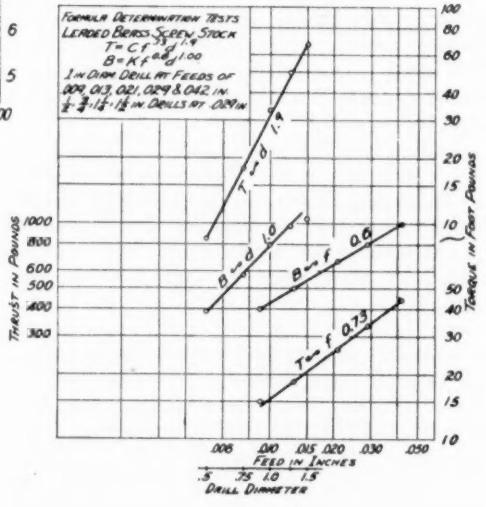


Fig. 17

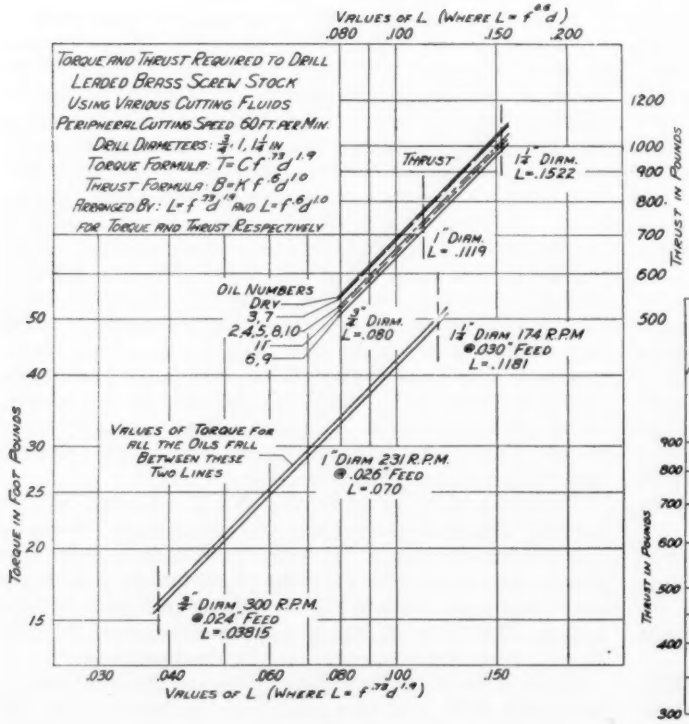


Fig. 18

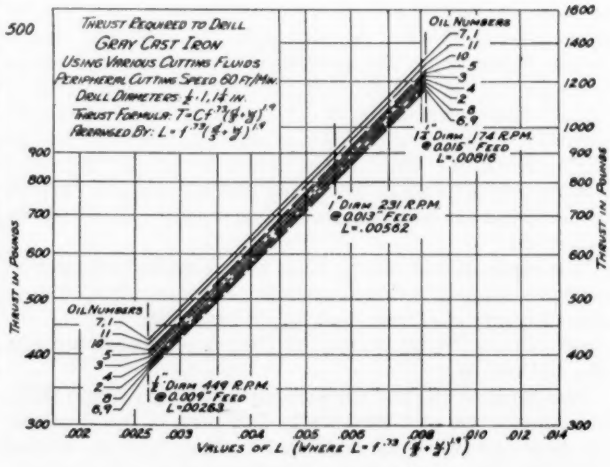


Fig. 21

TABLE 4 SUMMARY OF DRILLING TORQUE AND THRUST FORMULAS

(The constants are computed from values taken from the log-log curves of experimental data)

Cutting fluid No.	S.A.E. 33 aluminum alloy bar AF $T = Cf^{0.83}d^{1.9}$ $B = Kf^{0.1}d^{1.2}$		Leaded-brass screw stock bar BF $T = Cf^{0.73}d^{1.9}$ $B = Kf^{0.6}d^{1.0}$		Cast-iron bar CF $T = Cf^{0.66}d^{1.3}$ $B = Kf^{0.72}\left(\frac{d}{5}+\frac{w}{d}\right)^{1.6}$		Malleable cast-iron bar DF $T = Cf^{0.66}d^{1.3}$ $B = Kf^{0.72}\left(\frac{d}{5}+\frac{w}{d}\right)^{1.16}$		S.A.E. 3150 steel bar KF $T = Cf^{0.73}d^{1.9}$ $B = Kf^{0.87}\left(\frac{d}{5}+\frac{w}{d}\right)^{2.12}$		S.A.E. 1020 steel bar EF $T = Cf^{0.73}d^{1.9}$ $B = Kf^{0.87}\left(\frac{d}{5}+\frac{w}{d}\right)^{2.12}$		S.A.E. 1035 steel bar FF $T = Cf^{0.73}d^{1.9}$ $B = Kf^{0.87}\left(\frac{d}{5}+\frac{w}{d}\right)^{2.12}$		Carbon tool steel annealed bar GF $T = Cf^{0.73}d^{1.9}$ $B = Kf^{0.87}\left(\frac{d}{5}+\frac{w}{d}\right)^{2.12}$		S.A.E. 1112 steel bar MF $T = Cf^{0.73}d^{1.9}$ $B = Kf^{0.87}\left(\frac{d}{5}+\frac{w}{d}\right)^{2.12}$	
	Constants		Constants		Constants		Constants		Constants		Constants		Constants		Constants		Constants		Constants		Constants	
	C for torque	K for thrust	C for torque	K for thrust	C for torque	K for thrust	C for torque	K for thrust	C for torque	K for thrust	C for torque	K for thrust	C for torque	K for thrust	C for torque	K for thrust	C for torque	K for thrust	C for torque	K for thrust	C for torque	K for thrust
1	646	75,030	418	6938	370	160,000	544	152,000	2215	912,500	2186	906,000	1843	795,000	2089	933,000	1222	624,000				
2	601	56,910	418	6636	340	146,500	544	152,000	1937	760,500	1815	772,000	1622	725,000	1848	901,000	1309	613,000				
3	589	53,230	418	6850	334	150,000	544	152,000	1995	780,200	1786	785,000	1622	702,000	1918	927,000	1222	580,300				
4	553	51,070	418	6636	344	148,000	544	152,000	2025	773,000	1758	785,000	1582	711,000	1864	921,000	1222	604,700				
5	482	45,970	418	6636	336	153,000	485	129,700	2100	864,700	1623	711,000	1498	629,000	1840	962,000	1222	613,000				
6	471	51,400	418	6400	334	143,000	485	129,700	2124	796,000	1711	751,000	1560	651,000	1984	942,000	1222	613,000				
7	471	49,270	418	6850	334	161,000	485	129,700	2068	810,200	1670	738,000	1560	651,000	1938	927,000	1222	624,000				
8	443	44,380	418	6636	324	144,000	485	129,700	2164	892,200	1645	738,000	1536	641,000	1953	965,000	1222	613,000				
9	453	45,300	418	6400	306	143,000	485	129,700	2294	927,000	1707	751,000	1536	663,000	1937	902,000	1222	613,000				
10	495	47,000	418	6636	314	154,000	485	129,700	1696	673,200	1457	651,000	1363	651,000	1649	759,000	1162	613,000				
11	463	45,270	418	6530	330	158,000	485	129,700	1779	683,000	1457	651,000	1363	641,000	1715	758,000	1139	613,000				
Kerosene	671	57,830	...	.....	...	.....	...	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....				
Kerosene plus No. 3	553	57,830	...	.....	...	.....	...	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....				

values of torque and thrust were taken from the torque and thrust curves of Figs. 12 and 13, one for the $^3/_4$-in.-diam drill at its speed and feed, a second for the 1-in.-diam drill at its speed and feed, and a third for the $1^1/_4$-in.-diam drill at its speed and feed. Using these values, three values of the constants C and K were obtained. The value shown in Table 5 is an average of these three values. This point is of particular significance in connection with the values of K for cutting fluids Nos. 2, 3, and 4, inasmuch as they do not follow the regular thrust formulas. The value for K, however, as computed for these three fluids, can be used as an average value.

In order to show to better advantage the effect on the torque and thrust values of the various cutting fluids, Figs. 14 and 15 have been prepared. Fig. 14 shows the torque in foot-pounds required for the several drills operating at their respective feeds and speeds, plotted over oil numbers arranged in order of increasing torque. The values plotted are computed and correspond to the values read from the log-log lines in Fig. 12.

Similarly, Fig. 15 shows the thrust in pounds required for the several drills plotted over the oil numbers also in order of increasing torque. If the mineral oils (Nos. 6 and 7) and the two kerosene oils are omitted, the rest give a continuously rising curve from left to right, as do the torque curves. The thrust values for the four oils mentioned are high, in proportion to their torque values. The net power input expressed in kilowatts required for the several drills, when using the thirteen cutting fluids tested, are shown plotted over the oil numbers arranged in order of increasing torque in Fig. 16. These curves correspond favorably to the torque curves of Fig. 14, as might be expected.

Tests on Free-Cutting Brass

As in the case of aluminum, a special series of tests were first run on the free-cutting brass (bar BF) to determine torque and thrust formulas as a function of drill diameter and feed. The results of these tests are shown plotted in Fig. 17. For the variable-feed data, the 1-in.-diam "R" drill was operated at feeds of 0.009, 0.013, 0.029, and 0.042 in., respectively. The cutting was done dry.

The torque and thrust values, as plotted in Fig. 17, give straight lines on the log-log paper. The exponent of the feed in the torque formula was found to be 0.73 and that of the feed in the thrust formula was found to be 0.6. With a constant feed of 0.029 in., additional tests were run with the $^1/_2$-in., $^3/_4$-in., 1-in., $1^1/_4$-in., and $1^1/_2$-in.-diam drills. The torque and thrust for these tests also are plotted in Fig. 17. These values give straight lines from which the exponent of the diameter in the torque formula is found to be 1.9 and that in the thrust formula is found to be 1.0. The drilling formula for the free-cutting brass (bar BF) is therefore as follows:

$$T = C f^{0.73} d^{1.9}$$
$$B = K f^{0.6} d^{1.0}$$

The values of the constants C and K are summarized in Table 4 for all of the cutting fluids, as well as dry cutting.

The average experimental values for torque and thrust are plotted for the different drill diameters and cutting fluids in Fig. 18. It is interesting to note from the torque curves that the cutting fluids have very little influence on the value of the torque when drilling this free-cutting brass. The minimum value of the torque is 49.5 when drilling dry with the $1^1/_4$-in.-diam drill at 0.030 in. feed and a speed of 174 rpm, whereas the highest torque for this same cutting condition is 50.8 for the borax water (No. 2), or an actual difference of 1.3 ft-lb, corresponding to 2.6 per cent. The torque values for all of the cutting fluids fall between the two lines shown, which is a very narrow range.

The values of thrust for the various cutting conditions also

are shown plotted for all the cutting fluids in Fig. 18. Again the divergence of results for the different cutting fluids is very small. Dry cutting, as opposed to the torque, gives the highest thrust values, whereas the light mineral oil (No. 6) and the light mineral oil containing oleic acid (No. 9) give the lowest values of thrust. The sulphurized lard-mineral oil is next to the lowest, immediately above which are the equal thrust values for cutting fluids Nos. 2, 4, 5, 8, and 10. It seems strange that next to the highest thrust values when drilling this free-cutting brass are obtained with the 50-to-1 soluble oil (No. 3) and the heavy mineral oil (No. 7). These two cutting fluids give the same results in thrust. It also is interesting to note that the heavy mineral oil (No. 7) gives next to the highest in thrust values, whereas the light mineral oil (No. 6) gives the lowest thrust values.

From values of torque and thrust taken from the curves of Fig. 18, and using the torque and thrust equations, average values for the constants C and K have been computed as given in Table 4.

Tests on Cast Iron

Preliminary drilling tests were first run on the cast iron, cutting dry, to develop suitable formulas for torque and thrust as a function of the feed and drill diameter. The variable-feed tests were run with the $1^1/_4$-in.-diam drill, with feeds of 0.011, 0.015, 0.021, and 0.029 in. per revolution. The variable-diameter tests with the $^1/_2$-in., 1-in., and $1^1/_4$-in.-diam drills were made with a constant feed of 0.021 in. per revolution. These tests all were run dry.

The results of these tests are shown plotted in Fig. 19. The exponent of the feed as a function of torque is found to be 0.6, and that as a function of thrust 0.73. The exponent for the drill diameter as a function of torque is found to be 1.7, and that as a function of the thrust was found to be indeterminate. By cut-and-try methods, the thrust values, as a function of drill diameter, were replotted and the web thickness of the drill was taken into consideration. It was found that, when the thrust was plotted over $(d/5 + w/d)$, a straight line was obtained on log-log paper, the slope of which was 1.9. The formulas for torque and thrust were then obtained as follows:

$$T = C f^{0.6} d^{1.7}$$
$$B = K f^{0.73} (d/5 + w/d)^{1.9}$$

The values of w for different drill sizes are given under the column of web thickness of the drill in Table 3.

In a previous paper, "Power Required to Drill Cast Iron and Steel,"[4] a formula was developed for a soft cast iron for torque, as follows:

$$T = C f^{0.6} d^2$$

The present formula gives the exponent of d of only 1.7. This difference apparently is due to the difference in cast irons, as the cast iron used in this paper contained 1 per cent nickel. The exponents of the feed, however, are the same in the two formulas. It appears from this that, for cast iron, no one set of formulas will hold for various analyses.

The results of a series of experiments to determine comparative values illustrating the influence of various cutting fluids on the cast iron are shown plotted in Figs. 20 to 24, inclusive. The averages of experimental values of torque are plotted on log-log paper in Fig. 20 for each of the eleven cutting fluids used. The torque values for the different drill sizes operating at their proper speeds and feeds are plotted over a variable L, in which L equals $f^{0.6} d^{1.7}$, as taken from the torque formula.

The lines in Fig. 20 illustrate that the greatest torque values are obtained when cutting dry (No. 1). It is interesting to note the close values obtained with cutting fluids 4, 2, 5, 7, 3, 6, 11, and 8, inasmuch as No. 2 is a borax water, Nos. 4 and 3 are soluble oils, No. 5 is a lard oil, Nos. 7 and 6, which give the same values as No. 3, are heavy and light mineral oils, respectively, No. 11 is a sulphurized lard-mineral oil, and No. 8 is a mineral-lard oil. The sulphurized mineral oil (No. 10) gives torque values lower than all those mentioned, while the mineral oil containing 5 per cent oleic acid gives the lowest results of all. Values of torque have been taken from the curves of Fig. 20 in order to compute the constant C for torque from the torque formula. The values of these constants for the eleven cutting fluids are shown in Table 4.

The torque for the $1^1/_4$-in.-diam drill operating at 0.015-in. feed and at a speed of 174 rpm is 43.6 for dry cutting (No. 1). This is the highest of all torque values. The torque value for cutting fluid No. 9 for this drill diameter is the lowest and is 35.9 ft-lb. This shows that the torque for cutting cast iron dry is 21.5 per cent higher than when cutting with oil No. 9.

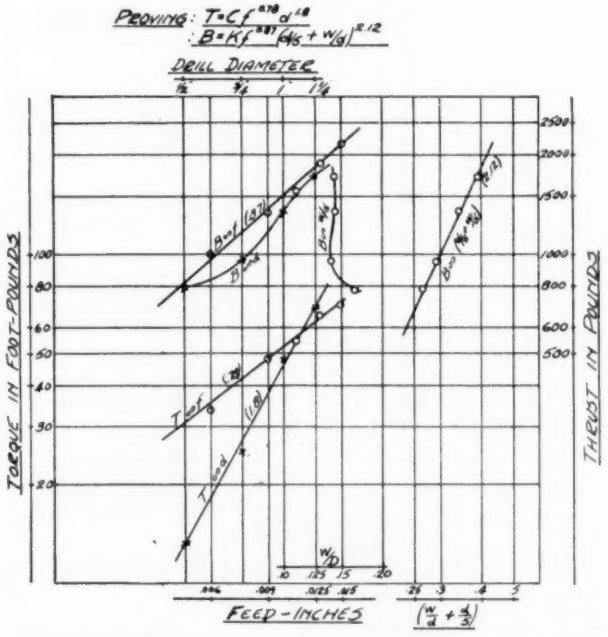


Fig. 29 Torque and Thrust Required to Drill S.A.E. 3150 Steel Using Oil No. 3

(Constant feed, 0.009 in., variable diameter; constant diameter, 1 in., variable feed.)

The average of experimental values of thrust when drilling cast iron also are shown plotted over the variable L in Fig. 21, in which L equals $f^{0.73} (d/5 + w/d)^{1.9}$. Here again straight lines are obtained representing thrust values for the various drill diameters at their respective feeds and speeds. The thrust is greatest when cutting dry (No. 1) and with the heavy mineral oil (No. 7). It is less for the light mineral oil (No. 6) and the mineral-oleic acid oil (No. 9). The conditions seem to be the reverse of what might be expected, inasmuch as the two sulphurized oils (Nos. 10 and 11) give thrust values near the top. Lard oil is in the middle, and the water compound and soluble oils are below the mean values.

The value of the constant K for the thrust formula for each of the cutting fluids, as computed from the curves of Fig. 21, are shown in Table 4.

Values of torque from Fig. 20 are shown plotted for the various drill diameters operating at their respective speeds and feeds over the oil numbers as abscissa in Fig. 22. The oil numbers are arranged in order of increasing torque. The computed

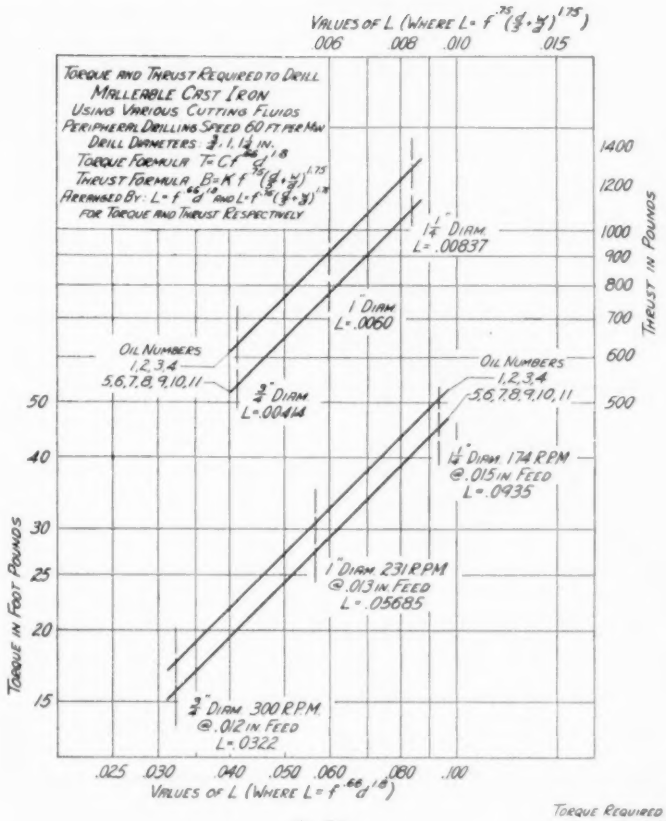


Fig. 26

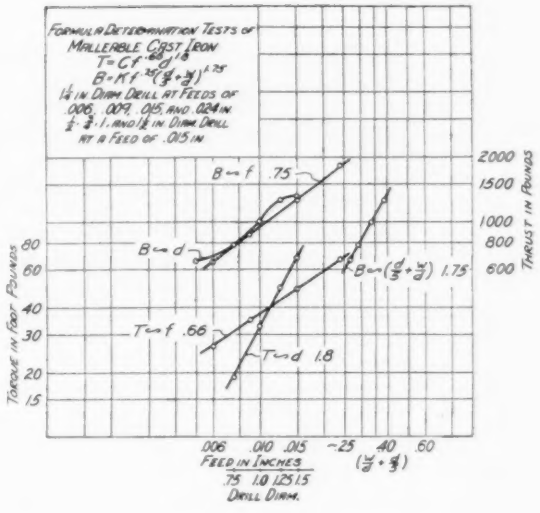


Fig. 25

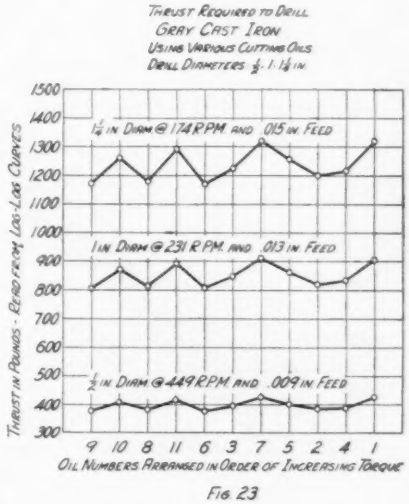


Fig. 23

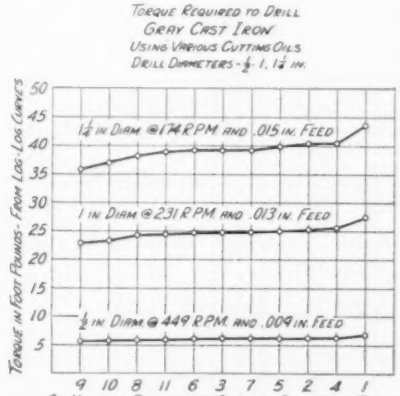


Fig. 22

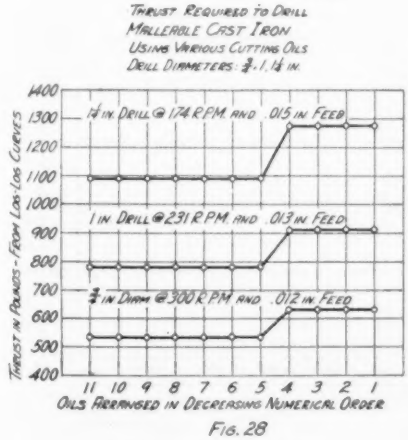


Fig. 28

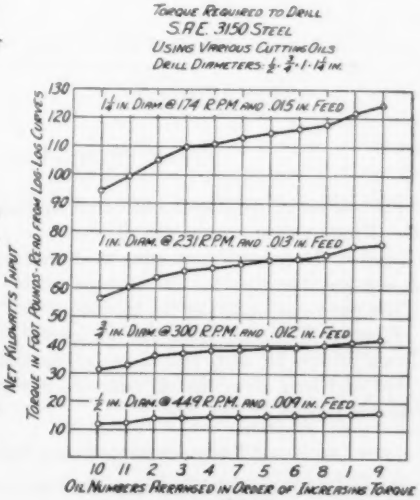


Fig. 32

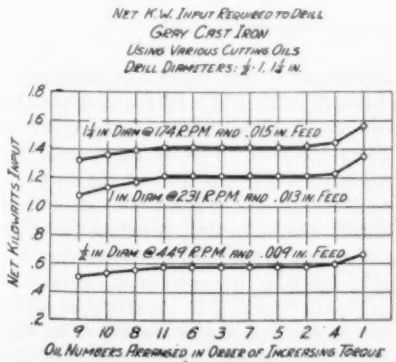


Fig. 24

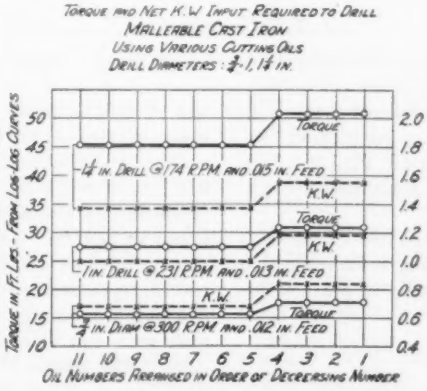


Fig. 27

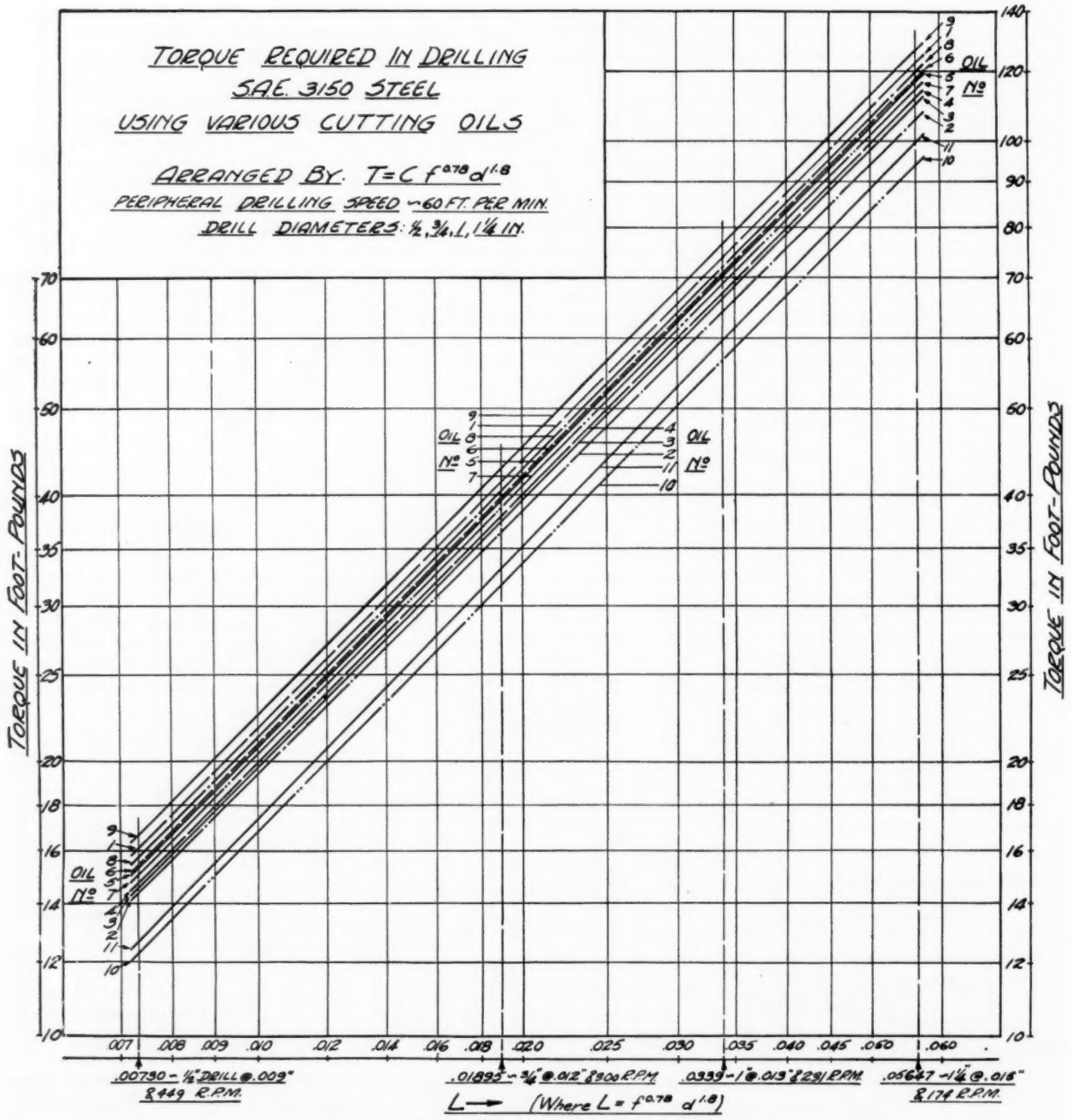


FIG. 30 TORQUE REQUIRED IN DRILLING S.A.E. 3150 STEEL, USING VARIOUS CUTTING OILS

thrust values for drilling the cast iron similarly are shown in Fig. 23. These thrust values are plotted over the oil numbers arranged in the order of increasing torque as used in Fig. 22. It is shown that the thrust for the different cutting fluids does not vary in the same proportion as the torque.

When drilling cast iron, the net power input in kilowatts is shown plotted for each drill diameter, operating at its respective speed and feed, over the oil numbers, arranged in order of increasing torque, in Fig. 24.

TESTS ON MALLEABLE IRON

A special series of drilling tests were run on the malleable-iron castings to determine formulas for torque and thrust as a function of the drill diameter and feed. The results of these tests are shown plotted on log-log paper in Fig. 25. In the variable-feed tests, a $1^1/_4$-in.-diam drill was used with feeds of 0.006, 0.009, 0.015, and 0.024 in. per revolution. The torque values are connected by a straight line, and the exponent of the feed was found to be 0.66. The thrust values for the $1^1/_4$-in.-diam drill at various feeds also indicate a straight line, the slope of which is 0.75, representing the exponent of the feed.

Using a constant feed of 0.015 in. per revolution, a series of tests were run with the $^1/_2$-in., $^3/_4$-in., 1-in., and $1^1/_2$-in.-diam drills. The results of the torque and thrust are shown in Fig. 25. The exponent of the drill diameter as a function of the torque is found to be 1.8. The values of thrust, plotted over

the drill diameter as a variable, do not give a straight line. By trial and error, the web thickness of the drill was introduced, so that a straight line is obtained using the thrust as a function of $(d/5 + w/d)$, the exponent of which is 1.75. The resulting formulas for torque and thrust when drilling malleable cast iron are therefore

$$T = C f^{0.66} d^{1.8}$$
$$B = K f^{0.75} (d/5 + w/d)^{1.75}$$

The results of the tests to determine the torque and thrust of the different drill diameters operating at their respective feeds and speeds, as influenced by the eleven cutting fluids, are plotted in Figs. 26 and 27. The torque values were plotted on log-log paper in Fig. 26 over the variable L, in which L equals $f^{0.66}\ d^{1.8}$, as taken from the torque formula. Only two straight lines, representing torque over the variable L, were obtained. The upper curve represents the torque values for all cutting fluids Nos. 1 to 4, inclusive, while the lower curve represents the torque values of the balance of the cutting fluids, consisting of the oils. It is interesting that malleable iron, whether drilled dry, with borax water, or with a soluble oil, produces the same torque, other conditions being equal. It also is interesting that if any oil, at least any of those from 5 to 11, inclusive, representing a wide range of types, is used, equal torque values are obtained. From these two curves, the constants in the torque formulas have been obtained and are shown listed in Table 4.

The values of thrust for this series of experiments also are shown plotted in Fig. 26 over the variable L, in which L equals $f^{0.75}\ (d/5 + w/d)^{1.75}$, as taken from the thrust formula. Again, two straight lines are obtained, the higher one of which represents the thrust for the various drills for cutting fluids Nos. 1 to 4, inclusive. The lower line represents the values of thrust for the balance of the cutting fluids—namely, the oils Nos. 5 to 11, inclusive.

From Fig. 26 there appears to be an increase in torque of about 13 per cent if the malleable cast iron is drilled dry or with water compounds instead of with any of the oils. An additional 17 or 18 per cent thrust is set up by cutting dry or using the water compound instead of the oils. Thrust coefficients also have been determined for each of the cutting fluids as listed in Table 4.

The values of torque and net power input in kilowatts are plotted for each drill diameter over the cutting oils arranged in numerical order in Fig. 27. The values of thrust for the various drill diameters are shown plotted over the cutting fluids arranged in numerical order in Fig. 28.

Tests on S.A.E. 3150 Steel

The results confirming the equation for torque and thrust in drilling the annealed S.A.E. 3150 steel are shown in Fig. 29. In these tests an emulsion consisting of 1 part of soluble oil to 50 parts water (No. 3) was used. The variable-feed tests were run with a 1-in.-diam "R" drill, whereas the variable-diameter tests were run with a constant feed of 0.009 in. per revolution.

The torque values at the 0.009-in. feed for various drill diameters are shown to be practically a straight line, the slope of which is 1.8. The torque values for various feeds for the 1-in.-diam drill also indicate a rather satisfactory straight line, the slope of which is 0.78. These values indicate the equation for torque

$$T = C f^{0.78} d^{1.8}$$

This equation confirms the equation for steel determined on a wide variety of types of steel and at different times, as reported on in the paper, "Power Required to Drill Cast Iron and Steel,"[4] and reported on further in a paper before the Society of Automotive Engineers, *Journal S.A.E.*, vol. 23, No. 3, p. 78.

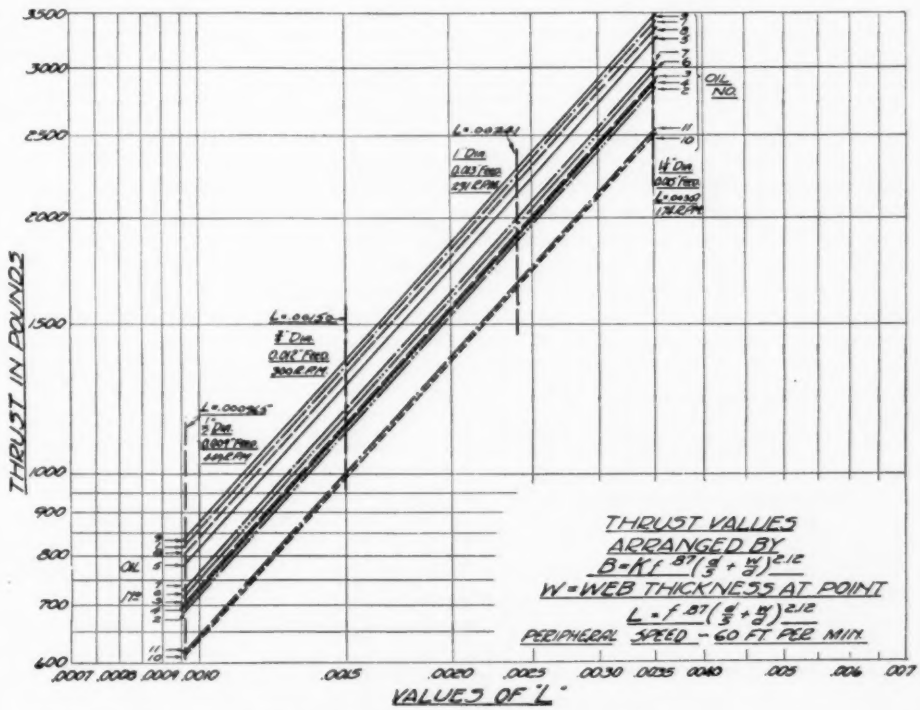


Fig. 31 Thrust for Various Values of L

The values of thrust for the 1-in.-diam drill operating at various feeds are shown as a straight line in Fig. 29, the slope of which is 0.87. The thrust, plotted for a constant feed of 0.009 but for variable drill diameters, appears to be a slightly curved line. A straight line is obtained when the thrust is plotted over the value $(d/5 + w/d)$. The slope of this line is 2.12. This indicates then that an equation for thrust is a function of the feed, $f^{0.87}$, and the value of $(d/5 + w/d)^{2.12}$, resulting in the equation for thrust

$$B = K f^{0.87} (d/5 + w/d)^{2.12}$$

These equations are confirmed when oils Nos. 6, 8, and 10 are used, as reported below Fig. 8 in the Progress Report No. 3.

The results of the tests for torque for the different drills, each drilled with all of the cutting fluids, are shown plotted on log-log paper in Fig. 30. The torque values were plotted over the variable L, where L equals $f^{0.78} d^{1.8}$. Straight lines are obtained. It is of interest to note that the mineral oil containing 5 per cent oleic acid produces a greater torque than when cutting dry (No. 1). The sulphurized oils (Nos. 10 and 11) produce

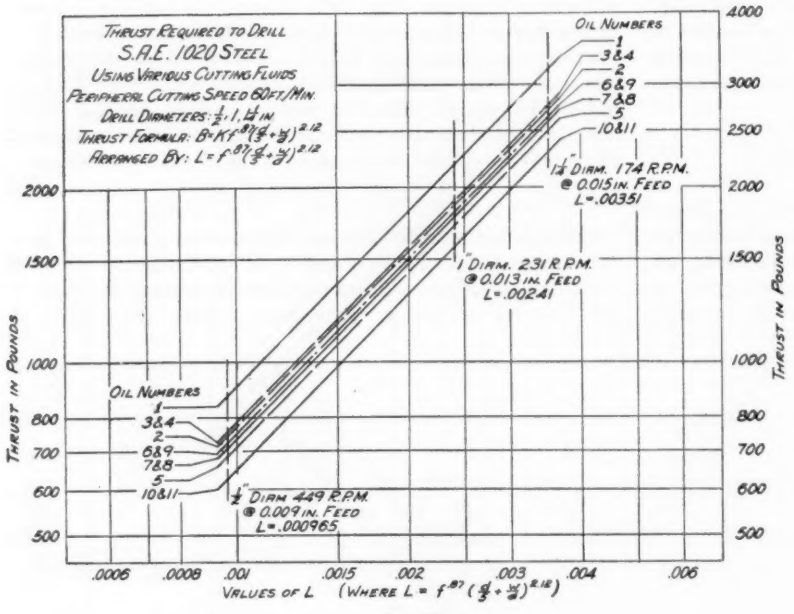


Fig. 36

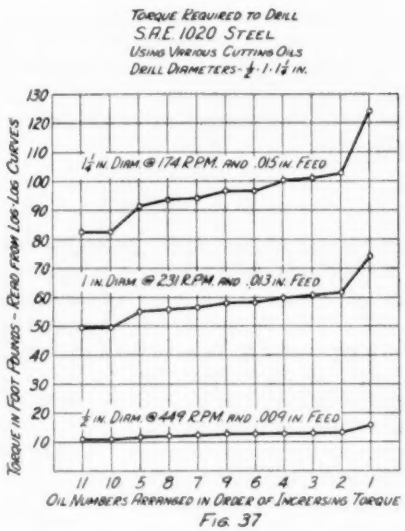


Fig. 37

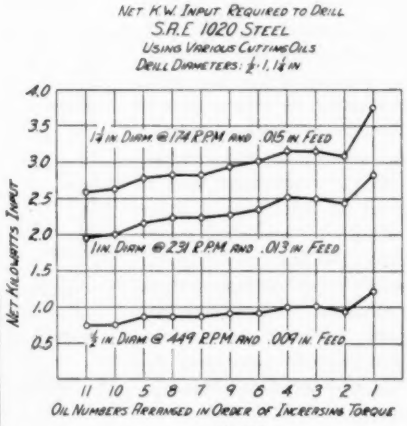


Fig. 39

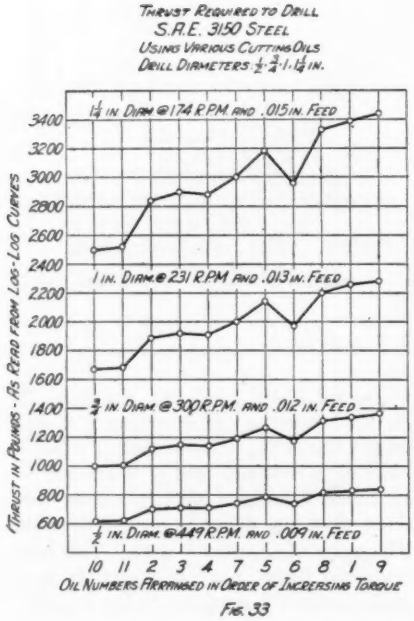


Fig. 33

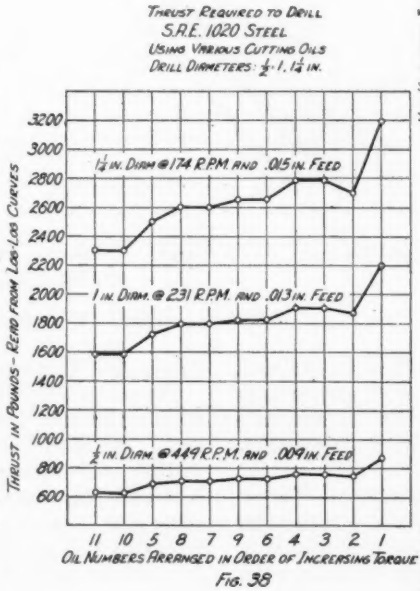


Fig. 38

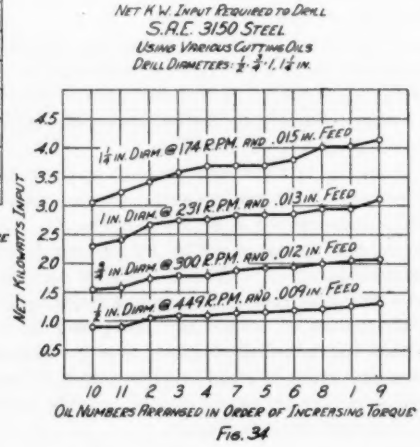


Fig. 34

the lowest values. These curves were discussed in detail in Progress Report No. 3.

The average experimental values of thrust are plotted on log-log paper in Fig. 31, over the variable L. Again, straight lines are obtained, and an order of oils quite similar to that for torque is found when arranged by magnitude.

Constants for both torque and thrust determined from the curves in Figs. 30 and 31 are listed in Table 4.

Values of torque for each drill size are shown plotted over the oil numbers, arranged in order of increasing torque, when cutting the S.A.E. 3150 steel, in Fig. 32. This shows an appreciable influence of the cutting fluid on the torque values. The values of thrust are shown in Fig. 33 plotted over the oil numbers arranged in order of increasing torque. Also, the net power input in kilowatts required to drill the S.A.E. 3150 steel is shown plotted for all four drill diameters, namely, $^1/_2$ in., $^3/_4$ in., 1 in., and $1^1/_4$ in., in Fig. 34. These results, relating to the S.A.E. 3150 steel, were previously reported on in detail in Progress Report No. 3.

Tests on S.A.E. 1020 Steel

Inasmuch as the special series of tests conducted to determine the torque and thrust formulas for the S.A.E. 1020 steel yielded the same formulas as determined for the S.A.E. 3150 steel, these data are not presented. Rather, the formulas for the torque and thrust, as developed for the S.A.E. 3150 steel, will be considered to apply to this steel. In fact, it has been found by many repeated tests that most steels, unless highly alloyed, follow these equations for torque and thrust. The constants, however, are dependent upon the particular steel as described below.

Comparative values of torque were determined for all of the eleven cutting fluids when drilling the S.A.E. 1020 steel, with the $^1/_2$-in., 1-in., and $1^1/_4$-in.-diam drills operating at their respective feeds and speeds. The torque values are shown plotted over the values of L, in which L is equal to $f^{0.78}\,d^{1.8}$ taken from the torque formula in Fig. 35. The values as plotted are the averages of experimental values. The torque values read from the straight lines of Fig. 35 were used in computing the constant C in the torque formula for the various cutting fluids. These are shown in Table 4.

A series of straight lines representing torque are obtained for each of the cutting fluids. The results as shown in Fig. 35 are of interest, inasmuch as dry cutting (No. 1) produces torque values much higher than any of the other cutting fluids. As with the S.A.E. 3150 steel, the sulphurized oils (Nos. 10 and 11) produce unusually low values of torque. It is also interesting to note that lard oil (No. 5) produces a torque lowest of all cutting fluids with the exception of Nos. 10 and 11, while the aqueous solutions (Nos. 2, 3, and 4) produce the highest torque values, except for dry cutting (No. 1).

The averages of experimental values of torque are plotted in Fig. 36 over a variable L, in which L equals $f^{0.87}\,(d/5 + w/d)^{2.12}$. Again, straight lines are obtained in almost identically the same order of magnitude as shown for torque in Fig. 35. Constants K for the thrust formula were computed from thrust values taken from these straight lines in Fig. 36, and are shown for each cutting fluid in Table 4.

The values of torque are shown plotted for each drill diameter over the cutting-fluid numbers arranged in order of increasing values of torque for the S.A.E. 1020 steel in Fig. 37. The increase in torque for dry cutting (No. 1) over sulphurized mineral-lard oil (No. 11) does not appear to be appreciative for the $^1/_2$-in.-diam drill operating at 449 rpm and 0.009-in. feed per revolution. The torque value is 15.96 ft-lb for cutting fluid No. 1 and only 10.62 ft-lb for cutting fluid No. 11. This represents 50 per cent increase in torque when cutting dry as compared to the torque when cutting with cutting fluid No. 11. This increase is more noticeable in connection with the torque values for the $1^1/_4$-in.-diam drill operating at 174 rpm and 0.015-in. feed per revolution.

The values of thrust are shown over the cutting-fluid numbers arranged in order of increasing torque in Fig. 38. It is seen that the thrust for borax water (No. 2) falls below the thrust for the soluble oils (Nos. 3 and 4), but otherwise the thrust values follow closely the order of torque. The increase of thrust for dry cutting (No. 1) over the sulphurized oil (No. 11) is 39 per cent. The net power input expressed in kilowatts required to drill the S.A.E. 1020 steel is shown for each drill diameter plotted over the oil number, arranged in order of increasing torque, in Fig. 39. Except for a slightly reduced value for borax water (No. 2), these curves correspond favorably with the torque curve, as might be expected.

Tests on S.A.E. 1035 Steel

The formulas representing torque and thrust when drilling the S.A.E. 1035 steel have been found to be identical with those formulas developed for the S.A.E. 3150 steel. They are therefore not repeated here.

The average values of experimental data for the torque and thrust are plotted over the variable L, computed from the torque formula in Fig. 40. The cutting fluids are arranged according to the magnitude of torque in practically the same order as when drilling S.A.E. 1020 steel, reported on. Values of the constant in the torque formula have been computed from torque values taken from the curves of Fig. 40 and are shown in Table 4.

The averages of the experimental values of thrust when drilling the S.A.E. 3150 steel with the various oils and drills are shown plotted on log-log paper over the value of L taken from the thrust formula in Fig. 41. The order of the oils, arranged according to magnitude of thrust, does not compare favorably with that when drilling the S.A.E. 1020 steel. Dry cutting (No. 1) gives the highest values. The aqueous solution, consisting of borax water (No. 2), and the soluble oils (Nos. 3 and 4), give next highest values. All of the oils are grouped together, giving the lowest values of thrust. Lard oil gives the lowest value of thrust, while the sulphurized lard-mineral oil (No. 11) and the mineral-lard oil (No. 8) give next lowest values of thrust. Constants K for the thrust formula are shown as computed in Table 4.

The values of torque as developed by each of the drill sizes are shown plotted in Fig. 42 over the cutting-fluid numbers, arranged in order of increasing torque. Again, the sulphurized oil produces the lowest torque, while dry cutting produces the highest torque value. The values of thrust for these drills are plotted over the cutting-fluid numbers arranged in order of increasing torque in Fig. 43. This shows some irregularities between the values of thrust and values of torque. The 1-to-10 soluble oil (No. 4), for instance, produces a thrust value slightly greater than the value for the 1-to-50 soluble oil (No. 3). The thrust values of the plain mineral oils (Nos. 6 and 7) are slightly lower than that for the mineral oil containing oleic acid (No. 9). The thrust value for lard oil is lowest of all. The net power input in kilowatts, as determined from the wattmeter readings, is shown plotted in Fig. 44 over the cutting fluid numbers arranged in order of increasing torque. Again, as might be expected, these lines correspond favorably to the torque lines in Fig. 42.

Tests on Carbon Tool Steel

The formulas for torque and thrust for the annealed carbon tool steel are identical with those developed for the S.A.E. 3150 steel, so that the results of the formula tests are not repeated.

The averages of experimental values of torque are shown plotted in Fig. 45 for each drill diameter when cutting with the various cutting fluids over the value of L determined from the torque formula. The line representing the values of torque when

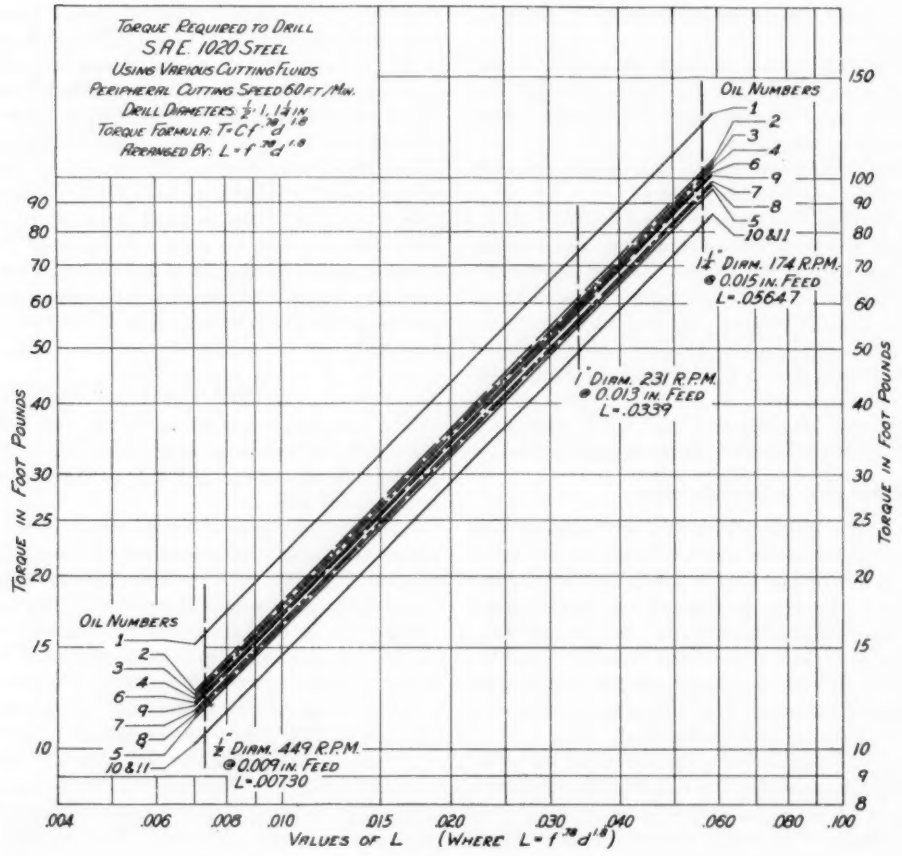

TORQUE REQUIRED TO DRILL
S.A.E. 1020 STEEL
USING VARIOUS CUTTING FLUIDS
PERIPHERAL CUTTING SPEED 60 FT./MIN.
DRILL DIAMETERS: ½-1, 1¼ IN.
TORQUE FORMULA: T=C f^.78 d^1.8
REARRANGED BY: L = f^.78 d^1.8
OIL NUMBERS
1
2
3
4
6
9
7
8
5
10 & 11
1¼" DIAM. 174 R.P.M.
@ 0.015 IN. FEED
L=.05647
1" DIAM. 231 R.P.M.
@ 0.013 IN. FEED
L=.0339
½" DIAM. 449 R.P.M.
@ 0.009 IN. FEED
L=.00730
TORQUE IN FOOT POUNDS
VALUES OF L (WHERE L = f^.78 d^1.8)

FIG. 35

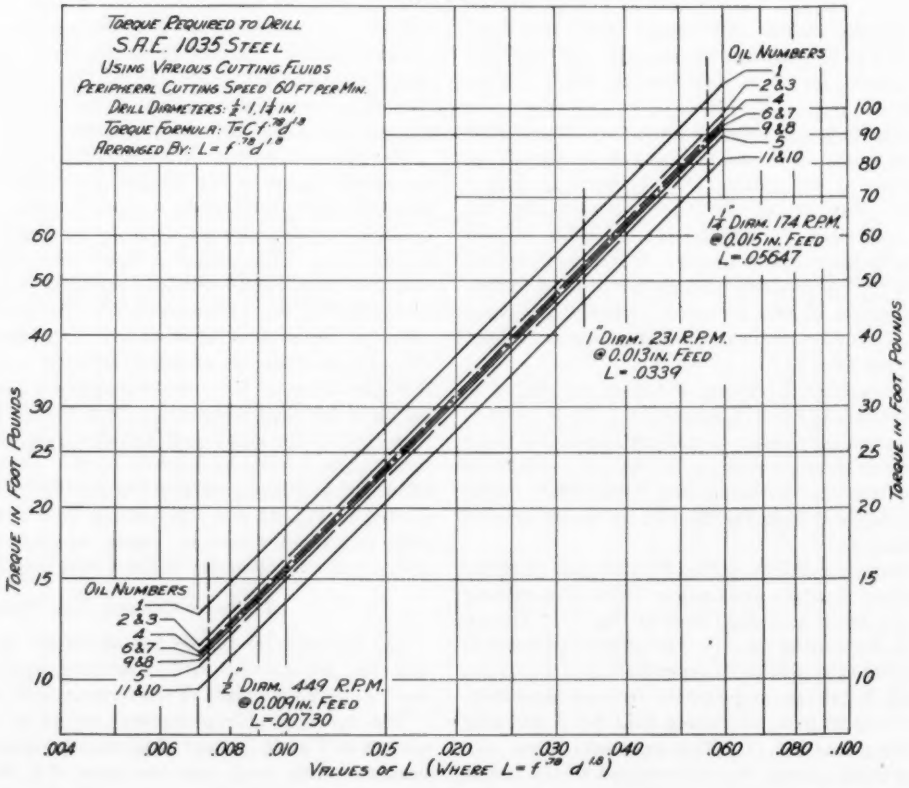

TORQUE REQUIRED TO DRILL
S.A.E. 1035 STEEL
USING VARIOUS CUTTING FLUIDS
PERIPHERAL CUTTING SPEED 60 FT PER MIN.
DRILL DIAMETERS: ½-1, 1¼ IN.
TORQUE FORMULA: T=C f^.78 d^1.8
REARRANGED BY: L = f^.78 d^1.8
OIL NUMBERS
1
2 & 3
4
6 & 7
9 & 8
5
11 & 10
1¼" DIAM. 174 R.P.M.
@ 0.015 IN. FEED
L=.05647
1" DIAM. 231 R.P.M.
@ 0.013 IN. FEED
L=.0339
½" DIAM. 449 R.P.M.
@ 0.009 IN. FEED
L=.00730
TORQUE IN FOOT POUNDS
VALUES OF L (WHERE L = f^.78 d^1.8)

FIG. 40

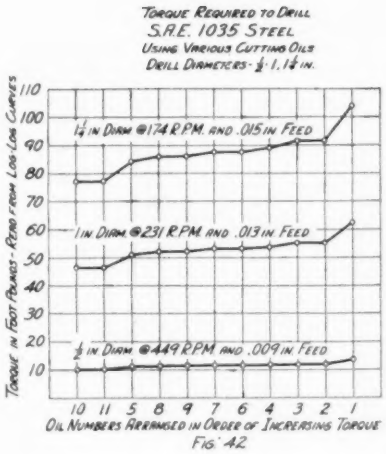


Fig. 42

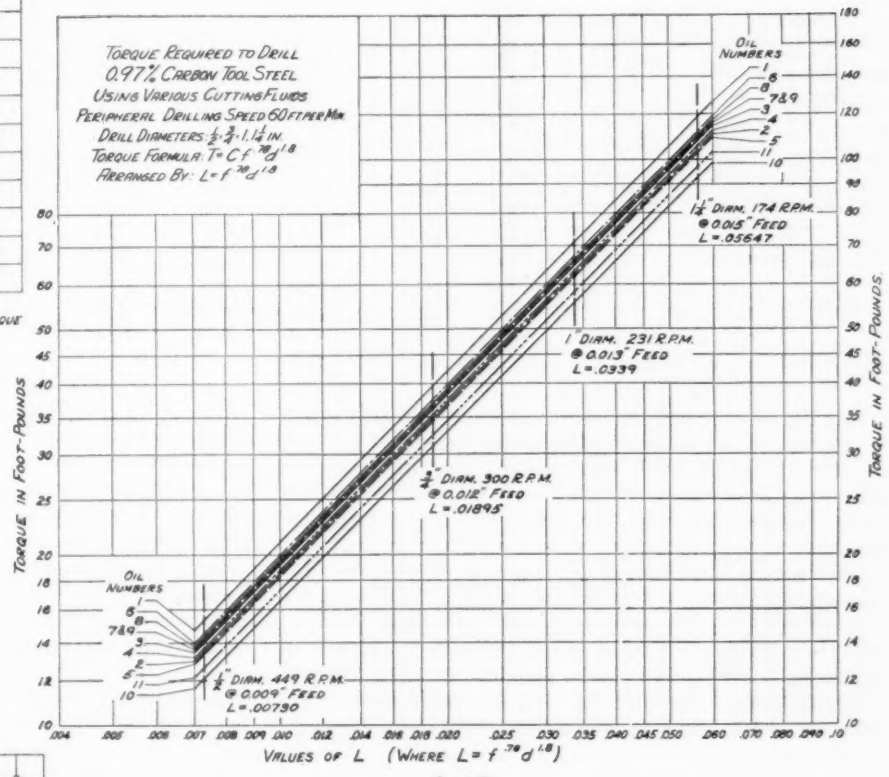


Fig. 45

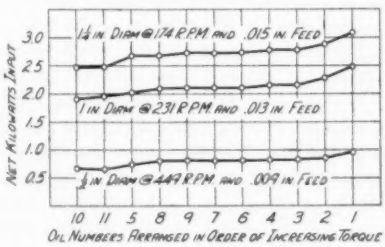


Fig. 44

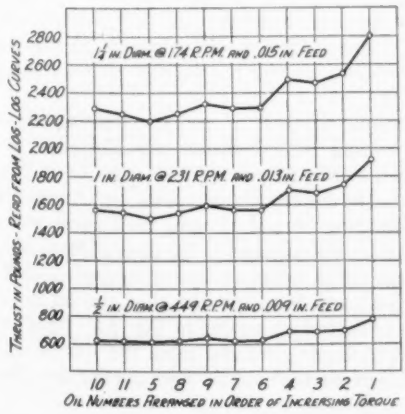


Fig. 43

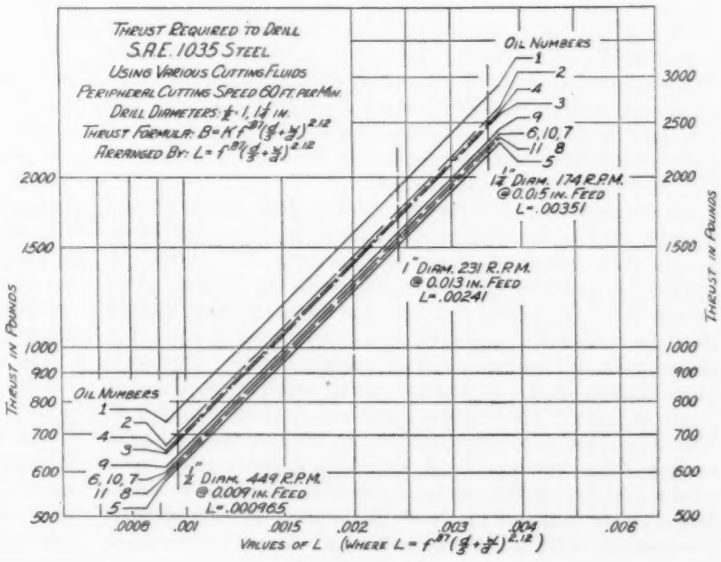


Fig. 41

TABLE 5 COMPARISON OF DRILLING TORQUE, THRUST, AND HORSEPOWER OUTPUT FOR THE SEVERAL METALS
(The values are computed from the formulas for dry cutting (No. 1) and for the best cutting fluid.)

Material	Oil No.	1/2"-diam drill[a] Torque	1/2"-diam drill[a] Thrust	1 1/4"-diam drill[b] Torque	1 1/4"-diam drill[b] Thrust	Hp output computed from torque and thrust: 1/2"-diam drill (a)	Hp output computed from torque and thrust: 1 1/4"-diam drill (b)	Output hp per cu in. per min: 1/2"-diam drill (a)	Output hp per cu in. per min: 1 1/4"-diam drill (b)	Percentage saved, best cutting fluid over dry cutting (No. 1)
Aluminum alloy, AF.....	1.....	3.480	204.0	30.2	1018	0.30	1.01	0.378	0.314	31.7
	8.....	2.383	120.5	20.7	601	0.205	0.69	0.258	0.215	
Free-cutting brass, BF.....	1.....	3.590	206.0	29.8	698	0.31	1.00	0.39	0.301	0.0
	9.....	3.590	190.5	29.8	645	0.31	1.00	0.39	0.301	
Cast iron, CF.....	1.....	6.75	421.0	43.6	1310	0.58	1 45	0.73	0.452	17.2
	9.....	5 59	375.0	35.9	1170	0.48	1.20	0.605	0.374	
Malleable cast iron, DF...	1–4.....	6.7	428.0	50.8	1275	0.58	1.69	0.73	0.527	10.5
	5–11.....	5.98	365.0	45.3	1090	0.52	1.51	0.655	0.470	
S.A.E. 1020 steel, EF.....	1.....	15.96	875.0	123.5	3180	1.38	4.12	1.74	1.28	33.3
	10–11.....	10.62	628.0	82.4	2285	0.915	2.75	1.15	0.859	
S.A.E. 1035 steel, FF.....	1.....	13.45	769.0	104.0	2790	1.16	3.47	1.46	1.08	26.0
	11.....	9.95	618.0	77.0	2250	0.86	2.57	1.08	0.80	
Carbon tool steel, GF.....	1	15.25	897.0	118.0	3280	1.31	3.94	1.65	1.23	21.0
	10.....	12.03	732.0	93.2	2665	1.04	3.11	1.31	0.97	
S.A.E. 3150 steel, KF.....	9.....	16.72	895.0	129.2	3260	1.44	4.32	1.81	1.35	23.0
	1.....	16.17	880.0	125.0	3200	1.39	4.17	1.75	1.30	
	10.....	12.38	648.0	95.5	2362	1.07	3.19	1.35	0.995	
S.A.E. 1112 steel, MF.....	2.....	9.53	592.0	74.0	2150	0.82	2.47	1.03	0.768	6.8
	1.....	8.90	603.0	69.0	2190	0.77	2.30	0.97	0.716	
	11.....	8.30	592.0	64.4	2150	0.715	2.15	0.90	0.670	

[a] At 0.009-in. feed and 449 rpm; [b] at 0.015-in. feed and 174 rpm.

cutting dry (No. 1) and that when cutting with a sulphurized mineral oil (No. 10) is lowest. The water compounds (Nos. 2, 3, and 4) are placed together about midway between the highest and lowest values, whereas the oils Nos. 6, 7, 8, and 9 are slightly above the water compounds, and lard oil (No. 5) is immediately below.

Constants for the torque formula have been computed from values taken from the curves of Fig. 45, as shown for each cutting fluid in Table 4. The averages of the experimental values of thrust have been plotted in Fig. 46 over the value of L taken from the thrust formula. The results obtained are quite different from any already presented for alloy or carbon steels. The thrust values obtained with the sulphurized oils (Nos. 10 and 11) are extremely low and in a class almost by themselves. The rest of the cutting fluids are grouped, the maximum differences of thrust for the various oils being relatively small. The thrust values when cutting dry (No. 1) appear centrally located in the group. The thrust values produced by the water compounds are shown below those for dry cutting (No. 1), as are the thrust values for the heavy mineral oil (No. 7) and the mineral oil containing oleic acid (No. 9). Strangely, thrust values produced by the light mineral oil (No. 6), lard oil (No. 5), and mineral lard oil (No. 8) lie slightly above those for dry cutting (No. 1). The constants K for the thrust formula have been computed from values taken from these curves in Fig. 46 and are shown for each of the cutting fluids in Table 4.

The values of torque for the different drill diameters are plotted over the cutting-fluid numbers, arranged in order of increasing torque, when drilling the carbon tool steel, Fig. 47. There is an appreciable saving in torque in drilling with the sulphurized oil over drilling dry (No. 1). The values of thrust are shown plotted over the cutting-fluid numbers, arranged in order of increasing thrust, when drilling the carbon tool steel, in Fig. 48. The favorable influence of the sulphurized oils in reducing the thrust is apparent. The highest thrust values are obtained with lard oil (No. 5) and the mineral-lard oil (No. 8), whereas the rest of the cutting fluids produce thrust values of approximately equal values. In Fig. 49 the net input power in kilowatts, as determined by the wattmeter, is shown plotted over the cutting fluids arranged in order of increasing torque. These curves again follow quite favorably the torque curves of Fig. 47.

Tests on S.A.E. 1112 Steel

It has been found by tests that the formulas developed for the S.A.E. 3150 steel also hold for the free-cutting steel corresponding to the analysis of S.A.E. 1112.

The averages of experimental values of torque are shown plotted on log-log paper over the value of L in Fig. 50. The results indicate that the torque when drilling a free-cutting steel is affected but little by changes in the characteristics of the cutting fluids. The two sulphurized oils (Nos. 10 and 11) give the lowest torque values. The borax water (No. 2) gives the highest torque values. Equal torque values are obtained for the rest of the cutting fluids, consisting of Nos. 1, 3, 4, 5, 6, 7, 8, and 9. The differences in torque values for this last-mentioned group were so small that they are considered equal. This shows that the torque value for cutting dry (No. 1) is not changed if cutting fluids of lard oil, straight mineral oil, or soluble oil are introduced. These results are due probably to the cutting characteristics of the high-sulphur steel. Constants C for the torque formula have been computed for each of the cutting fluids as shown in Table 4.

The averages of experimental values of thrust were plotted on log-log paper in Fig. 51 over the value of L as computed from the thrust formula. The differences in thrust values for the various cutting fluids are very small. It is of interest to note that the thrust values are lowest when the 1-to-50 soluble oil (No. 3) is used and the next lowest when the 1-to-10 soluble oil (No. 4) is used. The highest values of thrust are obtained when cutting dry (No. 1) and with the heavy mineral oil (No. 7), which gave equal values. The balance of the cutting fluids give intermediate and equal values. It is of interest to note that the borax water (No. 2) gives the same values of thrust when cutting this free-cutting steel as lard oil (No. 5), the sulphurized oils (Nos. 10 and 11), the mineral and mineral-lard oils (Nos. 7 and 8). This order of thrust does not compare favorably with the order of torque shown in Fig. 50. The values of the constants in the thrust formula have been determined for the various cutting fluids and are listed in Table 4.

The values of torque required to drill the free-cutting steel, S.A.E. 1112, for various drill diameters are plotted over the various cutting-fluid numbers, arranged in order of increasing torque in Fig. 52. The lowest torque values are obtained with the sulphurized oils (Nos. 10 and 11) and the highest with the borax water (No. 2). The values for the intermediate oils are practically identical. The values of thrust required to drill the free-cutting steel also are shown plotted over the oil numbers arranged in order of increasing torque in Fig. 53. This brings out the low values of thrust produced when the 1-to-50 soluble oil (No. 3) is used, and the high values when cutting dry (No. 1) and with the heavy mineral oil (No. 7).

The net power input in kilowatts, as determined by the recording wattmeter, required to drill the S.A.E. 1112 steel is also

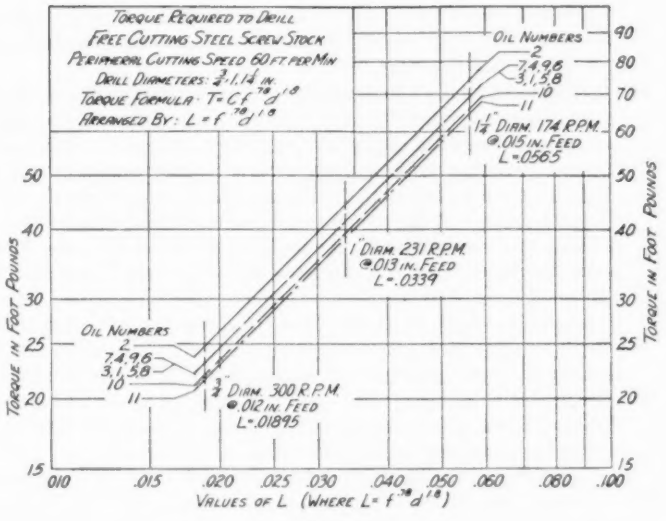


FIG. 50

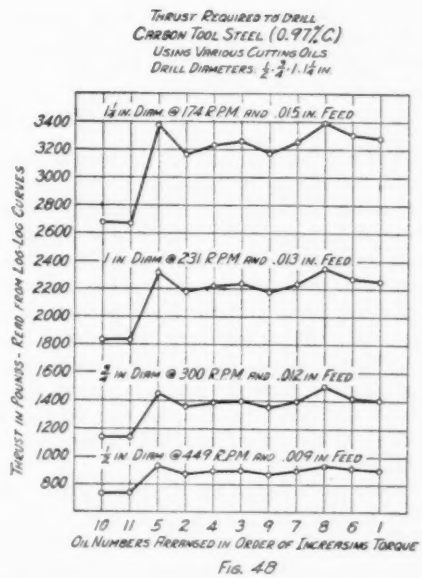


FIG. 48

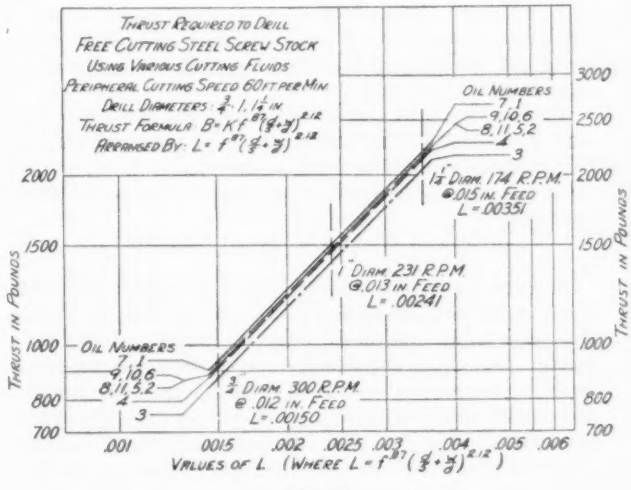


FIG. 51

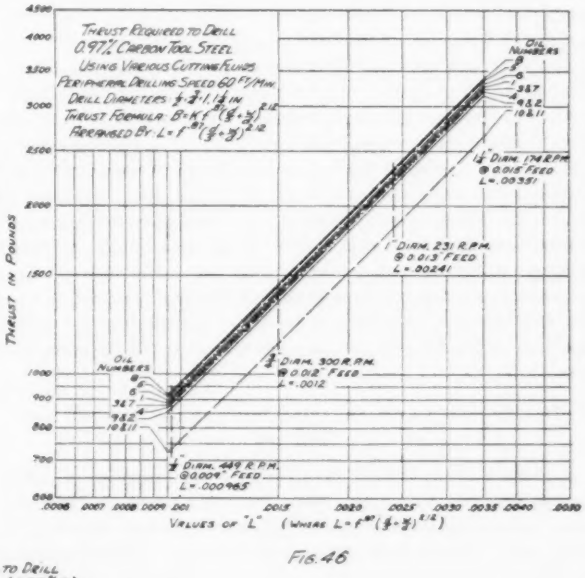


FIG. 46

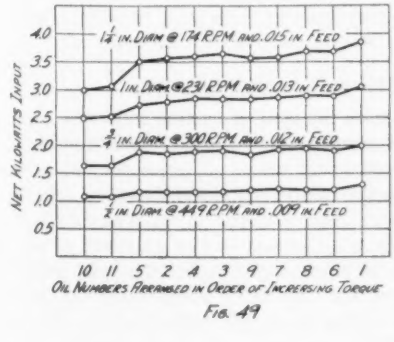


FIG. 49

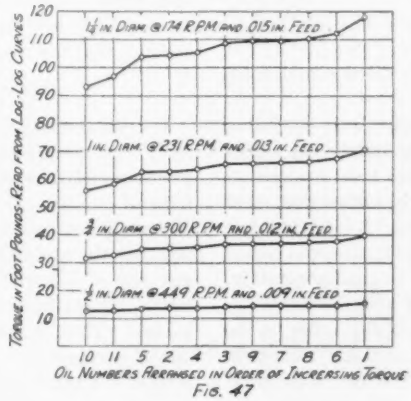


FIG. 47

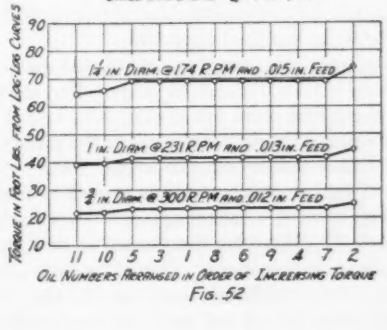


FIG. 52

plotted over the cutting-fluid numbers arranged in order of increasing torque in Fig. 54. These curves show the same general tendency as the torque curves in Fig. 52.

DISCUSSION OF DRILLING TESTS

In Table 4 are summarized the torque and thrust formulas for each of the nine metals tested. Values of the constants C and K in the torque and thrust formulas, respectively, are listed. Computed values of torque and thrust for the $^1/_2$-in.-diam drill operating at 0.009-in. feed per revolution and 449 rpm and for the $1^1/_4$-in.-diam drill operating at 0.015-in. feed per revolution and 174 rpm are given in Table 5. These torque and thrust values are given for the oil numbers appearing in the column to the left, the first of which gives the highest torque values and the second of which gives the lowest torque values. In general, the values of the thrust for the different cutting fluids follow the magnitude of the torque, although there are a few exceptions to this rule.

In every case, except for the free-cutting steel, bar MF, and the S.A.E. 3150 steel, bar KF, dry cutting (cutting fluid No. 1) gives the highest torque value. In the case of these two steels, the torque values for dry cutting have been included along with the maximum and minimum values to make the table more complete. The horsepower output as computed from the torque and thrust formulas, or that power actually delivered at the point of the drill, is shown for both the $^1/_2$-in.-diam and the $1^1/_4$-in.-diam drill. Also, values of horsepower output per cubic inch of metal cut per minute are tabulated for the various materials when drilled by the several cutting fluids. The cutting speed in rpm does not appear in the torque and thrust values as presented, although the speeds at which these values were obtained are recorded. These speeds, however, are used in determining the horsepower output as well as the horsepower output per cubic inch per minute. The power output per cubic inch of metal cut per minute computed for each drill size at its respective speed and feed furnish, therefore, a basis for comparing the drilling qualities of the various metals when cut with the different coolants.

Development of Drilling Formulas for Torque and Thrust

A routine adopted after developing a large number of drilling torque and thrust formulas when cutting a variety of materials with many standard and special types of drills is as follows: The drills were selected alike in respect to angles. They ranged in size from $^1/_2$ in. in diameter to $1^1/_2$ in. in diameter. In some instances the specimen of metal tested was not large enough to permit the use of the $1^1/_2$-in.-diam drill. The drills were very carefully sharpened on a Blau drill grinder and were checked to see that both lips or cutting edges were alike. One lip did not protrude beyond the other more than 0.0005 in. for the small drills or 0.001 in. for the larger drills.

Several holes were then drilled with each drill to take the initial keenness from the cutting edge. The torque and thrust values of these first holes are usually low, and it has been found that the smaller the drill, the greater the number of inches that must be drilled before uniform values result. The $^1/_2$-in.-diam drill was then run at a feed heavy enough to insure torque and thrust values of such a magnitude that they could be read accurately. All of the drills were then tested at that feed and the same peripheral cutting speed. One of the larger drills, depending on the size of the metal being tested, usually the 1-in. or the $1^1/_4$-in.-diam drill, was selected and tests run employing a number of different feeds. Three holes were drilled with each drill diameter and feed combination. If for any reason the results when tabulated did not check, a fourth or fifth hole was drilled. Average values were then plotted on log-log paper, as has been described, to determine the formulas.

For most metals and especially the free-cutting varieties, very favorable experimental results were obtained. For some metals, such as those of a sticky or non-uniform character, additional tests were often necessary, inasmuch as average results from the first test did not seem to fall on straight lines when plotted on log-log paper. It has been found by repeated tests that the same values of the exponent for the feed and depth are obtained with any combination of drill diameter and feed range; that is, the slope of a variable-feed line on log-log paper run with a $^1/_2$-in.-diam drill is the same as that for a line run with a 1-in.-diam drill with variable feed. The two lines are merely displaced vertically, one being above or below the other. This same condition maintains when the various drill diameters are used with a light feed or a heavy feed as a constant, or with any cutting fluid except for aluminum, as has been noted.

However, if the angles of the drills used in these formula tests are changed, different formulas result. One metal will respond more favorably to an increased helix (or rake) angle of the drill than another. The drills used to develop the formulas presented in this report were what are believed to be a representative standard type. It has been found in some instances that two drills, apparently identical, will give values consistently different by several per cent. This difference obviously is due to some other feature of the drill.

The formulas presented in this report will agree very closely with those obtained by standard drills of other manufacturers. The formulas presented were all determined using the same set of drills, so that they are directly comparative for the different materials.

The Formulas

In reviewing the formulas for torque and thrust presented in Table 4, several interesting facts are immediately apparent. The same torque and thrust formulas are obtained for all of the steels. The constants, however, vary between the different steels, as well as for the different cutting fluids. For aluminum, free-cutting brass, cast iron, and malleable iron the formulas show considerable variation. The web thickness seems to have about the same effect on thrust values for cast iron and malleable iron as for steel. It does not have nearly as much effect, however, when cutting aluminum and brass, and for this reason the simpler thrust formulas, omitting the factor of web thickness, are presented as giving satisfactory accuracy. When comparing the steels cutting dry or with any of the cutting fluids, a direct comparison may be made between the constants, as the formulas and exponents are alike. Such comparisons, however, cannot be made between the non-steels by comparing constants, due to the fact that the exponents of feed and drill diameter vary.

Influence of Cutting Fluids

It is interesting to note the type of cutting fluid which gives the lowest torque and power values for the different metals cut. For the steels it is always a sulphurized oil, either the sulphurized mineral oil (No. 10) or the sulphurized lard-mineral oil (No. 11). For malleable cast iron it does not seem to make any difference whether the material is cut with any one of a variety of oils or any one of a variety of water compounds. The oils give the lower values, however, and the water compounds the higher values. That is, all cutting fluids for malleable iron fall into one of two groups. When cutting cast iron and the non-ferrous metals, a compounded oil, such as mineral-lard oil (No. 8) or the mineral-oleic acid (No. 9), seems to give the lowest values. Better to indicate the possible power saving over dry cutting by

Curves Showing Values of $(\frac{d}{3}+\frac{d}{4})$ Raised to Various Powers
For Use in the Drilling Thrust Formulae

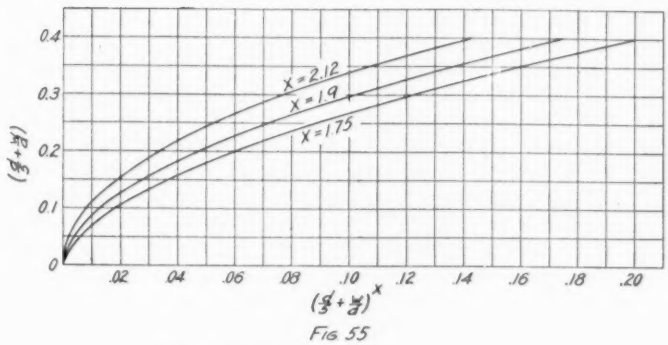


Fig. 55

Thrust Required to Drill
Free Cutting Steel (S.A.E. 1112)
Using Various Cutting Oils
Drill Diameters $\frac{3}{4}$, 1, $1\frac{1}{4}$ in

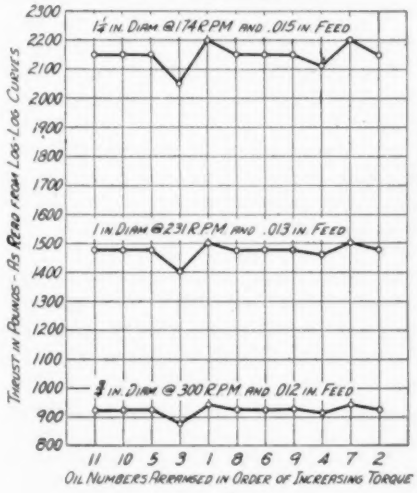


Fig. 53

Net K.W. Input Required to Drill
Free Cutting Steel (S.A.E. 1112)
Using Various Cutting Oils
Drill Diameters: $\frac{3}{4}$, 1, $1\frac{1}{4}$ in

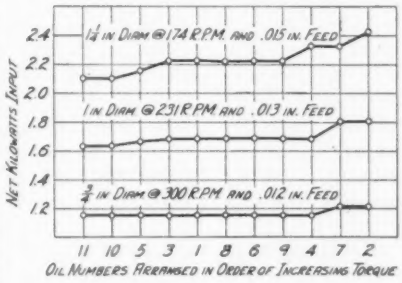


Fig. 54

Comparison of Thrust Required to Drill
Several Ferrous and Non Ferrous Metals
Using Various Cutting Fluids
1¼ in Diam. Drill @ .015 in. Feed
Computed Values

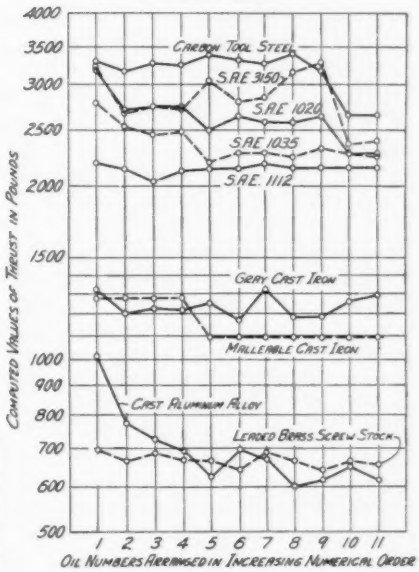


Fig. 57

Comparison of Torque Required to Drill
Several Ferrous and Non Ferrous Metals
Using Various Cutting Fluids
1¼ in Diam. Drill @ .015 in Feed
Computed Values

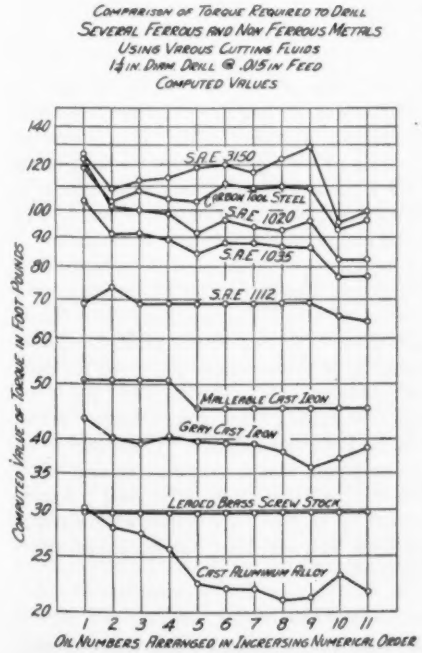


Fig. 56

Comparison of Torque Required to Drill
Several Ferrous and Non Ferrous Metals
Cutting Dry and with Oils No. 9 and No. 10
1¼ in Diam. Drill @ .015 in Feed
Computed Values

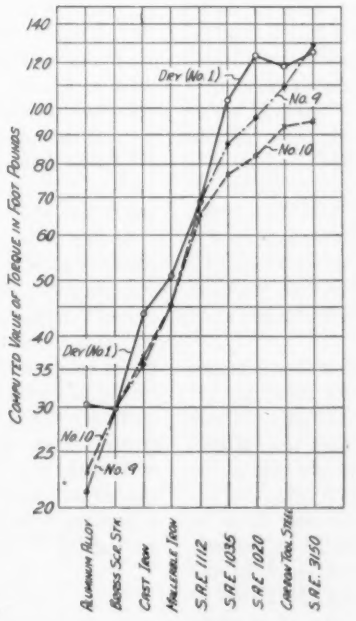


Fig. 58

Comparison of Thrust Required to Drill
Several Ferrous and Non Ferrous Metals
Cutting Dry and with Oils No. 9 and No. 10
1¼ in Diam. Drill @ .015 in Feed
Computed Values

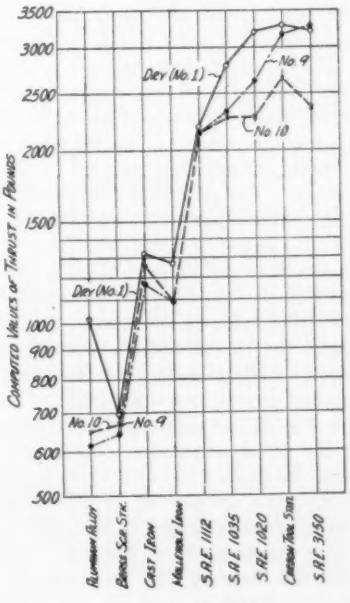


Fig. 59

using the best oil for each metal, a column has been added to Table 5 showing the percentage saved, based on the values for dry cutting. The greatest saving of 33$^1/_3$ per cent results when drilling the S.A.E. 1020 steel, bar EF, with a sulphurized oil, either Nos. 10 or 11. The sulphurized oils produce a saving of 26 per cent, 21 per cent, and 23 per cent over dry cutting when cutting the S.A.E. 1035, carbon tool steel, and S.A.E. 3150 steel, respectively. It is interesting, however, that when cutting the free-cutting steel, S.A.E. 1112, only 6.8 per cent is saved through the use of the sulphurized lard-mineral oil (No. 11). The influence of the cutting fluids on free-cutting brass is seen to be negligible. The torque values are the same for all drill sizes, and there is only a slight reduction in thrust for the mineral oil containing 5 per cent oleic acid (No. 9).

It appears that as long as an oil is used, a saving of about 10 per cent in power results when drilling malleable cast iron. Except for a possible increase in tool life, there is no advantage in using a water compound as against dry cutting. A somewhat greater saving of 17.2 per cent is made over dry cutting when cast iron is drilled with the mineral oil containing 5 per cent oleic acid (No. 9). A saving of 31.7 per cent is made when cutting the cast aluminum over dry cutting by using the mineral-lard oil (No. 8). The straight or compounded oils as a group produce the lowest torque values when drilling aluminum. The water compounds, in the form of straight soluble oils or soluble oils containing kerosene, as well as pure kerosene, give torque values considerably above those of the oil group. Kerosene, therefore, does not appear to advantage when cutting this type of aluminum, particularly when power is the chief consideration.

The values of torque for the 1$^1/_4$-in.-diam drill operating at a feed of 0.015 in. per revolution and at a speed of 174 rpm (equivalent to 60 fpm) are shown plotted in Fig. 56 over the cutting-fluid numbers, arranged in numerical order, for all nine materials tested. A direct comparison is possible between the drilling properties of these metals with one another and shows the influence of the cutting fluids on each metal. The vertical scale is logarithmic, so that equal percentages are represented by equal spaces. The four uppermost curves, giving the torque for S.A.E. 3150 steel, carbon tool steel, S.A.E. 1020, and S.A.E. 1035 steel, are shown to be quite similar. The curve fifth from the top for S.A.E. 1112 steel seems to be in a class by itself, different somewhat from the other four steels. The torque as a function of the cutting fluids for each of the other metals—namely, malleable cast iron, gray cast iron, leaded brass, and cast aluminum alloy—is quite independent, and no satisfactory correlation seems possible. The results of torque as presented for these various metals seem to justify the many existing conflicting opinions as to the advantages or disadvantages of various cutting fluids. It appears that definitely to specify any cutting fluid for a given material complete information must be had as to the influence of all cutting fluids on that metal. Obviously it would be only by chance that a satisfactory solution of a cutting-fluid problem could be made when the selection of a cutting fluid is based upon so-called general practical results.

In Fig. 57 are shown plotted the corresponding thrust values for the 1$^1/_4$-in.-diam drill operating at a feed of 0.015 in. per revolution and a speed of 174 rpm (equivalent to 60 fpm). These thrust values are plotted for all nine materials over the cutting-fluid numbers, arranged in numerical order as in the case of the torque values given in Fig. 56. Again, lack of complete coordination between the performances of the cutting fluids as affecting thrust is seen for the different metals. In general, each curve corresponds in nature of slope to its respective torque curve of Fig. 56. The thrust values for the S.A.E. 1112 steel of Fig. 57 appear to be in closer agreement with the other steels above than is the case of the torque values for the S.A.E. 1112 steel in Fig. 56. Furthermore, the thrust values of malleable cast iron appear to be lower in the general scale than are the torque values in Fig. 56. In Fig. 56 all torque values for malleable iron are consistently (approximately 15 per cent) higher than the torque values for gray cast iron. In Fig. 57 the thrust values for malleable cast iron for oils Nos. 5 to 11, inclusive, are below the corresponding values for cast iron by about 9 per cent. It is also seen that the thrust values for the leaded brass are slightly lower than the torque values of Fig. 56, while the thrust values for aluminum, Fig. 57, are slightly higher than the torque values for aluminum in Fig. 56. Inasmuch as the torque values of Fig. 56 for each metal and the thrust values of Fig. 57 for each metal were computed from the formulas developed for identical cutting conditions—that is, with the same feed per revolution and speed in rpm—the torque values of Fig. 56 for any condition are directly comparable with any other torque value shown. This holds true also for thrust values given in Fig. 57.

In order to show clearly the influence of what might be considered the maximum torque values for each metal as produced by dry cutting (No. 1), the minimum torque values for each metal as produced by the sulphurized lard oil (No. 10), as well as the torque values for each metal determined when the mineral oil containing 5 per cent oleic acid (No. 9) was used, the torque values produced by each of the three oils are shown plotted over each material in Fig. 58. The materials are arranged from left to right in order of increasing torque. These three cutting fluids were selected inasmuch as they represent extreme cutting conditions for the different metals. Fig. 58 is prepared to show at a glance the influence of the cutting fluids on each metal. It is seen, for instance, that when cutting aluminum, dry cutting (No. 1) gives the greatest torque value, 30.2 ft-lb, and the mineral oil containing 5 per cent oleic acid (No. 9) gives the least torque value, 21 ft-lb. The difference is about 9.2 ft-lb. This 9.2 ft-lb represents a saving of 31.7 per cent due to oil No. 9 over dry cutting, as shown in Table 4, or it shows an increase in torque when cutting dry of 44.3 per cent over the torque value for oil No. 9. Also, it is seen from Fig. 58 that, when drilling the free-cutting brass screw stock, the torque values for the three oils, Nos. 1, 9, and 10, are identical. Perhaps the greatest range of torque for the different oils is for the S.A.E. 1020 steel. A torque of 123.5 ft-lb is produced cutting the S.A.E. 1020 steel dry (cutting fluid No. 1), while a torque of only 82.4 ft-lb is produced with oil No. 10. This represents a saving for oil No. 10 of 41.4 ft-lb. In other words, cutting dry requires 50 per cent more torque than when cutting with oil No. 10. It is of interest to note the relative position of the torque values for each metal produced when oil No. 9 is used. When cutting aluminum with oil No. 9, the lowest torque value is obtained, but when cutting the S.A.E. 3150 steel, the highest value of torque is obtained.

In Fig. 59 are shown the values of thrust plotted to logarithmic ordinate scale over the various metals for the 1$^1/_4$-in.-diam drill operating at a feed of 0.015 in. and 174 rpm. Again, the thrust values for cutting fluids Nos. 1, 9, and 10 are plotted. These curves indicate the diversity in thrust values produced for each metal between different cutting fluids. The diversity produced by the oils when cutting free-cutting brass is very small. When cutting S.A.E. 1112 steel, it is quite negligible. It appears to be greatest when cutting aluminum, and is very appreciable when cutting the S.A.E. 1020 steel and the S.A.E. 3150 steel. Again, the thrust values produced by oil No. 9 are lowest when cutting aluminum, free-cutting brass, and cast iron, but maximum when cutting the S.A.E. 3150 steel.

CONCLUSIONS

Conclusions as to the influence of cutting fluids when cutting the nine materials listed in the paper are first presented for each

of the several materials separately. After this more general conclusions are added.

Drilling Aluminum

1 The torque and thrust values resulting from the drilling of aluminum vary a great deal, depending on the type of cutting fluid used. The torque and thrust values are greatest when drilling the cast aluminum alloy dry (cutting fluid No. 1).

2 The water compounds consisting of cutting fluids Nos. 2, 3, and 4, as well as pure kerosene and 1 part kerosene in 4 parts cutting fluid No. 3, produce much the same results of torque and thrust when drilling the cast aluminum alloy. The torque values produced by these water compounds and kerosene are appreciably higher than the values produced by any of the oils.

3 There appears to be little difference in torque or thrust values when drilling with any oil, whether straight mineral oils, lard oil, compounded oils, or sulphurized oils.

4 Sulphur does not appear to be desirable in a cutting fluid for this cast aluminum alloy. Compounded oils, such as Nos. 8, 9, and 11, seem to produce the lowest values of torque and thrust in cutting this aluminum. The compounded oils are better than the straight mineral oils or the straight lard oils.

5 It appears that the formula for thrust as developed when cutting dry (No. 1) holds for all of the oils plus the kerosene and kerosene emulsion, but does not hold for the water compounds Nos. 2, 3, and 4. This is the first time that a cutting fluid has been noticed to exert such an influence on the drilling formula developed for any material.

6 A saving of 31.7 per cent power at the drill point is made if a mineral oil containing 10 per cent lard is used when drilling this aluminum instead of dry cutting.

7 The formulas for torque and thrust when drilling cast aluminum alloy, together with constants for various cutting fluids, are summarized in Table 4.

Drilling Free-Cutting Brass

1 All of the cutting fluids tested give almost identical torque values when drilling the free-cutting brass.

2 But little difference in values of thrust was found for the various cutting fluids used.

3 A light mineral oil gave the lowest thrust value and a heavy mineral oil gave the highest thrust value, except for dry cutting, indicating that when drilling free-cutting brass with mineral oil, thrust increases with viscosity.

4 There is no saving in power output at the drill point through the use of cutting fluids.

5 The drilling torque and thrust formulas, together with constants for free-cutting brass, are summarized in Table 4.

Drilling Cast Iron

1 It was found that cutting fluids have considerable influence on the torque when drilling cast iron, but less influence on the thrust. Almost identical values of torque and thrust were obtained for a number of the cutting fluids.

2 The lowest torque when drilling cast iron was obtained with the mineral oil containing 5 per cent oleic acid (No. 9); the next lowest torque was obtained with the sulphurized mineral oil (No. 10).

3 The highest torque when drilling cast iron was obtained when drilling dry.

4 A saving of 17.2 per cent of the power output at the drill point is made if the oil No. 9 is used instead of dry cutting.

5 The sulphurized oils (Nos. 10 and 11) give values of thrust among the highest, while the water compounds give values slightly below the average.

6 The formulas for torque and thrust, together with the constants for various cutting fluids when drilling cast iron, are summarized in Table 4.

Drilling Malleable Cast Iron

1 Cutting fluids exert a marked influence on both torque and thrust when drilling malleable cast iron. The cutting fluids fall into two groups: The first group, consisting of dry cutting and the water compounds (cutting fluids Nos. 1 to 4, inclusive), gives high and equal torque and thrust values, while the second group, consisting of all of the oils, gives low and equal torque and thrust values.

2 There is a distinct saving in power for thrust in drilling malleable cast iron with an oil. The saving in power amounts to 10.5 per cent as compared to the power required to drill dry.

3 The torque and thrust formulas for malleable cast iron, together with constants for various cutting fluids, are summarized in Table 4.

Drilling S.A.E. 3150 Steel

1 Cutting fluids exert a considerable influence on the torque and thrust values when drilling the chrome-nickel steel, S.A.E. 3150. The highest torque value is obtained when the mineral oil containing 5 per cent oleic acid is used, with dry cutting (No. 1) a close second.

2 The sulphurized oils (Nos. 10 and 11) give by far the lowest values of torque and thrust.

3 The water compounds (Nos. 2, 3, and 4) give torque and thrust values below those for all of the oils, Nos. 10 and 11 excepted.

4 A saving in power at the drill point of 23 per cent may be made by drilling the S.A.E. 3150 steel with sulphurized mineral oil (No. 10) over dry cutting.

5 The formulas for torque and thrust for the S.A.E. 3150 steel, together with the constants for each cutting fluid, are summarized in Table 4.

Drilling S.A.E. 1020 Steel

1 The various cutting fluids materially influence the torque and thrust values when drilling the S.A.E. 1020 steel. All cutting fluids give torque and thrust values considerably below those for dry cutting.

2 As was the case with the S.A.E. 3150 steel, the sulphurized oils (Nos. 10 and 11) produce unusually low values of torque and thrust.

3 The aqueous solutions (Nos. 2, 3, and 4) give the highest values of torque and thrust of all of the liquids.

4 The cutting fluids for reducing torque and thrust are arranged in almost identical order.

5 Approximately 33 per cent saving in power output at the drill point is effected by drilling the S.A.E. 1020 steel with the sulphurized oils (Nos. 10 and 11) rather than with dry cutting.

6 The formulas for torque and thrust, together with the constants for each of the cutting fluids when drilling the S.A.E. 1020 steel, are summarized in Table 4.

Drilling S.A.E. 1035 Steel

1 The various cutting fluids exert a marked influence on the torque and thrust values when drilling the S.A.E. 1035 steel.

2 The torque curves, plotted over the cutting-fluid numbers arranged in numerical order, of the S.A.E. 1020 and 1035 steels are practically identical, the torque line for the S.A.E. 1020 steel being approximately 10 per cent higher. The thrust lines for the S.A.E. 1035 and S.A.E. 1020 steels are in close agreement, except for the values of thrust produced by the sulphurized oils (Nos. 10 and 11). The sulphurized oils do not seem to produce such low values of thrust when drilling the S.A.E. 1035 steel as

might be expected from the corresponding curve for the S.A.E. 1020 steel.

3 For the S.A.E. 1035 steel the lowest value of thrust is obtained when lard oil (No. 5) is used. The next lowest values of thrust are obtained with the sulphurized oils (Nos. 10 and 11), with the thrust for mineral lard oil (No. 8) almost equal in value.

4 Power savings at the drill point of 26 per cent can be effected by the introduction of the sulphurized mineral oil (No. 11) as compared with dry cutting.

5 Torque and thrust formulas when drilling the S.A.E. 1035 steel, together with constants, are shown in Table 4.

Drilling Carbon Tool Steel

1 The torque and thrust values vary considerably as different cutting fluids are introduced when drilling this carbon tool steel.

2 The sulphurized oils (Nos. 10 and 11) produce the lowest values of torque and thrust.

3 The highest torque values in drilling the carbon tool steel are obtained when cutting dry (cutting fluid No. 1).

4 The water compounds (Nos. 2, 3, and 4) and lard oil (No. 5) give torque values averaging 6 per cent below the torque values for the straight mineral oils and compounded oils.

5 The torque values for the carbon tool steel are about 5 to 10 per cent below those for the S.A.E. 3150 steel, whereas the thrust values for the carbon tool steel are, with the exception of the value for oil No. 9, 8 to 10 per cent above those for S.A.E. 3150 steel.

6 About 21 per cent saving in net power output at the drill point can be made by drilling the high-carbon tool steel with the sulphurized mineral oil as compared with dry cutting.

7 Torque and thrust formulas when drilling carbon tool steel, together with constants for the various cutting fluids, are summarized in Table 4.

Drilling S.A.E. 1112 Steel

1 Cutting fluids seem to have little effect on torque and thrust values when drilling this free-cutting steel.

2 The lowest torque values are obtained with the sulphurized oils (Nos. 10 and 11), whereas the highest torque value is obtained with the use of the borax water (No. 2).

3 The torque for dry cutting seems to be about equal to that when cutting fluids Nos. 3 to 9, inclusive, are used.

4 The lowest value of thrust when drilling the S.A.E. 1112 steel is obtained for the 1-to-50 soluble oil (No. 3). The values of thrust for all other cutting fluids are practically identical.

5 The maximum saving in net power output at the drill point by using sulphurized mineral oil (No. 11) is 6.8 per cent as compared with that for dry cutting.

6 Torque and thrust formulas when drilling S.A.E. 1112 steel, together with constants for the various cutting fluids, are summarized in Table 4.

General Conclusions

1 In the derivation of torque and thrust formulas for drilling various metals, the values of the exponents of the variables are not changed by using different values of the constant. That is, if the exponent of feed as the variable is determined for the constant drill diameter of $^1/_2$ in., the same feed exponent would be obtained if a 1-in.-diam drill were used. Similarly, in determining the exponent of drill diameter as the variable for a constant feed, if a light feed is used, the same exponent would be obtained as with a heavy feed. The lines drawn through the torque values as a function of variable feed or diameter on log-log paper are always straight. Thrust lines are straight only for aluminum and brass. For the other metals the web thickness must be considered to get satisfactory straight lines.

2 The values of the exponents of feed and diameter in the torque and thrust formulas do not change for different cutting fluids. Aluminum S.A.E. 33 is the only exception as far as the data presented in this paper show. The constant of the equation alone is affected in most cases by the different cutting fluids.

3 One torque and thrust formula has been found to hold for all steels tested in this paper. Each of the other metals, however, such as aluminum, free-cutting brass, cast iron, and malleable cast iron, has its own formula.

4 There appears to be no general single conclusion that can indicate the best cutting fluid for any given material. The torque lines for the various carbon steels, when plotted over the oil numbers, appear in general to be similar in shape but above or below one another. The torque line, Fig. 56, for the S.A.E. 3150 steel agrees, in a general way, with the tendency of the lines for the straight carbon steels. The compounded oils (Nos. 8 and 9), however, give torque values disproportionately high for the S.A.E. 3150 steel.

5 There is little similarity between the torque curves when plotted over cutting-fluid numbers for malleable iron, gray cast iron, leaded brass, cast aluminum alloy, and the steels. This indicates that the results of a series of cutting-fluid tests on one material would be of little value in predetermining good practice in the use of cutting fluids on another material.

6 The torque and thrust values for free-cutting steel and brass are affected but little by the various cutting fluids.

7 Theoretically the value of thrust contributes less than 1 per cent to the power developed at the drill point. It adds more to the input power due to the added friction for higher thrust values.

8 The sulphurized oils give torque values lower than any other cutting fluid on all of the steels tested. The sulphurized oils do not appear to show an advantage over the other oils, such as the lard, mineral, and compounded, when cutting malleable cast iron, gray cast iron, and free-cutting brass. When cutting aluminum, the sulphurized mineral oil appears to disadvantage when compared with the other oils. (See Fig. 56.) The various water compounds and oils do not appear to reduce the power when drilling the free-cutting steel and the free-cutting brass.

Acknowledgments

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The authors wish to express their appreciation to Mr. Charles E. Kraus, who conducted the tests and worked up the data, and to Mr. Theodore R. Thoren, who prepared the drawings and assisted in the experimental work.

The National Twist Drill and Tool Company, of Detroit, furnished the drills gratis. The various cutting oils were furnished gratis by the Standard Oil Company of Indiana, Chicago; the Sun Oil Company, Philadelphia; the Atlantic Refining Company, Philadelphia; Swift & Co., Chicago; and D. A. Stuart & Co., Chicago. The malleable cast iron was furnished gratis by the Cadillac Malleable Foundry.

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Discussion

ROBERT COREY DEALE.[5] The writer was impressed by the amount of experimental work necessary to secure the data here recorded, but cannot feel that these results are of any great value to the industry as a whole, because of the manner in which the experiments have been carried out. No information is given as to the tool life of a given drill point, cutting a given grade of material at a given speed and feed, and with a specific type of cutting fluid. Furthermore, no data are given to show any relationship between torque and tool life.

F. W. Taylor, in "The Art of Cutting Metals," gives considerable attention to the method of conducting cutting tests and emphasizes that the important fact in the testing of cutting tools is the tool life secured when cutting under specific conditions of speed, feed, etc. His statements, while based primarily upon his very extensive experiments with roughing tools in lathes and boring mills, were made after he had retired from the installation of scientific methods in industrial plants, and it is believed that he intended them to apply to all types of metal cutting. If no relationship could be discovered between cutting speed and tool pressure in the case of a simple turning tool, it is still more difficult to discover such a relation in the case of more specialized forms of tools, such as drills and milling cutters.

The production man must know the speeds and feeds to be used under a given set of conditions under which he will get the lowest cost of removing metal, after taking into consideration all factors, including tool cost and grinding time. For him to make direct use of cutting data, they must be in the form of curves or a table giving cutting speeds for given conditions of feed, depth of cut, metal being cut, shape of tool, size of tool, material of tool, cutting fluid used, quantity of cutting fluid, under known conditions of tool life. Preferably he should be provided with data showing the variation of tool life with cutting speed.

The manufacturer of machine tools must know the feed pressure and power required for the cut for which he expects to design the machine, but these values are generally secured by a single set of tests on a late type of his own machines, although formulas giving this information with the necessary degree of accuracy have been available for at least 20 years. When the writer was assistant chief engineer of the Pond plant of the Niles-Bement-Pond Company, he always found it desirable to make such a test before the final design of a new machine in order to check any figures previously available. The manufacturer of cutting tools must have the same information as that required by the production man, although he must investigate the effect of changes in design and material, while the shop is ordinarily limited to the tools commercially available.

[5] Consulting Management Engineer, Babylon, N. Y. Mem. A.S.M.E.

For these reasons, it is believed that tests such as are described in this paper are of little practical value. While it is realized that tests based on tool life are considerably more expensive to make than tests based on torque or tool pressure, in the present state of knowledge it is believed that it is necessary to conduct such tests in such a way that cutting speeds and tool life will be secured directly, rather than to depend on unproved inferences, as is necessary when conclusions are reached from tool pressure data. Furthermore, it is highly desirable to secure data which may be used directly by industry.

The writer believes that the standardization of tests for metal-cutting tools which he proposed in the October, 1932, issue of *Mechanical Engineering*, might well be extended to include other types of tools than the turning tools described. He has never seen data on the life of drills under varying conditions, but sees no reason why all drilling tests, whether intended to show possible speeds and feeds, the comparative value of various designs of drills, or the comparative value of various cutting fluids should be based on a definite tool life, either expressed in minutes of cutting time or inches of hole drilled. It would probably be necessary to further standardize the depth of test hole and determine whether the hole was to be allowed to break through or not. Probably this depth of hole should vary with the size of drill, because of the different lengths of the drills.

It would be further desirable to determine the equation expressing the variation of drill life with the variation of peripheral speed, so that accelerated tests might be made, and then reduced to some standard tool life.

Determination of an equation accurately expressing the variation of drill life with drill diameter, peripheral speed, and feed would be very valuable so that the speeds and feeds giving most economical operation might be readily determined for any machine.

When making drilling tests, it would be desirable to calibrate the machine used in the tests, so that both power input and power at the tool might be recorded. Such data would give all necessary information for the design of the driving mechanism of drilling machines, and in addition would allow production men to readily determine whether definite speeds and feeds were within the capacity of a given machine.

No data have been given showing any relation between tool pressures and cutting speeds for a definite life of the tool between grinds. Until such a relation has been well authenticated, it would seem that any experiments based on tool pressures are of doubtful value, particularly as they give no information as to proper speeds and feeds. Such experiments as those reported in this paper should be so conducted as to show the relation between cutting speed and tool life for a given set of conditions.

ALVAN DAVIS.[6] The following remarks refer to only two metals: leaded free-cutting brass and S.A.E. 1112 free-cutting steel. In the case of free-cutting brass, it is the opinion of some of our machine-shop foremen that there is but little gain in using a cutting fluid so long as the cutting speeds are of the general order of those used in the interesting tests covered by Progress Report No. 4. Thus there is some practical opinion in support of the findings reported. However, in commercial production, our speeds for automatic machines running on free-cutting brass are about five times as fast as in the tests (1430 rpm for a $^{3}/_{8}$-in. drill with a feed of 0.01 in.). At this speed, we use a copious stream of light paraffin oil as a coolant. Otherwise, the tools will burn up.

[6] Research Engineer, Scovill Manufacturing Company, Waterbury, Conn. Mem. A.S.M.E.

In the case of free-cutting steel, we do not use S.A.E. 1112, because the modern super-free-cutting steel is so much better,[7] permitting nearly 50 per cent higher speed. With such steel, our cutting speeds are about two-thirds those for brass, and the energy consumed is about the same. Were we to use S.A.E. 1112, our speeds would be reduced to about 46 per cent of those for brass. This, again, gives about equal rate of energy delivery through the nose of the tool. This is a check on the relative horsepower per cubic inch per minute given in column 10 of Table 5 of the report. That is, 0.301 hp for brass and 0.670 hp for 1112 steel show a ratio of 45$^1/_2$ per cent.

HANS ERNST[8] AND MARIO MARTELLOTTI.[9] The selection of the proper kind of cutting fluid in relation to the cutting tool and the material being cut has always been a matter of conjecture, owing to a lack of information on the performance of cutting fluids. In presenting this report the authors have greatly extended the field covered by their earlier paper, Trans. A.S.M.E., MSP-54-2. It is to be hoped that they will extend this work to cover other types of cutting tools.

The most significant feature developed is the consistent superiority shown by the sulphurized oils Nos. 10 and 11 when drilling steel. This fully corroborates the results of the tests previously reported on S.A.E. 3150 steel. Of the two sulphurized oils, the lower cost oil No. 10 is actually better than the higher cost oil No. 11 for all the steels except the free-cutting steel S.A.E. 1112; yet in the latter case the benefit obtained by the use of any cutting fluid is relatively small owing to the free-cutting nature of this material.

The superiority shown by the sulphurized oils in this test is in agreement with our own experience, where such oils have given remarkably good results in certain difficult machining operations. In one case involving a threading operation on a low-carbon-steel pipe, the cutting fluid which had been used was a straight lard oil, and difficulty was encountered in obtaining a smooth thread due to the tearing of the material. After several unsuccessful attempts to improve the conditions by changing the angles of the chasing tools, it was decided to try various cutting fluids. The best results, with a complete elimination of tearing, was obtained with a sulphurized paraffin-base oil of the composition of 1 part of 9 per cent sulphurized mineral oil to 5 parts of paraffin oil.

In another case a heavy face-milling operation on S.A.E. 1050 steel was materially improved by the use of the same sulphurized oil. In this instance the sulphurized oil gave a much better and smoother finish, and in addition caused an appreciable reduction in power consumption.

The superiority of sulphurized oil in the cutting of steel is so marked as to warrant a thorough investigation as to its cause; thus any information which the authors may have obtained or theories which they have developed on this subject would be of universal interest. Our own observations lead us to believe that the effectiveness of a given cutting fluid is a function both of its lubricating qualities and of its surface tension with respect to the material being cut, at the temperatures reached when the fluid is interposed between tool and chip. The lower the surface tension, the more rapidly a drop of the fluid will spread over the surface of the work and the more readily the surface will be wetted.

In the case of sulphurized oils it is probable that the finely divided particles of sulphur, which are carried in suspension in the oil, will be trapped between the face of the tool and the under surface of the chip. Under the high temperature existing at this point it is possible that the sulphur will actually become fluid, and thus act directly as a lubricant between the flowing chip and the tool. The existence of a high temperature at the nose of a cutting tool is clearly indicated by the smoke evolved when using any type of oil as the cutting fluid. With sulphurized oil, in fact, the quantity of smoke evolved is a fair indication of the degree of dullness of the tool.

It is probable that the addition of sulphur to a mineral oil (which already possesses good lubricating characteristics and low surface tension as compared with water) will provide a further reduction of surface tension, particularly at the temperature of operation. The reduction in surface tension by increase of temperature in oils of this type is very great indeed, and in view of the apparent importance of this property we would like to see it included in the list of the properties of the cutting fluids given in Table 1. For proper comparison, of course, it is necessary to determine the surface tension with respect to the work material, at the temperature encountered in normal operation.

It is noteworthy that the torque formulas show that the exponent affecting the drill diameter is practically independent of the material being cut, while this is not the case with respect to the exponent of the feed. On the other hand, for the various steels tested the exponents of both the feed and the diameter remain constant. This further emphasizes the importance of the subject of chip flow and indicates the need for further investigation of the mechanism of chip removal. In materials where the mechanism is similar, such as the various steels, the exponents are constant, while for dissimilar materials, where the mechanism of chip flow is radically different, the exponents (particularly for the feed) vary with each material.

In the thrust formula for all materials except the aluminum alloy and the leaded brass, the thickness of the web is introduced. The ratio of web thickness to drill diameter is not the same for the different drills used, but varies as follows:

$^1/_2$-in.-diam drill $W/d = 0.162$
$^3/_4$-in.-diam drill $W/d = 0.138$
1 in.-diam drill $W/d = 0.141$
1$^1/_4$-in.-diam drill $W/d = 0.144$
1$^1/_2$-in.-diam drill $W/d = 0.141$

It would be of interest to determine whether the thrust formula could be expressed in the simple form $B = Kf^a d^b$ if the ratio of web thickness to diameter of drill could be maintained constant. Values of w/d for $^3/_4$, 1, and 1$^1/_4$ in. drills are not far different, and in Fig. 29 it will be seen that the thrust values for these three drills may be very nearly represented by the equation $B = K f^{0.87} d^{1.11}$. Thus it would seem probable that a similar relation would hold true for any diameter of drill if the ratio w/d were kept constant.

Further progress reports of the Sub-committee on Cutting Fluids will be awaited with interest.

W. H. OLDACRE.[10] It is almost impossible to consider the machinability of metal without considering lubricants, as certain metals which it is difficult if not impossible to machine with improper lubricants become easily machinable when properly lubricated. The report of the committee clearly demonstrates this fact and should result in an aroused interest and further investigation of the effect of lubricant on machinability.

In the November issue of *Machinery*, C. W. Bettcher, secretary of the Eastern Machine Screw Corporation, of New Haven, gives an interesting report of an investigation of the power con-

[7] Super-free-cutting steel contains 0.20 to 0.30 per cent S, average 0.25 per cent, as against 0.075 to 0.150 per cent S in S.A.E. 1112, average 0.115 per cent.

[8] Research Director, Cincinnati Milling Machine Company. Mem. A.S.M.E.

[9] Engineer, Research Department, Cincinnati Milling Machine Company. Mem. A.S.M.E.

[10] Director of Engineering and Research, D. A. Stuart & Co., Chicago, Ill.

sumed by self-opening die heads. An interesting check on the report of the committee might be secured in a test on the machine described.

Further, some interesting work has been done by various research organizations, and several machines have been developed for testing the load-carrying or anti-scuffing values of lubricants. Among these are the Mougey and Almen machines, developed in the General Motors Research Laboratories, and the Timken lubricant tester, developed by the Timken Roller Bearing Company. Unquestionably, the resistance to metal scuffing, which is one of the characteristics measured by these machines, plays an important part in the lubrication of cutting tools, and an investigation of various cutting lubricants with these machines and an attempt to coordinate the results with practical experience in the machining of metals should prove of definite interest.

GEORGE M. CUNNINGHAM.[11] Much present-day discussion revolves around various sulphurized animal and vegetable oils and their blends with mineral oils for use in the metal-working industry. Yet the several methods of sulphur determination in organic combinations are at variance, and the results of analysis are often misleading. This paper is presented in the belief that the Carius method gives the most consistent results in sulphur analysis with the least necessary equipment and with the hope that an attempt will be made by the industry to standardize on its method of sulphur analysis, so that producer and consumer of sulphurized oils may speak the same language when talking sulphur content. This standardization of sulphur analysis will become increasingly more important if it is later proved that increased efficiency of a sulphurized cutting fluid is directly proportional to its sulphur content.

The Nopco Laboratories had occasion to analyze samples of sulphurized oils, to determine the percentage of sulphur in each sample. Two methods of analysis were used in these determinations:

1 The Eschka method, described in "Standard Methods of Chemical Analysis," by W. W. Scott, 4th edition, Vol. 1, p. 495. This method is also described in A.S.T.M. Standards, 1930, part 2, p. 701.

2 A modification of the Carius method, described in "Standard Methods of Chemical Analysis," p. 145, under the title "Determination of Halogens in Organic Compounds."

In the Carius method we used a 0.1–0.2 gram sample of sulphurized oil held in a small-bore, thin-walled glass tube closed at one end and 4–5 cm long. The sample was then placed in a heavy-walled bomb-glass tube 50 cm long, 2 cm in diameter, having a 2 mm wall thickness and sealed at one end. About 10 cc of fuming nitric acid was introduced by means of a long-stemmed funnel, to avoid wetting the upper portion of the tubing. The bomb-glass tube was then sealed at its open end by fusion, placed in an iron tube in a bomb furnace, and heated 8 hr at 200 C. The tube was then cooled, opened by cutting its end off, and the contents thoroughly washed out with hot water. Sulphur content was then determined by precipitation with barium chloride, as barium sulphate, which was then filtered, washed, dried, ignited, and weighed to constant weight.

We found a difference of nearly 10 per cent sulphur content in the samples analyzed by the Eschka and Carius methods, the lower value being obtained by the Eschka method. From our raw-material balance of the amount of elemental sulphur going into the manufacture of our sulphurized oils, we found that the Carius method gave us a very close check as to the true amount of sulphur in the material analyzed.

[11] Technical Department, National Oil Products Company, Harrison, N. J.

We are therefore of the opinion that the Carius method gives an exact analysis in the determination of sulphur in sulphurized oils and that the Eschka method tends to give low values. A third method, consisting of repeatedly digesting the sample with hot concentrated nitric acid, with subsequent oxidation by means of bromine and more nitric acid, with final precipitation as barium sulphate, gave results which checked the Eschka method, but which were low.

The oxygen-bomb method for sulphur determination, as described in A.S.T.M. Standards, 1930, p. 542, was not used, inasmuch as a less complicated procedure was required.

J. F. BEKOLA.[12] Certain cutting fluids, especially the sulphurized oils, have recently acquired considerable importance due to the increase in research on the part of the users. This work has brought out much new information and many new ideas. The writer wishes to congratulate the authors on their contribution. In this and preceding papers by Mr. Boston there has been a striving toward methods for testing cutting fluids under actual working conditions.

The conclusions arrived at by the authors agree fairly with the writer's observations. However, in the cutting or drilling of the softer metals and alloys, the writer has found that a light mineral oil of a paraffin base, with a little addition of fatty acid, gave as a general rule the best results.

In agreement with the author, the sulphurized oils appear to be the most desirable in the cutting and drilling of the harder alloys. These sulphurized oils may contain anywhere from $^1/_2$ to 4 per cent added sulphur. The general tendency seems to be toward highly sulphurized oils, though the writer cannot see any appreciable advantage in cutting oils of over 2 per cent added sulphur. It is interesting to note that the No. 10 fluid used by the authors contained 4.38 per cent sulphur "compounding by analysis." The writer is inclined to believe that this was the per cent sulphur by analysis and not by compounding. If this is so, the oil probably contained about 2 per cent inherent natural sulphur and the remainder as added sulphur. It is quite commonly accepted that a straight mineral oil, of a base and viscosity similar to that of No. 10 fluid, will hold only $1^1/_2$ to 2 per cent added sulphur at saturation. This may be increased to as high as $3^1/_2$ per cent by keeping the oil in constant agitation, but this excess will immediately begin to precipitate as soon as the agitation has ceased. The heavy mineral oils will hold as high as 8 or $8^1/_2$ per cent added sulphur. Fixed or stable sulphur, of which state natural inherent sulphur is, is of little or no benefit in cutting oils. The sulphur must be in the unstable or semi-unstable form, and therefore added sulphur. If all this is true, the No. 10 fluid gave the results of a mineral oil containing $^1/_2$ to 2 per cent added sulphur.

To increase the ability of an oil to hold added sulphur requires compounding with other fats or oils. Lard oil is most commonly used for that purpose, though it is also added for its lubricating quality. A 10 per cent lard oil addition to a mineral oil of a paraffin base with a viscosity of about 150 at 100 deg will bring the added sulphur content at saturation up to about 3 or $3^1/_2$ per cent. Soy-bean oil, linseed oil, and other oils of the semi-drying and drying type will increase the amount of added sulphur retained in mineral oil much more than will the addition of lard oil. However, they have a tendency to oxidize in use, and therefore are likely to become thickened or sluggish. Pine oil has recently come to the attention of oil users mainly due to its claimed high added sulphur content at saturation, approximately 23 per cent. Whether this high percentage of sulphur will remain in suspension during use has not as yet been definitely agreed upon.

[12] Detroit, Mich.

The main function of sulphur in oils is, to all appearance, to act as a coolant. Sulphur is said to combine with oil in three states—unstable, fixed, and semi-fixed. The semi-fixed sulphur is of the form that, though nearly fixed, will become unstable after a period of time or under heat or pressure.

Many users specify that their cutting fluids be free from acids or alkalis mainly because of their corrosive effect on tools, machines, and work, and also being detrimental to the workman. Some of the other properties considered are viscosity, heat conductivity, specific heat, and at times, depending on the particular type of cutting, the film strength and adhesiveness.

The first sentence in point 4 of the general conclusions, "There appears to be no general single conclusion that can indicate the best cutting fluid for any given material," seems to be inconsistent with the data of the paper. The superiority of sulphurized oils in the drilling of the harder alloys appears to be quite clearly shown.

In conclusion, the writer would suggest that in an investigation of cutting fluids, more attention should be given to the fluids and less to the derivation of torque and thrust formulas.

Authors' Closure

Apparently Mr. Deale expects complete information as to speeds and feeds when drilling various metals with the 11 cutting fluids in which drill wear or tool life is included. General cutting speeds under various conditions are recommended for commercial purposes in much of the technical literature of drill manufacturers. It was not the purpose of this paper to give commercial data, but rather to throw light on performance under definite cutting conditions. Before tool life is studied, it was felt desirable to select, from the numerous cutting fluids used in these tests, those which were outstanding from a point of view of performance or economy, inasmuch as drill-life tests require more time and a great quantity of material to be cut. It was not the object of the paper to give a relationship between torque and tool life. In Progress Report No. 3, some information as to wear on a drill and its relation to the value of the resultant torque is shown for dry cutting in Figs. 4 and 5 and for drilling with cutting fluid No. 4 in Fig. 6. These curves show the rather gradual increase in torque and thrust as a function of the number of inches of S.A.E. 3150 steel drilled until the point of failure is reached.

The authors doubt the reliability of drilling data which have been available for the last 20 years, inasmuch as the materials being drilled today are quite different from those of 20 years ago and the drills themselves differ materially in shape and steel. The formulas presented in this paper were developed in order that a complete background of the performance of these drills when operating in the various materials with the selected cutting fluids could be known definitely. It is believed that, until these Reports Nos. 3 and 4 were issued, there was no reliable information to show that the exponents of feed and drill diameter remained constant whether cutting dry or with any type of liquid. Many comparative tests have been run from which the wrong conclusions have been reached, inasmuch as the tests covered too limited a range. It has been proved that, for a given tool shape and material, the curve expressing torque and thrust for various feeds and drill diameters for drilling and the tool-life cutting-speed curve for turning are displaced only vertically when the cutting fluid is changed. If, however, in turning, the profile of the tool is changed, the slope of the cutting-speed tool-life curve also may be changed. Various experimenters have found that the exponent of the tool life in the cutting-speed equation, when other factors are constant, is $1/7$ or $1/8$. It is known that, by changing the profile of the tool, the exponent of T can be changed to $1/5$ or $1/20$ or any other exponent desired. Unless this is definitely understood or unless the tools are identical, conclusions may be, and often are, misleading.

French and Digges found in their paper, "Rough Turning With Particular Reference to the Steel Cut" (Trans. A.S.M.E., Vol. 48, 1926, p. 533), that for their tool shape, size, and general cutting conditions, the exponent of tool life was $1/7$ (see p. 543). On the other hand, in their paper, "Turning With Shallow Cuts at High Speeds" (Trans. A.S.M.E., Vol. 52, 1930, p. 55), they found that, for light cuts, the value of the exponent of T was $1/10$ or $1/12$. A different tool shape was used, along with lighter cuts, than was used in the first paper, so it is difficult to tell whether these different exponents are due to the tool shape or to the size of cut, and yet no change from the straight line on log-log paper is obtained in the range of cuts used in the first paper, particularly as the light cuts are approached. In some of the authors' work, as yet unpublished, it is definitely proved that the change in exponents is due to the change in tool profile.

Again, in Fig. 14 of the French and Digges paper, "Turning With Shallow Cuts at High Speeds," it is shown that the exponent of tool life in the cutting-speed tool-life formula is 0.10 for dry cutting, 0.115 for lard oil, and 0.131 when water was used as a cutting fluid. Obviously, if one were to obtain comparisons at a point to the left of the intersection of any two of these three lines, one conclusion would be reached, whereas the opposite conclusion would be reached if the tests were run under conditions on the right side of the intersection. It was to make sure that no such conditions occurred in these drilling tests that the numerous combinations of feed and drill diameter were used and that the complete formula determinations were made.

It also is obvious, from the fact that various formulas are obtained for different metals, that erroneous conclusions would be reached if the performance of a cutting fluid when drilling one metal were compared with that when drilling another.

Mr. Davis brings out very clearly the usefulness of the cutting fluid as far as tool life is concerned when machining free-cutting materials. Higher speeds may be used due to the cooling action of the cutting fluid, as might be expected.

The confirming results from Mr. Ernst's discussion are indeed encouraging. His inquiry as to why sulphurized oils are superior to those of other types is appropriate. The whole investigation of cutting fluids, as conducted by the authors, has been one to find facts upon which further work can be based, with hopes that eventually enough scientific information may be obtained which will explain the phenomena of metal cutting and the use of cutting fluids. A program already has been laid out for a continuation of this study when confined particularly to sulphurized oils of different types and with different amounts of sulphur. This project, however, at present lacks sufficient support to carry it forward. Obviously, the condition of the sulphur in the oils, whether it is in a chemically active or chemically inactive state, has a great deal to do with its resultant performance. This point, however, has not been proved. It is not yet definitely known what the complete purpose is of the cutting fluid with or without sulphur, so it appears that much work is to be done before a complete understanding of metal cutting can be had and the application of cutting fluids put on a scientific basis.

Numerous experiments have been conducted by the authors with drills of the same diameter but having different thicknesses of web, but no experiments have been conducted with drills of different diameters having web thicknesses of a constant ratio to the diameter, as suggested by Mr. Ernst. The torque in drilling is affected but little by a wide range in thickness of web as a variable. The thrust, however, is affected as indicated by the formulas. The results with the 1-in.-diam drills with various thicknesses of web agree with the formulas as determined with the commercial drills used in the work of the paper.

Mr. Bekola raises a number of questions of interest. The influence of the quantity of sulphur as the only variable in a sulphurized oil has been a project for future experiments by the authors. Nothing comprehensive has been done as yet, however.

Some tests have been run with sulphurized oils in which three oils containing, respectively, 5, 10, and 25 per cent of flowers of sulphur were held in suspension. No particular difference in performance was found in several types of tests. There must be some limiting quantity of sulphur beyond which further material gain in performance, either as indicated by reduction in power, increase in tool life, or better finish, is obtained, but the authors are not in a position to state what this limit is, because of lack of information.

The authors are gratified to observe the interest in the work of this sub-committee. It is hoped that efforts may be correlated so that step by step positive information can be obtained which will lead to a final and complete solution of the problem.

RP-55-2

Grinding Cemented Tungsten- and Tantalum-Carbide-Tipped Tools Economically

By J. M. HIGHDUCHECK,[1] EAST PITTSBURGH, PA.

Present-day speeds and efficiency in handling enormous production have been made possible only by the development of modern tool equipment and set-ups which hasten service and assure quality. In this paper the author outlines a tungsten-carbide-grinding set-up which is efficient and profitable.



PERHAPS one does not exaggerate in saying that cemented tungsten-carbide cutting materials have received more publicity over a period of four years than any other product of the present machining age. In spite of all this, there are many who still consider cemented tungsten carbide as impractical for their use. The manufacturers of carbide cutting materials have been generous in presenting research information to the public, so that today the fundamentals of this cutting material are no longer a secret. The trade papers have outlined practical applications and the proper grades of carbide to use. Rules and helpful suggestions in the selection of proper speeds and feeds in the machining of certain materials are available. The materials that can and that cannot be machined by turning, milling, or drilling have been presented to the machining public, until all that is left to develop is perhaps the maintenance of carbide cutting tools.

Before describing the grinding set-up at the Westinghouse company's East Pittsburgh Works, it might be of interest to outline the system of tool control and how carbide applications are selected. Every tungsten- and tantalum-carbide application in production is demonstrated by a demonstrator who understands the full merits of this material and who can operate any machine as well as the operator on the job. Each application is watched until the operator agrees with the demonstrator that he can maintain the speed. For every application two sets of tools are provided. These tools are charged to the operator by a check system at the department toolroom. Every tungsten-tantalum-carbide application is recorded on a standard form in triplicate—one copy for the time-study department, one for the tungsten-carbide division, and the third copy for the demonstrator who made the application. Every carbide tool, large or small, is numbered. No two tools carry the same number, regardless of size, shape, or type. These tools are recorded at the tungsten-carbide division office. At this office there is kept a record of the purchase order number, cost, size and type of tool, grade of carbide material, and the division to which the tool was assigned.

When an operator is assigned to operate a machine that is equipped with carbide tools, he is charged by the toolroom with every carbide tool on that application. When the tools become dull, the operator on presenting the dull tools at the toolroom receives new or resharpened tools. The toolroom cancels all checks bearing the dull tool numbers and charges the operator with the sharp tools. In a large plant it is always difficult to keep the different sub-toolrooms from losing tools, especially when all carbide cutting tools are resharpened by the main toolroom. To overcome this condition, the sub-toolroom requisitions its tools by type and serial number. Each sub-toolroom is provided with a board upon which are five tools, representing different steps in the manufacture and maintenance of cemented tungsten- or tantalum-carbide-tipped tools. The sub-toolroom tender is responsible for noting on the requisition the classification of the reconditioning of the tool. These tools are sent to the main toolroom and resharpened by an experienced tool grinder. At the main toolroom, where the carbide tools are resharpened, there is a similar board. On this board class A represents a new tool (generally known as a new tipped tool); class B is one with the tips broken off, not more than 20 per cent of the original size; class C, a dull tool; class D, a tool requiring hand honing; class E, a tool generally used for fine finishing, which is always highly diamond lapped. These classifications have time studies in respect to type and size of tool, and the man reconditioning these tools is paid accordingly. On this board is displayed a sketch showing the correct and incorrect method of grinding (see Fig. 1). The main toolroom records the classification and the tool serial number on a form (see Fig. 2) provided by the tungsten-carbide division. The operator turns in his time as approved by the time-study department and inspector for the grinding and resharpening of carbide tools.

There are many factors to be considered in determining the proper and economical use of tungsten carbide. Most of these various factors, however, have been overestimated. In general, in preparing and maintaining carbide tools so that they may perform efficiently, grinding plays a most important part. A material so entirely foreign to steel in composition and physical properties naturally requires different grinding methods and wheels, and the principles to be observed in grinding must be given special consideration. There is no question that tungsten and tantalum carbide has been widely adopted as a "super" cutting tool, in much the same manner that its predecessor, high-speed steel, was received in the beginning of the century.

Cutting carbides should be ground with appropriate care because of their unusual properties. Localized heating of any one part of a tool bit must be avoided in order to prevent cracking

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[1] Supervisor of cutting tools and applications, Westinghouse Elec. & Mfg. Co. The author was born in Hoboken, N. J., in 1887. After graduating from high school, he attended the Gressley's Academy of West Newton, Pa., also enrolled with the International Correspondence School, of Scranton, Pa., and completed the study of electrical and mechanical engineering. He received his industrial training in pattern making and tool making while engaged with the United States Radiator Company, of West Newton, Pa. Later he was engaged by the Kelly & Jones Company, of Greensburg, Pa., as a mechanical engineer. During the World War he was employed by the Railway Industrial Engineering Company in the capacity of tool supervisor and assistant superintendent of production. After the war he was employed by the Westinghouse Elec. & Mfg. Co. as a mechanical and electrical engineer in the circuit breaker department. Later he became chief inspector of manufacturing in that division, and then was moved into his present position.

Contributed by the Special Research Committee on Cutting of Metals and presented at the Machine Shop Practice session at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

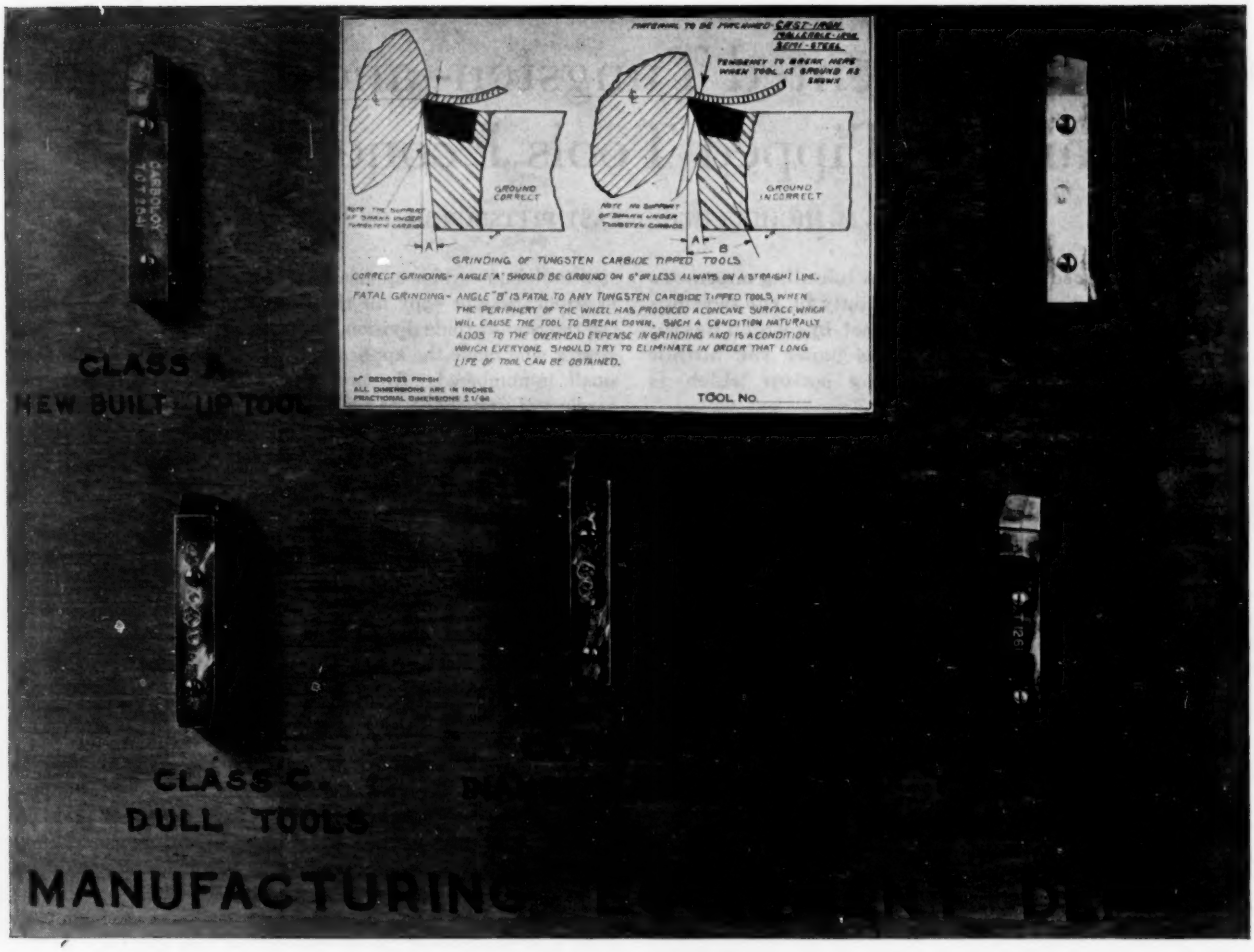


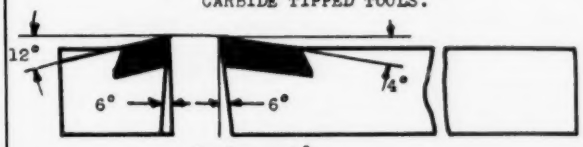
FIG. 1 MAIN TOOLROOM BOARD

(On this are shown five tools representing different steps in the manufacture and maintenance of cemented tungsten-carbide-tipped tools.)



FIG. 4 A CEMENTED TUNGSTEN-TANTALUM-CARBIDE GRINDING DEPARTMENT

DAILY
GRINDING REPORT OF TUNGSTEN
CARBIDE TIPPED TOOLS.



Angles ± 0° 2°

Date. Aug. 10 ....1931 Division..Small Tool...

Tool No.	Div. or Dept.	Classification of Gr.	Tool No.	Div. or Dept.	Classification of Gr.
403	B.C.D.	B	388	B.C.D	C
571	"	"	389	"	"
753	"	C	390	"	"
594	"	"	1676	"	B
588	"	"	586	"	"
587	"	"	180	"	B-1
544	"	B-1	376	"	D
2542	"	"	375	"	B-3
591	"	"	598	W-5	C
1427	"	"	12	"	"
1165	MAD	C	54	R	"
2873	"	"	27	"	"
2872	"	"	2477	Z-20	E
595	B.C.D	"	2476	"	"
364	"	"	1397	MD	C
499	"	"	1385	"	"
543	"	"	1588	"	"
473	"	"	636	MAD	B
470	"	"	1194	"	"

Workman's Signature..McKeag.....Check No....5....
Inspected by...H..Gabok.....Date...8/11/31...
Approved by...S.S..Enacksuch.....Time Study Dept..
This form is to be turned in daily by the Time Study Department to Foreman of W-54 after approval.

FIG. 2 EVERY TOOL IS GIVEN A NUMBER
(A record is made of each grind as to date, form, and classification. Copies are sent to the time-study and tungsten-carbide divisions.)

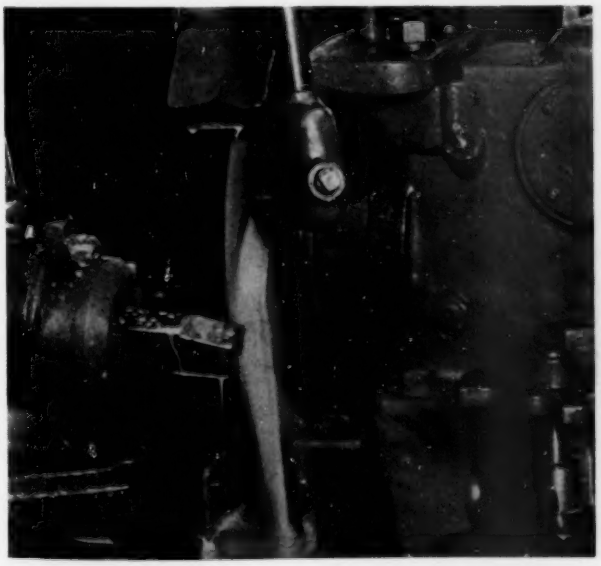


FIG. 3 ROUGH-GRINDING CLEARANCE ANGLES ON TUNGSTEN-CARBIDE-TIPPED TOOLS
(This is done on the periphery of the wheel, by the wet process.)

and checking. Soft grinding wheels should be used, and the wheel must have sufficient strength to prevent its breaking down too rapidly. Again, wheels that are too soft will make the grinding operation unprofitable. After several years of experimenting with different grinding wheels and machines, we learned that to efficiently rough-grind clearance angles on tungsten-carbide-

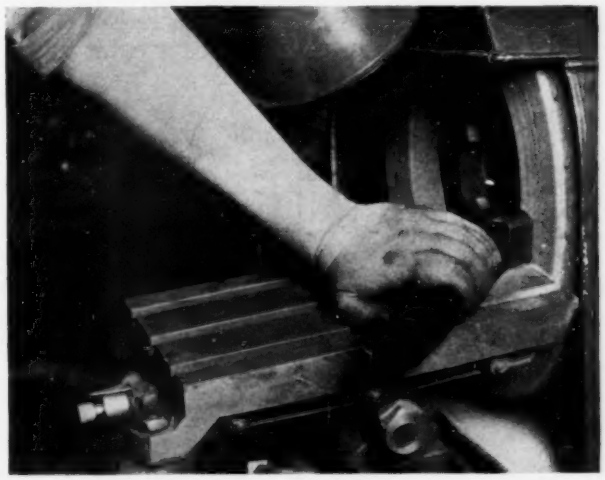


FIG. 5 FINISH-GRINDING TUNGSTEN-CARBIDE TOOLS, DRY PROCESS

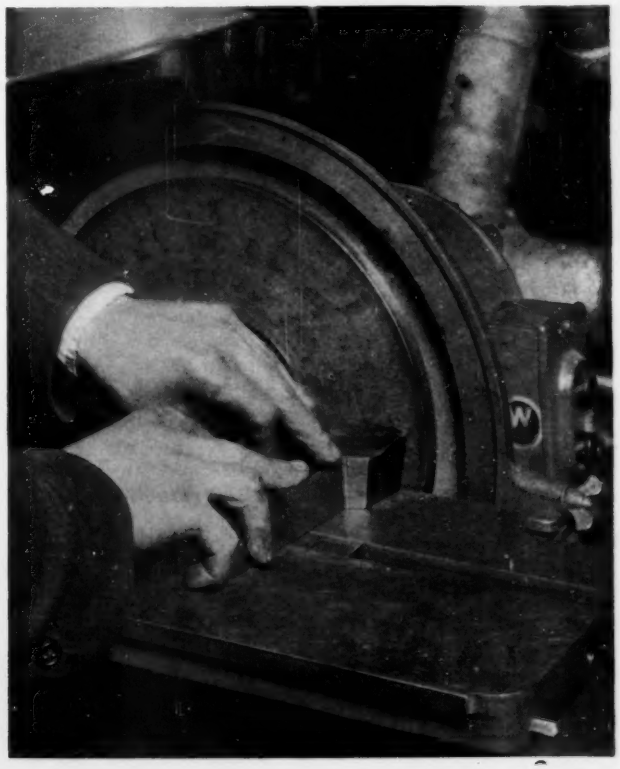


FIG. 6 DIAMOND-LAPPING TUNGSTEN-CARBIDE-TIPPED TOOLS

tipped tools, it is necessary that the contact between the tool and the grinding wheel be a line and not a surface, using plenty of water when grinding (see Fig. 3). Our experimenter demonstrated repeatedly that a plane surface of a grinding wheel cannot be used for rough-grinding carbide tools, wet or dry, successfully and profitably because the wheel surface and the surface being ground soon coincide, with the results of no cutting and

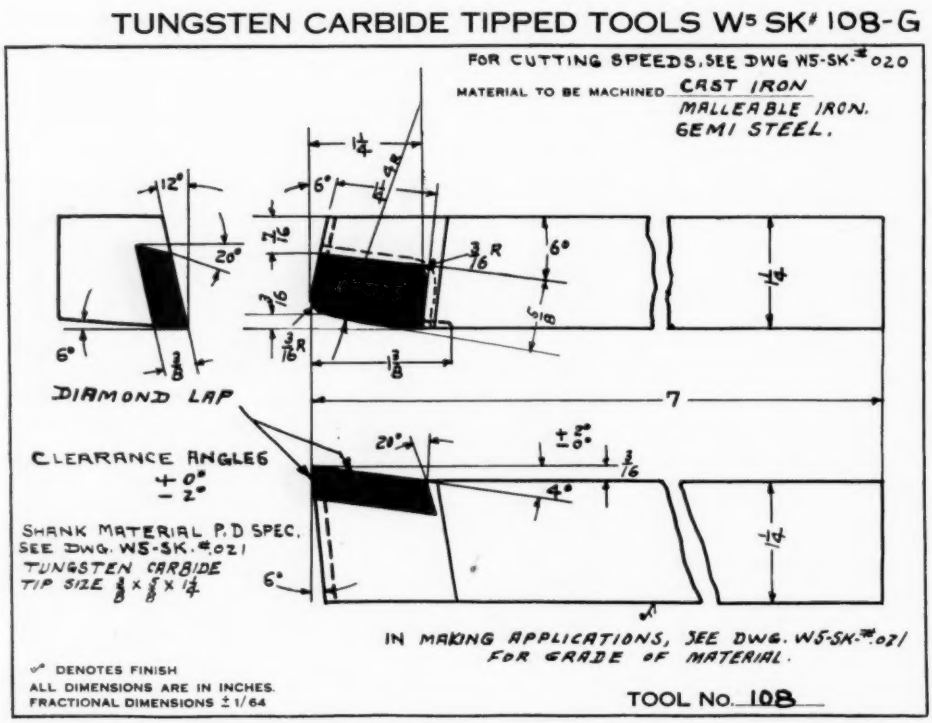


FIG. 7 DIMENSION DRAWING OF A POPULAR PRODUCTION TUNGSTEN-CARBIDE-TIPPED TOOL

much heating, generally ending in checking or cracking the carbide tip. It must be remembered that the carbides are not steel or alloys, but are bonded materials composed of almost microscopic particles approaching the diamond in hardness.

In a plant where thousands of carbide tools are used, the system of maintenance becomes important. To allow operators to grind their own tools when in production is out of question, both from the grinding-equipment and skill standpoints. We have found that the central grinding department system is the most economical system. In Fig. 4 is shown a carbide grinding department which permits the grinding of tipped tools from $1/4 \times 1/4$ to 3×3 in. square. As already stated, all tools that are new tipped are rough-ground by line contact. There are two sizes of machines to cover the field of shank sizes. The machine used for roughing tools from $1/4$ in. square to $1\frac{1}{2}$ in. square is shown in Fig. 3. After the tool is roughed down, it is transferred to the machine shown in Fig. 5, where it is finish-ground dry, semi-offhand. If the tool is assigned to machining resinous products, it receives a diamond lapping operation on a machine as shown in Fig. 6. The tools are carefully inspected and returned to the department to which they are assigned.

Generally, the most interesting point to the user of carbide tools is the cost of the maintenance. Before we inaugurated the set-up, as shown in Fig. 4, to grind a tool, as shown in Fig. 7, it required $1\frac{1}{2}$ hr. At the present time, by means of the new set-up, the same tool is ground to finish size at an average time of 21 min, or a saving of 1hr 9 min on each tool. The tool shown in Fig. 7 is generally recognized by the trade as Carboloy type 14. We have purchased these tools in semi-finished form, in which the supplier of carbide furnishes the tip and brazes it to the shank, all grinding being done by us. In brazing the carbide tip to the shank the supplier generally allows the carbide to project from the shank, which leaves from 10 to 20 grams of cemented carbide to be ground off on the type of tool shown in Fig. 7. An experienced tool grinder can remove this carbide at the rate of 1 gram per minute. Of course, this record is based on equipment such as shown in Fig. 3. To remove carbide by the dry-grinding process, as shown in Fig. 5, the best results obtained were 1 gram removed in 7.1 min, but with considerable danger of heating the tool. The grinding wheels play a very important part in the success of good carbide tools. The proper selection of grinding wheels can be made only by experience and trial. The manufacturers of grinding wheels can furnish the user general information, but special cases require special attention. At the Westinghouse plant, in the set-up shown in Fig. 4, the following grinding wheels are used. For rough grinding (see Fig. 3), Carborundum 80 grit, grade R, bond W.G.G., or Norton No. 3760/1-19. For finishing (see Fig. 5) on one end of the machine, all tools are semi-finished on a Norton or Carborundum wheel, the same as the one used on the machine in Fig. 3, except on a cup wheel design. After this operation the tools are finally finished on the opposite end of the machine on a similar cup wheel, Carborundum or Norton No. 37100/2-H6. The cup wheels are of the same diameter and bore as those used on the machine shown in Fig. 3. The idea is to use the grinding wheel after it is worn down on the machine for which it was intended. In this way maximum wheel life can be obtained. The wheel cost in grinding new tipped tools averages 35 cents per tool, plus the operator's time and the department overhead. The best wheel performance is obtained between 4500 to 5000 sfpm.

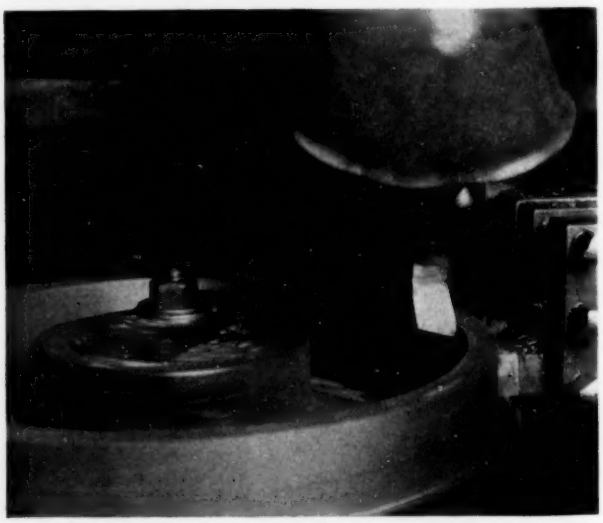


FIG. 8 PRODUCTION JOB ON A BORING MILL
(Two carbide tools turning semi-steel at 280 sfpm, using 1/16 in. feed per revolution.)

Discussion

MALCOLM F. JUDKINS.[2] Because the difficulties and expense of grinding cemented carbide tools have been asserted by adherents to older cutting materials, it is gratifying to have a large user affirm that grinding is no obstacle to use.

We have found that, if plenty of coolant is thrown directly on the tool being ground, wet grinding is cheaper, faster, and safer than dry grinding. The coolant keeps the carbide cool and flushes away the abrasive particles resulting from wheel wear and maintains a clean, free-cutting wheel surface.

One type of tool bit, $^3/_8$ in. thick and 1 in. wide, with a $^1/_8$ in. thick tip across the 1 in. blade width, gave trouble until we used a hydraulic-feed surface grinder. Carbide can be ground very rapidly and with rather hard bond wheels, with no danger of cracking, checking, or chipping, if the time the wheel is in contact with the work is held to a minimum. We use a table traverse of 540 in. per minute and a wheel speed of 5500 sfpm, with a down feed of 0.001 in. and a cross feed of 0.042 in. per pass. This operation is done under a copious supply of 1-to-50 soluble oil coolant thrown directly on the work. The output per 9-hr day is about 200 blades of the inserted-blade milling-cutter type. The blades must be held to 0.375 in. plus or minus 0.001 in. all over. The magnetic chuck load is six pieces. About 0.015 in. stock is removed—0.003 in. from the carbide face and 0.012 in. from the reverse side.

Recent grinding tests on the milling-cutter blades referred to, using a much heavier hydraulic surface grinder than the one installed in our plant, with a 3-in.-width wheel and a table feed of 1200 fpm, showed that the chuck load could be increased from 6 to 12 and that 0.0058 in. and 0.0065 in. were removed in 4 and 4.6 min, respectively, for total vertical feeds of 0.025 in. and 0.030 in. Blades were ground under these conditions until the carbide was only $^5/_{64}$ in. thick. These blades are tipped with a hard grade of Firthite.

Because we have little use for the part remaining after a cup wheel has been used to the scrapping point, we use ring wheels cemented with sulphur in steel adapter plates. Our grinding-wheel scrap is very low.

The author reports a much higher rate of stock removal when grinding wet than when grinding dry. We do almost all of our finishing wet. The writer would like to know the author's objections to wet finishing.

Almost all of our tools are subjected to a lapping operation, but we no longer use diamond. We have found that on a Belgian cast-iron lapping disk with Carborundum finishing compound No. RS-9 or RI medium, the lapping action is very rapid and effective and is of course much cheaper than using diamond dust suspended in olive oil or some other carrier.

The author recommends roughing tools on the Sellers type of wet-tool grinder which provides for line contact, because the tool is moved in the vertical plane tangentially past the periphery of a grinding wheel. For those shops which do not have this type of grinder we have recommended that the steel shank beneath the tip be ground on an aluminum-oxide wheel at from 2 to 5 deg. more clearance than the desired clearance on the carbide tip itself. The carbide is then ground on a silicon-carbide wheel. This method is desirable because if the carbide and steel are ground together on a wheel suitable for cemented carbide the steel loads the wheel and glazes the surface, tending to overheat the carbide, with possible danger of cracking or checking it.

COLEMAN SELLERS, 3D.[3] The author states that for efficient rough grinding of tungsten-carbide tools it is necessary that the contact between the tool and the grinding wheel be a line and not a surface. This is understandable, as when the surface of the tool is in contact with the surface of the cup wheel, heat is generated quickly, and no amount of water will cool this surface. Water cannot reach any further than the edge of the tool, so that the edge will be cool and the center of the surface hot. For this reason, with surface contact, wet grinding is worse than dry.

Those who have ground a large number of carbide tools in the last few years have usually come to the same conclusion—that wet grinding is safer and quicker. With a line contact the water can at all times reach the part of the tool being ground and prevent overheating.

Grinding with a line contact will not do by hand. If a tool is pressed against the periphery of the wheel, we soon have surface contact. The tool must be passed back and forth in a plane while in contact with the wheel or the wheel must be moved in a plane and the tool held stationary.

The tool-crib system is certainly the most economical in any shop unless very small. The old custom of having every man grind his own tools can no longer be tolerated. When most of the tools used were made of high-speed steel, it was possible to have individual grinding only because the tool steel was capable of considerable abuse and the tools would cut even though the angles, rakes, etc., were not the best or most efficient. With the carbide-tipped tools, improper grinding is fatal, and shop executives are realizing that they must have central grinding. They are also finding that their tools made of high-speed steel or other materials are giving much more efficient performance when they are handled by a tool crib and centralized grinding.

We have found that the time required to grind depends on the hardness of the material as well as on the amount to be ground off. The time of 21 min. for the tool shown in Fig. 7 seems about right for the harder grade of Carboloy, grade 44. From our experience, a similar tool of grade 55 with $^3/_{16}$ in. projection of the tungsten carbide would take about 16 min.

The figure of 1 gram per min is interesting. This is probably a very hard grade of the material. Our rate of grinding on grade such as 44 is from 2 to $2^1/_2$ grams per min and on grade 55 or equivalent 4 to 5 grams per min. In fact, we have rough ground the softer grade as fast as 13 grams per min.

Wheel wear will vary from 0.15 to 0.35 cu in. per gram of carbide, depending on the hardness, provided the proper wheels are used. With wheels too soft, this figure would be much higher. Lower figures indicate too hard a wheel, which tends to glaze and burn the tool rather than to grind it.

The cost of wheel is about $8^1/_2$ cents per cu in. of wheel. Our grinding rate averages 0.2 cu in. of wheel per gram of tungsten carbide on a grade such as 55 Carboloy. With 20 grams to be ground off, our wheel cost would be $0.085 \times 0.2 \times 20 = 0.34$, which agrees with the author's figure within 1 cent.

ROGER D. PROSSER.[4] From this paper it would appear that all of the finish grinding is done dry. It is the writer's opinion that wet grinding, with an ample flow of water, is far superior to the dry process, for both roughing and finishing. There is much less danger of cracking the hard metal tip, because it is kept cool by the water. The grinding can also be done much faster as the wheel does not glaze, a much heavier feed can be used, and it is not necessary to change tools or stop and wait for the tool to cool off because it can no longer be held in the hand. Furthermore, with wet grinding, there is no danger that the operator will suddenly place the hot tool in water, with consequent cracking of the tip.

[2] Engineer, Firth-Sterling Steel Company, McKeesport, Pa.

[3] Executive Engineer, Wm. Sellers & Co., Inc., Philadelphia, Pa. Mem. A.S.M.E.

[4] Thomas Prosser & Son, New York, N. Y. Assoc-Mem. A.S.M.E.

Another point is that the rest should have an indexing device, so that the clearance and other angles can be maintained accurately. An important factor in the operation of tungsten-carbide tools is to keep the clearance angle as small as possible consistent with free cutting, and a few degrees' excessive clearance angle makes considerable difference in tool life.

The author's estimates of the amount of cemented carbide which it is possible for an experienced tool grinder to remove in a given time seem excessively small. In finishing, of course, much depends on the class of finish desired and the type of tool being ground, but the figure of 1 gram per min given for wet grinding seems very low. In the Widia tool manufacturing department at Krupp's, the average time required to rough grind away the excess metal on a simple tool, making it ready for finish grinding, is less than 1 min, with from 3 to 10 grams to be removed. The finish grinding is also done very rapidly, by the wet process. It is not often necessary in this country to remove large amounts of cemented carbide, nor is it desirable, on account of the expense, but the foregoing will indicate how much a proper wet grinder with plenty of power and a large wheel will facilitate and speed up the grinding processes where there is considerable of this work to be done.

Author's Closure

The author appreciates the frank discussion, which has brought out many interesting points regarding economical grinding of cemented tungsten- and tantalum-carbide cutting tools.

It is particularly evident that all discussing this subject think alike with regard to the importance of fast and accurate rough grinding of carbides, especially by the wet process, and on the periphery of the grinding wheel (see Fig. 3). The question asked by Mr. Judkins and Mr. Prosser regarding the author's objections to wet finishing brings up the point that the author has none. This operation, perhaps, is a matter of opinion and the kind of set-up one has for grinding carbide tools. The rough-grinding operation as explained in the paper is so accurate and fast that very little material is removed when finish grinding the tools. Therefore a simple dry-grinding operation (first a semi-finish and then a final finish operation) brings the cutting edge to a very fine degree of finish. The author has found that very little heat is generated in the method of finish grinding. Therefore it would be useless to use water. On the other hand, on this type of a machine the water would splash around and would cause the workman to become more or less inaccurate in his work.

RP-55-3

A Test Method for Determining the Quantitative Effect of Tube Fouling on Condenser Performance

By P. H. HARDIE[1] and W. S. COOPER,[2] BROOKLYN, N. Y.



P. H. Hardie

The purpose of this paper is to show the importance and practicability of adding another vital measurement to condenser tests which will give a specific value for the degree of tube fouling existing at the time of the test. Since the majority of condenser installations are subjected to some type of fouling, and since conventional means of cleaning seldom if ever restore the tubes to their original state, it is evident that the performance of many condensers cannot be correctly appraised without a knowledge of their cleanliness factors. Cleanliness factor is the term used to express the degree of tube fouling, and is defined as the ratio of the heat transmittances[3] of fouled and new tubes subjected to the same operating conditions. Several methods of measuring the cleanliness factor are described, and data from six tests are included. The importance of avoiding inconsistent procedures in making the measurements and computing the data is stressed.



W. S. Cooper

CONDENSER performance, like that of other power-plant equipment, has shown marked improvement in the last ten years. During this period there has been a growing tendency toward more specific and stricter guarantees. This was possible only for plants that were fortunate enough to be free from condenser-tube fouling. For the others, which represent by far the majority, equitable bases for making guarantees were impossible because of the lack of a reliable method for determining the condenser cleanliness factor. If condenser performance is to be correctly appraised, it is just as important to know the cleanliness factor as to know the circulating-water temperature and velocity. The quantity of scale or slime on the tube surfaces is not in itself a direct measure of the reduction in performance. Therefore attempts to estimate the cleanliness factor by visual inspection of the tubes cannot be considered as being any better than rough estimates.

Although under certain conditions fouled tubes can be cleaned thoroughly enough to restore their performance to that of new tubes, their cleanliness factor after a thorough cleaning is generally sufficiently below unity to warrant measurement. Some contracts specify the manner of cleaning to be used just prior to the acceptance test, with the expectation that such cleaning will produce either new-tube condition or some known cleanliness factor less than unity. This procedure is only approximate, because the same manner of cleaning will not produce at all times the same cleanliness factor at a given station, and the variation will probably be greater at different stations.

The authors will describe the development over a period of five years of a test method that is designed to eliminate the uncertainty of fouling, and when applied to acceptance tests, manufacturers have a rational basis on which to make performance guarantees and purchasers have a reliable means of checking them. Since the A.S.M.E. Power Test Code Committee has recently decided to incorporate in the condenser code a method for measuring cleanliness factor, it is believed that a description of this test method, which has been successfully applied in the acceptance tests of five large condensers, will be of interest and value to others.

Cleanliness Factor Defined

For a given water velocity, cleanliness factor is the ratio of the heat transmittance of a fouled tube to that of a new tube through which water has been passing only the short time necessary to get the test started. The two tubes must be supplied with circulating water at the same inlet temperature, and the velocities through the tubes must be equal. Similarly, the tubes must be subjected to the same conditions on the steam side.

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[1] Test Engineer, Brooklyn Edison Company, Inc. Assoc-Mem. A.S.M.E. Mr. Hardie was graduated from the Alabama Polytechnic Institute with the degree of B.S. in Mechanical Engineering in 1921 and later received the degree of M.E. The following year was spent at the Massachusetts Institute of Technology, specializing in design and testing. He received early practical training with the Hardie-Tynes Manufacturing Company, of Birmingham, Ala., and was associated with the Westinghouse Elec. & Mfg. Co. until 1926, doing steam-turbine design work. Since 1926, he has been engaged in power-station testing for the Research Bureau of the Brooklyn Edison Company, Inc.

[2] Assistant Engineer, Brooklyn Edison Company, Inc. Assoc-Mem. A.S.M.E. In 1917–1918, Mr. Cooper was a member of the Instrument Test Department, Taylor Instrument Companies; in 1924, was graduated from Cornell University with the M.E. degree; in 1932, received the M.M.E. degree; was appointed to the engineering teaching staff, Cornell University, in 1924, and was connected with the Turbine Engineering Division of the Westinghouse Elec. & Mfg. Co. during the following three summers; has been associated with the Research Bureau of the Brooklyn Edison Company, Inc., since 1926.

[3] Variously termed thermal transmittance, heat-transfer, and heat-transfer coefficient, all expressed in Btu per hr per sq ft per deg F log m.t.d. (or mean temperature difference).

Presented at the meeting, New York, N. Y., November 28, 1932, of the Metropolitan Section of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

By "new tube" is meant a tube in the same condition as when received from the manufacturer, and of course it must be of the same size and material as the fouled tube.

Having defined the cleanliness factor of a single tube, it is merely a matter of selecting and testing a sufficient number of representative tubes from different parts of the tube bank and averaging their cleanliness factors in order to obtain the cleanliness factor for the entire condenser. This is similar to well-known sampling processes, which while not 100 per cent accurate, are sufficiently so to justify their use. Certainly it is a far more reliable procedure than resorting to guesswork, which has been the general practise in the past.

Two Means of Measurement

The measurement of tube-cleanliness factor can be done in two ways: first, by isolating the test tubes and testing them in place in the condenser; second, by removing the selected tubes and testing them in a calorimeter. Both methods have their good points, but the authors consider the test in place preferable for the following reasons:



Fig. 1 Typical Installation of Test-Tube Thermometers in Outlet Connections

(1) The removal of tubes and subsequent shipping, which is required if a tube calorimeter is not available at the plant, may result in an improvement in the cleanliness of the tubes, thereby giving an erroneous value for the cleanliness factor of the condenser.

(2) The apparent increase in accuracy of a laboratory calorimeter test is offset by possible inaccuracies inherent with this method. For instance, the measurements must be made on short sections of the condenser tubes, and therefore greater accuracy is required in measuring the temperature rise. Actual condenser tubes are seldom an exact multiple of the length required for a calorimeter, which means that a small section of each tube would have to be discarded. If the cleanliness factors of the sections of a test tube differed greatly, their average might not give the same value as that obtained if the full-length tube had been tested.

(3) Should the cleanliness factors of the full-length tubes differ considerably, there would be no means of computing the weighted average. With the test in place, this can be done easily, since the new tube in each group is a measure of the relative effectiveness of its respective part of the condenser. The cleanliness factors of the tubes near the top, for example, have a greater effect on the performance than ones near the air offtake, because they transfer more heat.

(4) Where tubes foul rapidly, it is important that the cleanliness factor be measured for each run. If the calorimeter method is employed, it would be impracticable to test any more than one group of tubes removed before the first run and one group removed after the last run.

Some of the advantages of both methods may be obtained by moving all the selected old tubes to the top of the condenser and testing them simultaneously with the overall test. Objections (1) and (3) given for the calorimeter method would still apply, however.

Test Procedure

For the cleanliness-factor determinations discussed in this paper the usual process of measuring individual tube heat transmittance was employed. The data required for its evaluation are of course the tube's cylindrical area, the enveloping vapor temperature, the inlet and the outlet temperature of the cooling water, and the rate of flow of the water. As implied in the foregoing section, the cleanliness factor data are observed simultaneously with the overall condenser data. This procedure provides a continuous record of variations in cleanliness factor throughout the test.

For the tests on the single-pass units reported, the selected test tubes were isolated in place by means of rubber-hose jumpers which carried their cooling-water supply across both water boxes. Rubber hose has the advantage of insulating the test-tube water from temperature effects of the water in the boxes. On the inlet end the test tubes were connected through the water-box cover to a distributing manifold receiving water from the same source as for the entire condenser. With this arrangement water of the same quality and very nearly the same temperature as at the inlet to the condenser was available. A "master valve" installed in the supply line to the manifold served to regulate the supply pressure to the values required by the test. The outlet end of each test tube was connected with rubber hose to a control valve, and a nozzle for measuring the water flow through it. This arrangement afforded a simple method for maintaining individual tube velocities equal to the estimated average water velocity through the entire tube bank.

Mercury thermometers inserted directly into the water stream through rubber stoppers were used to indicate the inlet and outlet temperatures of each tube. Fig. 1 shows an installation of thermometers in the outlet of the test tubes for condenser *A*. Where several tubes are tested in a group, only one inlet thermometer per group need be used, but a thermometer is required in the outlet of all tubes. The type of thermometer used by the authors was 10 in. long and had a scale graduated from 30 F to 100 F by 0.5-deg divisions. The 0.5-deg divisions were large enough to permit a fairly accurate estimation of 0.1 deg.

Where two-pass units are involved, modifications in procedure are needed as follows: Fundamentally, the same number of test tubes may be used as in the single pass, but each tube in the first pass is combined with a corresponding one in the second pass. The test tubes are projected across the reversing box just as with the single-pass unit, but the corresponding tubes of any two groups are connected together outside the reversing box. The inlet-water temperature of a given group is read as before, and the outlet temperatures of all tubes are read at the outlet of both passes.

For measuring the head on the nozzles, mercury manometers were used for some of the tests and water-column sight glasses for the others. With a mercury manometer and closed nozzle chamber, the range of measurement is greater, but this means is not as accurate as measuring the head on the nozzle directly in feet of water. The construction of the equipment for the latter method, only, will be described. Fig. 2 shows the arrange-

ment of the standpipes used on one of the tests described. The standpipes shown are 5-ft lengths of standard 3-in. iron pipe open at the top and threaded at the base. The nozzles were made from 3-in. brass pipe plugs and attached to the standpipes by means of couplings. The plugs were bored and reamed to 0.5 in. in diameter, rounded and polished on the inside to a $1^3/_8$-in. radius, and given a $^1/_{16}$-in., 45-deg chamfer on the discharge side. Each nozzle was given a weighed flow calibration over the range of flows expected. Fig. 3 shows diagrammatically the flow circuit for test tubes installed in a single-pass condenser.

The question of selecting test-tube locations that will represent the condition of an entire condenser in regard to fouling is one of importance. The number of old tubes required to determine with engineering accuracy the cleanliness factor for any given condenser depends on the size of the condenser, the layout of the tube bank, and the uniformity of fouling. The more uniform the distribution of fouling, the more nearly will a limited number of sample tubes represent the condition of the entire tube bank. As an expedient in producing uniformity of fouling, the condenser tubes should be cleaned just prior to the test by the customary method in use in the plant. No rational formula may be stated that will give the correct number or location of test tubes which will adequately represent the average condition of the entire condenser. The exercise of experience and judgment must therefore be relied upon. Eighteen old tubes and six new tubes have been found by the authors to be adequate for large condensers.

Two general schemes of test-tube arrangement have been used. With the first arrangement, a "single group" located near the top of the tube bank is used. The group contains one or more

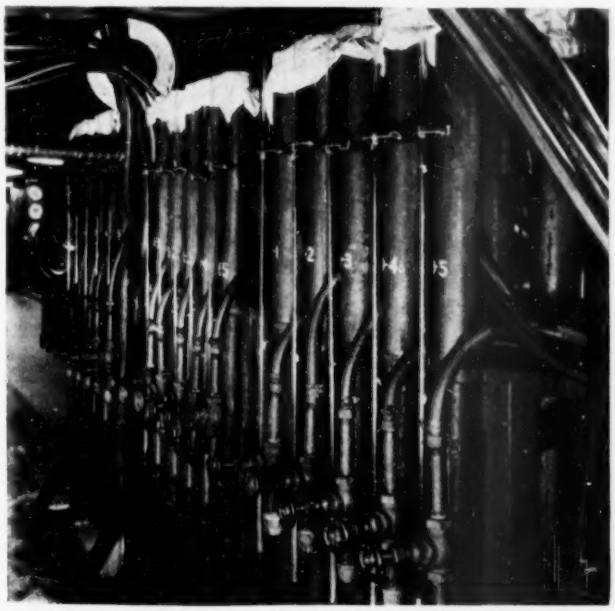


FIG. 2 NOZZLE STANDPIPES USED FOR REGULATING TEST-TUBE WATER FLOW

new tubes installed just prior to the test and several old tubes that have been removed from various representative parts of the tube bank and inserted in this group. The group may also contain an old tube that has not been moved. The location of the test group for this method is shown in Fig. 4 for condenser *A*. The second scheme, known as the "multiple group" method, consists in selecting several groups of tubes at various representative positions in the condenser's tube bank and testing them in place. Each group contains one new tube installed just before the test and one or more old tubes that have been in continuous service up to the time of the test. This method was used for the other five tests reported. The test on condenser *C* was made with two test tubes per group consisting of one new tube and one old tube. For the tests on condensers *A* (multiple group) and *B*, one new tube and three old tubes per group were used, and for *D* and *E* one additional new tube was included which was supplied with non-fouling fresh water (city water). The purpose of the fresh-water new tubes was merely to indicate any tendency of the standard new tube to foul during the period of test.

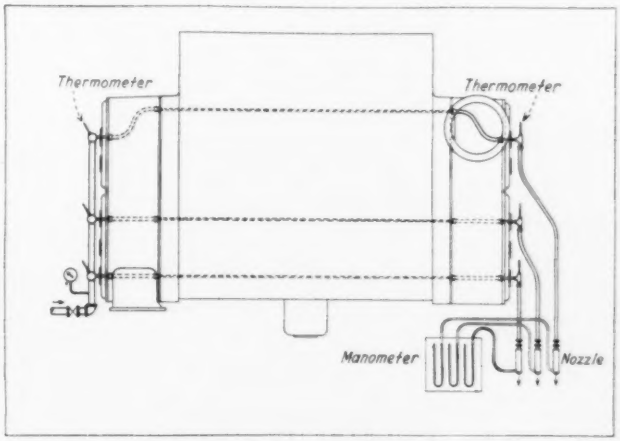


FIG. 3 ARRANGEMENT OF APPARATUS FOR TESTING TUBES IN CONDENSER

In tests *A* and *B*, the test groups were located symmetrically with respect to the vertical center line. Two of the groups were in the top peripheral row of tubes, while the others were taken fairly near the periphery. The thought at the time these two tests were run was that groups located too far into the tube bank would not yield reliable results because of the reduced tube loading and increased air richness in these parts. Nevertheless, it is probably a tenable assumption that these factors would affect both old and new tubes alike and so would not affect the reliability of the cleanliness factor as measured. In tests *C*, *D*, and *E*, the group locations were taken in more diverse positions without regard to symmetry of locations. For these three tests the average performance of the old test tubes naturally agreed more closely with the overall performance. No doubt certain other combinations would have proved equally adequate.

Test runs of 1 hr duration should be made at several condenser loads (Btu per hr) with each velocity at which it is possible to operate the circulating pumps. The test-tube temperatures should be read every $2^1/_2$ min in order to produce reliable averages. If test runs are longer than 1 hr, the frequency of reading these temperatures may be safely reduced. Test runs should be made in rapid succession with as little time allowed between them as possible in order to minimize the tendency of the new tubes to foul. If there is considerable time in between runs, as when tests must be discontinued overnight, the water supply to the new tubes should be shut off on the inlet end and allowed to drain in order to preserve their cleanliness as well as possible. The water flowing through the old tubes must, however, continue flowing at a velocity corresponding to the normal operating velocity in order that they remain representative of the entire condenser.

METHOD OF COMPUTATION

The heat transmittance of the old and new tubes is readily computed from the following equation:

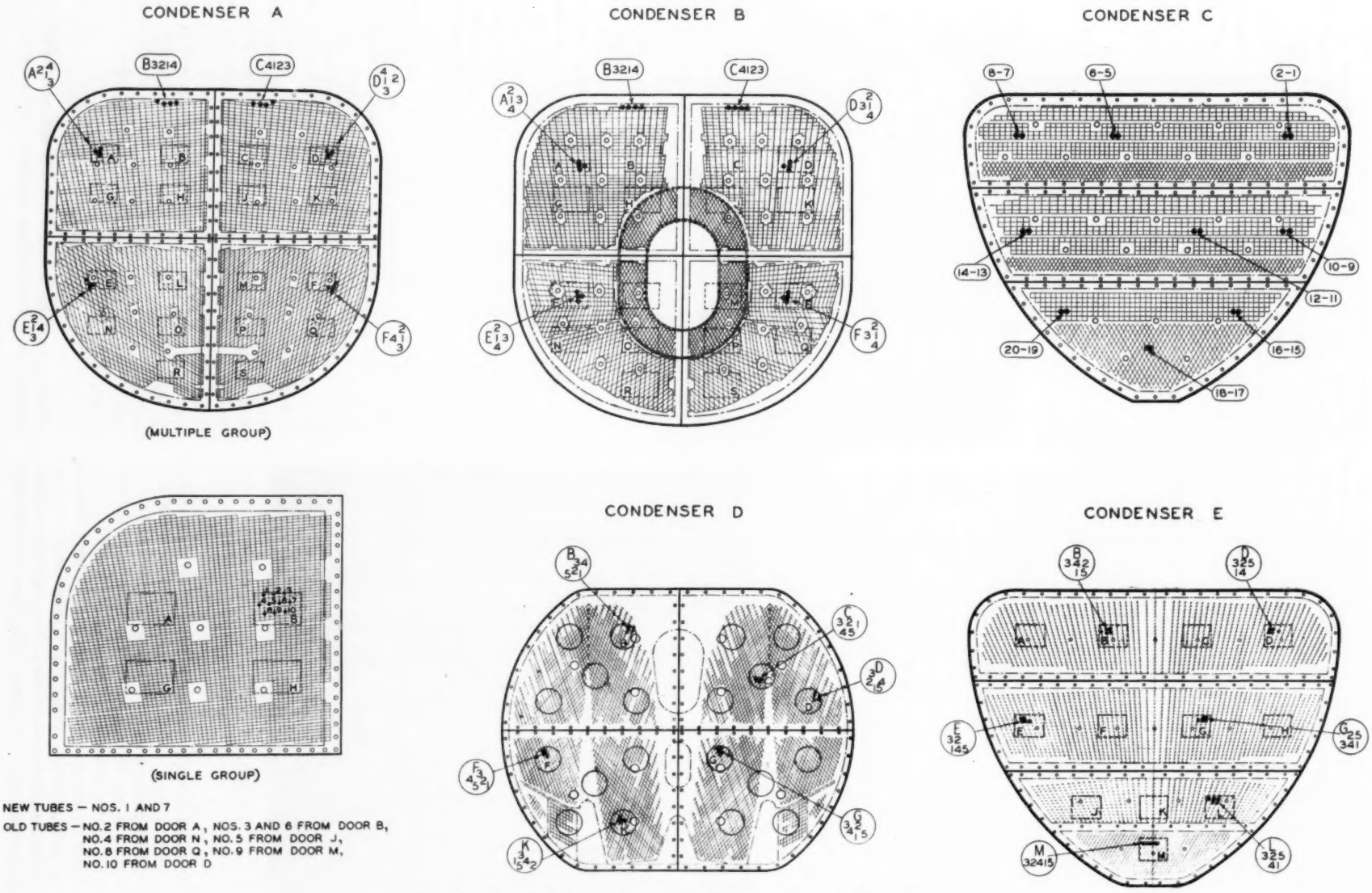


FIG. 4 TUBE-SHEET DIAGRAMS, SHOWING LOCATIONS OF TEST TUBES

$$U = \frac{WC}{S} \log_e \frac{T_a}{T_b} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where

U = heat transmittance, Btu per hr per sq ft per deg F log m.t.d.
W = water flow, lb per hr
C = specific heat of water
S = surface area per tube, sq ft
T_a = initial temperature difference, deg F
T_b = terminal difference, deg F

The equation given is based on the true, or logarithmic, mean temperature difference. For any water velocity, WC/S is a constant and may be used as such for all runs at a given velocity. For the group method, the cleanliness factor of each old tube is determined, as implied in the definition, by dividing the transmittance of each old tube in the group by that of the new. The average cleanliness factor for the entire condenser, at any water velocity, is properly found by first averaging the transmittances of all the old tubes and of all the new tubes, followed by computing the ratio of these two averages.

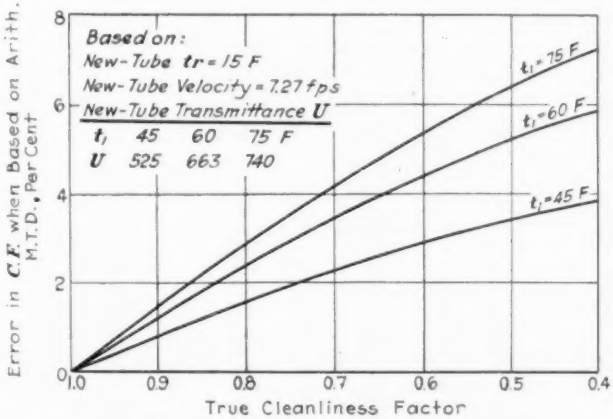


FIG. 5 EFFECT OF ARITHMETICAL MEAN TEMPERATURE DIFFERENCE ON ACCURACY OF CLEANLINESS FACTOR

In computing the transmittance either for the entire condenser or for individual tubes, the true, or logarithmic, mean temperature difference must be used if serious error is to be avoided. The effect of using the arithmetic mean temperature difference is to indicate a higher cleanliness factor than actually exists. The magnitude of this error is shown graphically in Fig. 5.

The performance of the test tubes should, in strict analysis, be referred to the saturated-vapor temperature corresponding to the absolute pressure adjacent to them. For those test tubes located well down into the tube bank, there is an appreciably lower vapor temperature due to the vapor pressure drop across the tube bank. Referring the performance of these test tubes to the vapor temperature at the condenser neck, results in giving a value of cleanliness factor which is higher than the true value, as can be seen from Fig. 6. For tests on condensers which have a low pressure drop, the error is small, and this simplification in the computation is therefore justifiable. For the test on condensers *A*, *B*, and *D*, the test-tube performance was computed using the vapor temperature at the top for all tubes, while for the test on condensers *C* and *E* the more accurate method was used. The vapor temperature for each group in the latter case was obtained by assuming a linear relation to exist between the number of tubes and the temperature drop from the top of the condenser to the air offtake.

The condenser circulating-water flows are generally known only approximately until after the test runs are concluded, unless preliminary checks have been made. Consequently it may be found after test that the estimated velocity at which the test tubes were operated did not exactly equal the average for the entire condenser. As explained in a later section, the cleanliness factor varies inversely as some power of the velocity. Using this relation, the actual cleanliness-factor measurement may be corrected, within limits, to the average velocity in the condenser which existed during the test.

The determination of the velocity exponent referred to in the foregoing may be accomplished as follows: Since,

$$CF = f(1/v^n) \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

where

CF = cleanliness factor
v = velocity in any units
n = velocity exponent for cleanliness factor,

it is found that

$$\frac{CF_1}{CF_2} = \left(\frac{v_2}{v_1}\right)^n \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

Hence

$$n = \log \frac{CF_1}{CF_2} / \log \frac{v_2}{v_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

Since test runs are usually performed at two or more water velocities, it is a simple matter to solve for the exponent. As an alternative method, CF may be plotted versus velocity on log-log cross-section paper. The slope of the resulting straight line is the value of the exponent.

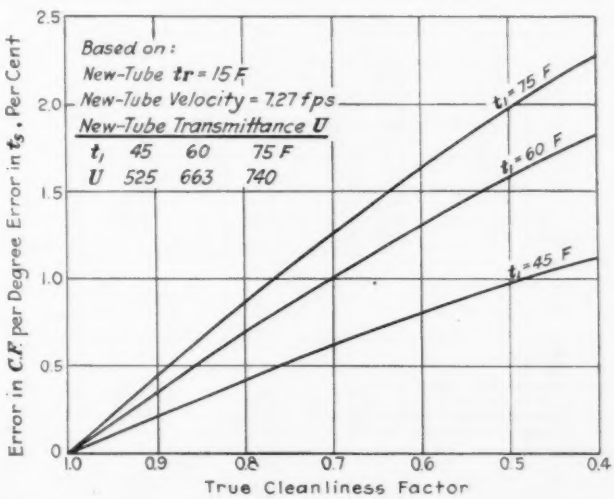


FIG. 6 EFFECT OF STEAM TEMPERATURE ON VALUE OF CLEANLINESS FACTOR

TEST DATA

Tables 1 to 6, inclusive, are introduced to show the results of typical cleanliness-factor tests. The size of the condensers in which these tests were made ranged from 85,000 sq ft to 101,000 sq ft of condensing surface. With the exception of condenser *C*, the test tubes were designated by letters and numbers, the letters referring to the groups and the numbers to the tubes in the group. In condenser *C*, using one new and one old tube per group, the test tubes were numbered consecutively. Fig. 4 may be used

conveniently in connection with the tables when studying the performance of the test tubes with reference to their location.

Application

The cleanliness-factor measurement probably will have its greatest field of application in connection with acceptance tests. In fact, the method described in this paper was developed with that particular purpose in mind, although it is equally applicable to other condenser tests.

The best method of illustrating the application of the cleanliness factor is to show how it is taken into account in the performance curves guaranteed by different condenser manufacturing companies during the last five years. These differ somewhat as to the expected reduction in the overall heat transmittance for cleanliness factors less than unity. Three typical sets of performance curves or cleanliness factors from 1.0 to 0.5 are shown, in Fig. 7. These curves are plotted for the variation in heat transmittance with load. A per cent scale is used for the ordinate because other conditions, such as water velocity and temperature, tube diameter, etc., influence the absolute value of the heat transmittance.

Fig. 7 (*a*) shows the simplest method of guaranteeing for cleanliness factors less than unity. As can be seen, the product of the cleanliness factor and the new-tube heat transmittance is the expected heat transmittance for the fouled conditions irrespective of the load.

Fig. 7 (*b*) shows another way of guaranteeing for cleanliness factors less than unity. Here the heat transmittance for a fouled condenser operating at high loads is the product of the cleanliness factor and the new-tube overall heat transmittance. These lines which are horizontal extend to the left until they intersect the unity cleanliness-factor curve. For lower loads the unity cleanliness-factor curve is followed. This method is similar to the first at high loads, but not at low loads.

The third method of guaranteeing is shown in Fig. 7 (*c*). These graphs show the shape of the performance curves affected by both cleanliness factor and temperature. Only at 75 deg circulating-water temperature is the performance for fouled conditions equal to the product of the new-tube overall heat transmittance and the cleanliness factor. For lower temperatures the overall performance is assumed to be reduced less than 1 minus the cleanliness factor; e.g., the performance at 55 deg

TABLE 1 CLEANLINESS-FACTOR DATA, CONDENSER *A* (SINGLE GROUP)

Inlet-water temperature, 48 F

		Water velocity, 6.9 fps								Water velocity, 4.8 fps							
Run No.		1 H.S.		2 H.S.		3 H.S.		4 H.S.		1 L.S.		2 L.S.		3 L.S.		4 L.S.	
Date		4–10–29		4–10–29		4–10–29		4–10–29		4–10–29		4–10–29		4–10–29		4–10–29	
Duration of run, hr		1.0		1.0		1.0		1.0		1.0		1.0		1.0		1.0	
Load, Btu per hr per sq ft		5850		4000		6900		9440		5940		4010		6890		9470	
Steam pressure, in. Hg abs		0.81		0.61		0.87		1.20		0.89		0.65		0.96		1.41	
Steam temperature, deg F		72.7		64.4		74.9		84.7		75.5		66.3		77.8		89.7	
Tube No.	Condition	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.
1	New	658		698		638		664		435		511		535		548	
7	New	644		658		634		662		413		509		537		552	
2	Old	525	0.81	541	0.80	504	0.79	552	0.83	379	0.89	431	0.84	451	0.84	490	0.89
3	Old	426	0.65	458	0.68	403	0.63	417	0.63	276	0.65	363	0.71	368	0.69	366	0.67
4	Old	508	0.78	504	0.74	484	0.76	513	0.77	340	0.80	413	0.81	420	0.78	449	0.82
5	Old	475	0.73	479	0.71	466	0.73	475	0.72	343	0.81	390	0.77	400	0.75	447	0.81
6	Old	501	0.77	492	0.73	501	0.79	533	0.80	349	0.82	390	0.77	410	0.77	467	0.85
8	Old	440	0.68	453	0.67	444	0.70	453	0.68	335	0.79	374	0.73	397	0.74	413	0.75
9	Old	466	0.72	462	0.68	444	0.70	466	0.70	337	0.80	377	0.74	392	0.73	418	0.76
10	Old	501	0.77	492	0.73	475	0.75	500	0.75	349	0.82	400	0.79	418	0.78	447	0.81
Average	New	651		678		636		663		424		510		536		550	
Average	Old	480	0.74	485	0.72	465	0.73	489	0.74	339	0.80	392	0.77	407	0.76	437	0.79
		(Average cleanliness factor, high speed, 0.73)								(Average cleanliness factor, low speed, 0.78)							

TABLE 2 CLEANLINESS-FACTOR DATA, CONDENSER *A* (MULTIPLE GROUP)

Inlet-water temperature, 53 F

		Water velocity, 6.9 fps						Water velocity, 4.8 fps					
Run No.		2		4		8		3		5		6		7	
Date		11–25–30		11–25–30		11–25–30		11–25–30		11–25–30		11–25–30		11–25–30	
Duration of run, hr		1.5		1.5		1.5		1.5		1.5		0.75		1.5	
Load, Btu per hr per sq ft		7310		4380		6120		7380		4360		2810		6150	
Steam pressure, in. Hg abs		1.13		0.74		0.97		1.24		0.77		0.61		1.04	
Steam temperature, deg F		82.9		70.0		78.2		85.9		71.0		64.4		80.4	
Tube No.	Condition	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.
A1	New	631		542		598		534		495		462		523	
A2	Old	377	0.65	329	0.61	360	0.60	371	0.70	326	0.66	308	0.67	330	0.63
A3	Old	424	0.67	378	0.70	410	0.69	375	0.70	357	0.72	336	0.73	370	0.71
A4	Old	441	0.70	...	....	425	0.71	389	0.73	...	....	...	....	409	0.78
B1	New	771		699		753		629		615		563		628	
B2	Old	439	0.57	412	0.59	429	0.57	400	0.64	389	0.63	364	0.65	398	0.63
B3	Old	369	0.48	346	0.50	359	0.48	339	0.54	335	0.55	302	0.54	339	0.54
B4	Old	455	0.59	399	0.57	440	0.58	398	0.63	370	0.60	329	0.59	394	0.63
C1	New	786		683		761		636		611		555		626	
C2	Old	480	0.61	430	0.62	465	0.61	417	0.66	401	0.66	370	0.67	409	0.65
C3	Old	444	0.57	413	0.60	435	0.57	400	0.63	385	0.63	355	0.64	394	0.63
C4	Old	506	0.65	451	0.65	494	0.65	441	0.69	413	0.68	377	0.68	430	0.69
D1	New	578		503		554		500		457		417		486	
D2	Old	356	0.62	303	0.60	336	0.61	328	0.66	298	0.65	274	0.66	312	0.64
D3	Old	329	0.57	282	0.56	316	0.57	305	0.61	282	0.62	252	0.60	294	0.61
D4	Old	480	0.83	418	0.83	466	0.84	424	0.85	392	0.86	359	0.86	416	0.86
E1	New	559		468		521		488		423		398		461	
E2	Old	301	0.54	269	0.58	289	0.55	283	0.58	258	0.61	253	0.64	271	0.59
E3	Old	329	0.59	300	0.64	316	0.61	302	0.62	286	0.68	276	0.69	295	0.64
E4	Old	338	0.61	304	0.65	330	0.63	311	0.64	291	0.69	258	0.65	307	0.67
F1	New	512		439		483		449		406		368		434	
F2	Old	291	0.57	298	0.68	272	0.56	274	0.61	287	0.71	236	0.64	262	0.60
F3	Old	295	0.58	238	0.54	278	0.58	277	0.62	247	0.61	216	0.59	266	0.61
F4	Old	327	0.64	323	0.74	318	0.66	300	0.67	293	0.72	241	0.66	291	0.67
Average	New	639		556		612		539		501		461		526	
Average	Old	388	0.61	347	0.62	374	0.61	352	0.65	330	0.66	300	0.65	344	0.65
		(Average cleanliness factor, high speed, 0.61)						(Average cleanliness factor, low speed, 0.65)					

TABLE 3 CLEANLINESS-FACTOR DATA, CONDENSER *B*

Inlet-water temperature, 54 F

		Water velocity, 7.1 fps								Water velocity, 4.9 fps							
Run No.		2		4		6		7		3		5		8		9	
Date		5-11-31		5-11-31		5-12-31		5-12-31		5-11-31		5-12-31		5-12-31		5-12-31	
Duration of run, hr		1.5		1.5		1.5		1.5		1.5		1.5		1.5		1.5	
Load, Btu per hr per sq ft		6730		10180		3670		5250		6700		3680		5260		2660	
Steam pressure, in. Hg abs.		0.93		1.28		0.67		0.79		1.02		0.71		0.86		0.62	
Steam temperature, deg F		76.9		86.6		67.0		72.0		79.7		68.7		74.6		64.7	
Tube No.	Condition	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.
A1	New	574		620		611		604		522		521		531		545	
A2	Old	530	0.92	596	0.96	515	0.84	526	0.87	482	0.92	451	0.87	456	0.86	428	0.79
A3	Old	547	0.95	596	0.96	519	0.85	530	0.88	487	0.93	446	0.86	469	0.88	445	0.82
A4	Old	521	0.91	551	0.89	510	0.83	513	0.85	453	0.87	440	0.85	447	0.84	472	0.87
B1	New	800		820		762		777		650		623		643		604	
B2	Old	407	0.51	424	0.52	397	0.52	399	0.51	355	0.55	339	0.55	353	0.55	334	0.55
B3	Old	402	0.50	426	0.52	381	0.50	389	0.50	364	0.56	345	0.55	368	0.57	375	0.62
B4	Old	419	0.52	435	0.53	405	0.53	402	0.52	365	0.56	339	0.54	354	0.55	339	0.56
C1	New	798		884		856		831		689		686		689		723	
C2	Old	434	0.54	463	0.52	475	0.56	485	0.58	392	0.57	413	0.60	432	0.63	483	0.67
C3	Old	439	0.55	491	0.56	510	0.60	509	0.61	424	0.62	454	0.66	481	0.70	493	0.68
C4	Old	489	0.61	522	0.59	496	0.58	511	0.62	451	0.65	483	0.70	436	0.63	432	0.60
D1	New	601		652		648		638		475		551		548		563	
D2	Old	390	0.65	411	0.63	374	0.58	369	0.58	347	0.73	330	0.60	328	0.60	343	0.61
D3	Old	402	0.67	435	0.67	409	0.63	405	0.64	370	0.78	357	0.65	354	0.65	356	0.63
D4	Old	383	0.64	414	0.64	378	0.58	373	0.59	359	0.76	353	0.64	358	0.65	443	0.79
E1	New	377		412		409		383		354		367		359		363	
E2	Old	384	1.02	429	1.04	397	0.97	387	1.01	358	1.01	351	0.96	364	1.01	347	0.96
E3	Old	375	0.99	411	1.00	387	0.95	371	0.97	351	0.99	345	0.94	350	0.98	352	0.97
E4	Old	363	0.96	397	0.96	387	0.95	359	0.94	338	0.96	334	0.91	337	0.94	348	0.96
F1	New	440		476		498		470		497		432		419		425	
F2	Old	313	0.71	335	0.70	342	0.69	321	0.68	296	0.75	304	0.70	295	0.70	296	0.70
F3	Old	317	0.72	340	0.71	325	0.65	312	0.67	303	0.76	301	0.70	302	0.72	301	0.71
F4	Old	303	0.69	326	0.69	320	0.64	305	0.65	296	0.75	301	0.70	296	0.71	287	0.68
Average	New	598		644		631		617		515		530		532		537	
Average	Old	412	0.69	445	0.69	418	0.66	415	0.67	377	0.73	371	0.70	377	0.71	382	0.71
		(Average cleanliness factor, high speed, 0.68)								(Average cleanliness factor, low speed, 0.71)							

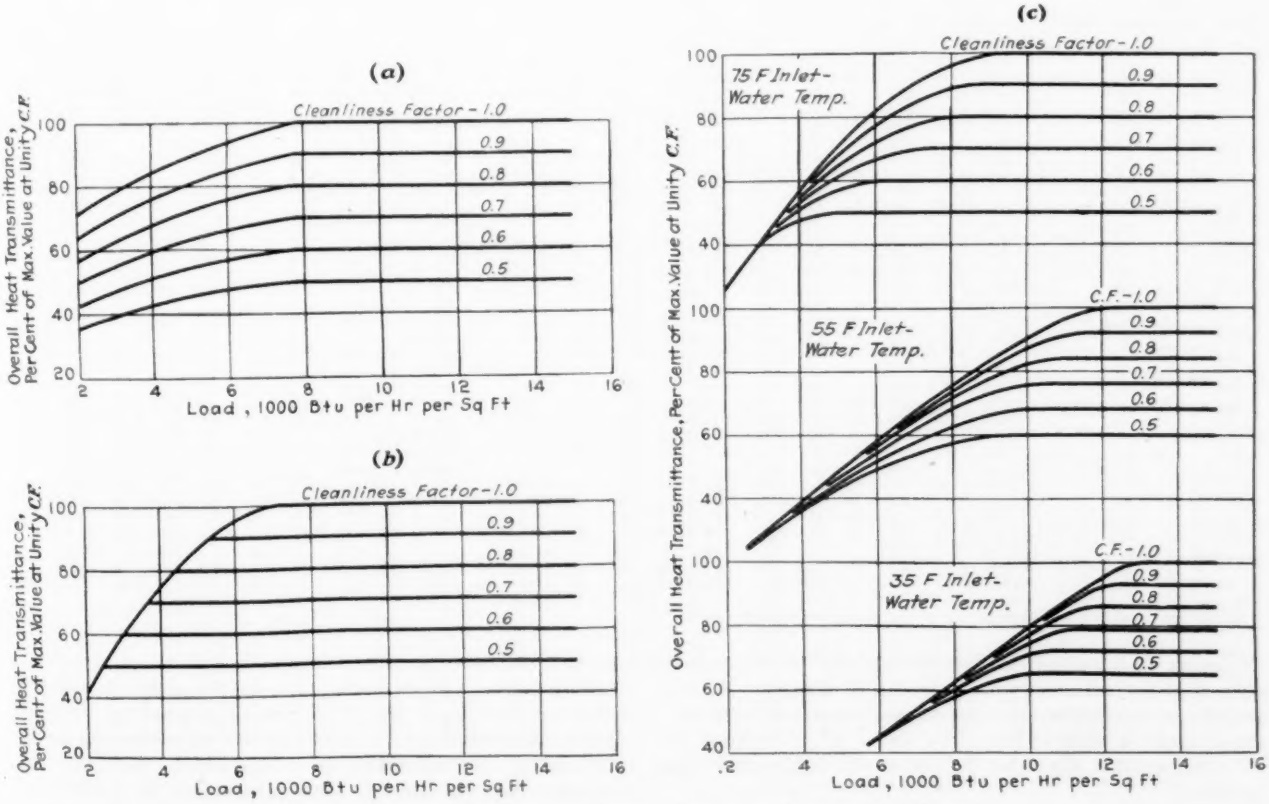


FIG. 7 TYPICAL PERFORMANCE CURVES, INCLUDING CLEANLINESS FACTOR

TABLE 4 CLEANLINESS-FACTOR DATA, CONDENSER *C*

Inlet-water temperature, 63 F

		Water velocity, 7.2 fps									
Run No.		1		2		3		4		5	
Date		6-3-30		6-3-30		6-3-30		6-3-30		6-4-30	
Duration of run, hr.		1.5		1.0		1.5		1.5		1.0	
Load, Btu per hr per sq ft		11590		11750		8940		7400		5630	
Steam pressure, in. Hg abs		1.99		2.19		1.76		1.42		1.20	
Steam temperature, deg F		101.0		104.3		97.0		90.1		84.6	
Tube No.	Condition	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.
1	New	817		806		743		726		719	
2	Old	487	0.60	482	0.60	435	0.59	423	0.58	414	0.58
6	New	848		824		755		719		696	
5	Old	435	0.51	431	0.52	392	0.52	378	0.53	375	0.54
7	New	841		837		778		752		729	
8	Old	420	0.50	410	0.49	385	0.50	371	0.49	376	0.52
10	New	717		699		639		617		564	
9	Old	363	0.51	357	0.51	319	0.50	323	0.52	304	0.54
12	New	717		698		645		623		593	
11	Old	288	0.40	297	0.43	258	0.40	250	0.40	237	0.40
14	New	688		662		613		591		550	
13	Old	391	0.57	374	0.57	343	0.56	329	0.56	324	0.59
15	New	719		677		646		612		537	
16	Old	396	0.55	385	0.57	350	0.54	325	0.53	305	0.57
18	New	600		540		590		454		238	
17	Old	367	0.61	340	0.63	348	0.59	268	0.59	118	0.49
19	New	683		653		592		551		12	
20	Old	438	0.64	436	0.67	398	0.67	379	0.69	30	....
Average	New	737		711		667		628		515	
Average	Old	398	0.54	390	0.55	359	0.54	338	0.54	276	0.54

(Average cleanliness factor, 0.54)

TABLE 5 CLEANLINESS-FACTOR DATA, CONDENSER *D*

Inlet-water temperature, 62 F

		Water velocity, 8.3 fps										Water velocity, 6.0 fps											
Run No.		1		5		6		9		10		2		3		4		7		8		11	
Date		6-13-32		6-13-32		6-14-32		6-14-32		6-14-32		6-13-32		6-13-32		6-13-32		6-14-32		6-14-32		6-14-32	
Duration of run, hr		1.0		1.0		1.0		1.0		1.0		1.0		1.0		1.0		1.0		1.0		1.0	
Load, Btu per hr per sq ft		6770		5260		9140		12690		6750		6680		4120		5210		9120		12730		6780	
Steam pressure, in. Hg abs		1.09		0.93		1.36		1.82		1.09		1.20		0.89		1.01		1.57		2.13		1.21	
Steam temperature, deg F		81.7		76.7		88.6		98.0		81.7		84.6		75.6		79.5		93.1		103.4		84.9	
Tube No.	Condition	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.
B1	New	837		796		858		853		844		698		669		714		728		732		733	
B2	New	877	1.00	825	1.00	882	1.00	870	1.00	867	1.00	721	1.00	682	1.00	729	1.00	734	1.00	740	1.00	756	1.00
B3	Old	611	0.70	583	0.71	547	0.62	531	0.61	518	0.60	532	0.74	503	0.74	526	0.72	484	0.66	468	0.63	466	0.62
B4	Old	590	0.67	576	0.70	554	0.63	545	0.63	541	0.62	515	0.71	496	0.73	506	0.69	497	0.68	487	0.66	486	0.64
B5	Old	567	0.65	546	0.66	518	0.59	509	0.59	516	0.60	495	0.69	483	0.71	490	0.67	460	0.63	454	0.61	460	0.61
C1	New	645		602		572		585		610		529		556		550		520		538		552	
C2	New	616	1.00	615	1.00	612	1.00	623	1.00	625	1.00	533	1.00	553	1.00	536	1.00	546	1.00	561	1.00	555	1.00
C3	Old	445	0.72	416	0.68	407	0.67	394	0.63	378	0.60	402	0.75	394	0.71	397	0.74	378	0.69	372	0.66	351	0.63
C4	Old	456	0.74	433	0.70	410	0.67	391	0.63	388	0.62	407	0.76	413	0.75	406	0.76	372	0.68	365	0.65	356	0.64
C5	Old	420	0.68	395	0.64	385	0.63	374	0.60	361	0.58	388	0.73	384	0.69	383	0.71	365	0.67	357	0.64	344	0.62
D1	New	914		871		911		904		891		738		744		757		757		764		786	
D2	New	869	1.00	826	1.00	881	1.00	890	1.00	856	1.00	714	1.00	693	1.00	711	1.00	727	1.00	749	1.00	745	1.00
D3	Old	537	0.62	496	0.60	461	0.52	440	0.49	434	0.51	465	0.65	445	0.64	454	0.64	405	0.56	391	0.52	382	0.51
D4	Old	562	0.65	539	0.65	484	0.55	455	0.51	450	0.53	492	0.69	464	0.67	485	0.68	424	0.58	407	0.54	400	0.54
D5	Old	589	0.68	568	0.69	500	0.57	475	0.53	466	0.54	500	0.70	492	0.71	501	0.70	440	0.61	419	0.56	411	0.55
F1	New	771		708		786		786		761		631		582		634		680		686		678	
F2	New	772	1.00	725	1.00	795	1.00	804	1.00	762	1.00	648	1.00	600	1.00	638	1.00	677	1.00	702	1.00	671	1.00
F3	Old	523	0.68	483	0.67	482	0.61	460	0.57	431	0.57	465	0.72	428	0.71	447	0.70	426	0.63	423	0.60	406	0.61
F4	Old	455	0.59	418	0.58	435	0.55	426	0.53	372	0.49	414	0.64	387	0.65	398	0.62	386	0.57	389	0.55	354	0.53
F5	Old	469	0.61	430	0.59	460	0.58	457	0.57	402	0.53	416	0.64	393	0.66	407	0.64	411	0.61	417	0.60	375	0.56
G1	New	660		621		666		670		650		560		559		582		594		604		597	
G2	New	651	1.00	618	1.00	671	1.00	679	1.00	645	1.00	559	1.00	532	1.00	556	1.00	588	1.00	603	1.00	588	1.00
G3	Old	458	0.70	422	0.68	415	0.62	408	0.60	389	0.60	410	0.73	394	0.74	406	0.73	381	0.65	381	0.63	366	0.62
G4	Old	476	0.73	458	0.74	420	0.63	407	0.60	408	0.63	417	0.75	422	0.79	426	0.77	389	0.66	381	0.63	371	0.63
G5	Old	462	0.71	422	0.68	421	0.63	412	0.61	389	0.60	416	0.74	387	0.73	400	0.72	389	0.66	383	0.64	368	0.63
K1	New	541		500		512		566		507		455		480		468		477		520		470	
K2	New	533	1.00	485	1.00	567	1.00	582	1.00	530	1.00	472	1.00	453	1.00	460	1.00	516	1.00	534	1.00	495	1.00
K3	Old	381	0.71	368	0.76	354	0.62	366	0.63	338	0.64	349	0.74	364	0.80	353	0.77	332	0.64	342	0.64	320	0.65
K4	Old	437	0.82	400	0.82	416	0.73	411	0.71	388	0.73	397	0.84	386	0.85	387	0.84	380	0.74	383	0.72	358	0.72
K5	Old	414	0.78	388	0.80	394	0.69	385	0.66	371	0.70	369	0.78	366	0.81	371	0.81	364	0.71	365	0.68	316	0.64
Average	New (1)	728		683		718		727		711		602		598		618		626		640		636	
	New (2)	720	1.00	682	1.00	735	1.00	741	1.00	714	1.00	608	1.00	586	1.00	605	1.00	631	1.00	648	1.00	635	1.00
	Old	492	0.68	463	0.68	448	0.61	436	0.59	419	0.59	436	0.72	422	0.72	430	0.71	405	0.64	399	0.62	383	0.60

NOTE: No. 1 tubes supplied with city water, all others with salt water.

circulating water and a cleanliness factor of 0.5 is shown to be 60 per cent of the maximum value instead of 50 per cent. At light loads the relation of the curves for different cleanliness factors is a sort of mean of those in Figs. 7 (*a*) and 7 (*b*). The lower the water temperature, the higher the load at which the curves begin to drop off.

The loading undoubtedly has an effect upon the amount of reduction in overall heat transmittance due to a given degree of fouling, and possibly temperature has some effect also, but how much effect either will have for a particular condenser is difficult to predict. Six of the leading condenser manufacturers have adopted standard commercial factors,[4] and these include a "loading curve," which is the same as the unity cleanliness-factor curve in Fig. 7 (*a*). This curve is assumed by the condenser manufacturers to be applicable for all makes and to be independent of circulating-water temperature. With this assumption there is probably little justification for a different shaped loading curve for each cleanliness factor. Then, only the unity cleanliness-factor curve is necessary, and the new-tube overall heat transmittance multiplied by the cleanliness factor will give the value for the fouled-tube condition, or, vice versa,

[4] See article in *Power*, September, 1932, p. 133.

TABLE 6 CLEANLINESS-FACTOR DATA, CONDENSER *E*

Inlet-water temperature, 69 F

		Water velocity, 8.9 fps									
Run No.		1		5		6		9		10	
Date		6-27-32		6-27-32		6-28-32		6-28-32		6-28-32	
Duration of run, hr.		1.0		1.0		1.0		1.0		1.0	
Load, Btu per hr per sq ft		7180		6020		9320		12500		7140	
Steam pressure, in. Hg abs		1.23		1.10		1.46		1.80		1.22	
Steam temperature, deg F		85.4		81.9		90.9		97.7		85.1	
Tube No.	Condition	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.
B1	New	898		878		908		919		906	
B2	New	813	1.00	793	1.00	841	1.00	850	1.00	806	1.00
B3	Old	675	0.83	632	0.80	616	0.73	614	0.72	628	0.78
B4	Old	663	0.82	650	0.82	628	0.75	636	0.75	631	0.78
B5	Old	564	0.69	540	0.68	570	0.68	583	0.69	574	0.71
D1	New	890		878		931		946		942	
D2	New	953	1.00	913	1.00	984	1.00	987	1.00	1002	1.00
D3	Old	680	0.71	652	0.71	654	0.66	664	0.67	658	0.66
D4	Old	707	0.74	690	0.76	678	0.69	684	0.69	677	0.68
D5	Old	742	0.78	730	0.80	702	0.71	706	0.72	670	0.67
E1	New	816		777		836		856		819	
E2	New	743	1.00	725	1.00	753	1.00	784	1.00	766	1.00
E3	Old	624	0.84	600	0.83	604	0.80	604	0.77	580	0.76
E4	Old	603	0.81	591	0.82	578	0.77	569	0.73	553	0.72
E5	Old	594	0.80	584	0.81	571	0.76	580	0.74	555	0.72
G1	New	712		680		734		783		721	
G2	New	783	1.00	749	1.00	776	1.00	781	1.00	765	1.00
G3	Old	564	0.72	553	0.74	546	0.70	560	0.72	539	0.70
G4	Old	598	0.76	588	0.79	563	0.73	568	0.73	554	0.72
G5	Old	617	0.79	593	0.79	580	0.75	599	0.77	581	0.76
L1	New	594		594		608		638		594	
L2	New	632	1.00	621	1.00	632	1.00	644	1.00	614	1.00
L3	Old	460	0.73	462	0.74	455	0.72	471	0.73	440	0.72
L4	Old	494	0.78	499	0.80	482	0.76	491	0.76	459	0.75
L5	Old	476	0.75	479	0.77	472	0.75	492	0.76	458	0.75
M1	New	699		700		721		741		707	
M2	New	688	1.00	690	1.00	710	1.00	727	1.00	692	1.00
M3	Old	591	0.86	594	0.86	590	0.83	603	0.83	568	0.82
M4	Old	577	0.84	584	0.85	577	0.81	585	0.80	560	0.81
M5	Old	615	0.89	613	0.89	600	0.85	609	0.84	586	0.85
Average	New (1)	768		751		790		814		782	
Average	New (2)	769	1.00	749	1.00	783	1.00	796	1.00	774	1.00
Average	Old	602	0.78	591	0.79	581	0.74	590	0.74	571	0.74

		Water velocity, 5.8 fps									
Run No.		2		3		4		7		8		11	
Date		6-27-32		6-27-32		6-27-32		6-28-32		6-28-32		6-28-32	
Duration of run, hr.		1.0		1.0		1.0		1.0		1.0		1.0	
Load, Btu per hr per sq ft		7180		4630		6020		9350		12570		7180	
Steam pressure, in. Hg abs		1.41		1.08		1.24		1.75		2.26		1.40	
Steam temperature, deg F		89.7		81.3		85.7		96.7		105.2		89.6	
Tube No.	Condition	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.	U.	C.F.
B1	New	751		729		750		743		764		768	
B2	New	660	1.00	661	1.00	656	1.00	673	1.00	709	1.00	678	1.00
B3	Old	571	0.87	567	0.86	557	0.85	526	0.78	515	0.73	539	0.80
B4	Old	567	0.86	566	0.86	564	0.86	528	0.78	540	0.76	546	0.81
B5	Old	487	0.74	476	0.72	478	0.73	486	0.72	497	0.70	492	0.73
D1	New	727		720		723		742		750		758	
D2	New	758	1.00	744	1.00	755	1.00	809	1.00	794	1.00	819	1.00
D3	Old	568	0.75	557	0.75	555	0.73	553	0.68	553	0.70	553	0.68
D4	Old	589	0.78	584	0.79	590	0.78	575	0.71	571	0.72	574	0.70
D5	Old	607	0.80	594	0.80	606	0.80	590	0.73	584	0.74	574	0.70
E1	New	654		644		662		685		710		694	
E2	New	612	1.00	611	1.00	625	1.00	640	1.00	658	1.00	651	1.00
E3	Old	522	0.85	514	0.84	526	0.84	513	0.80	513	0.78	506	0.78
E4	Old	498	0.81	516	0.85	512	0.82	493	0.77	494	0.75	493	0.76
E5	Old	501	0.82	504	0.83	515	0.82	497	0.78	490	0.74	490	0.75
G1	New	582		572		575		614		632		616	
G2	New	626	1.00	619	1.00	631	1.00	640	1.00	652	1.00	640	1.00
G3	Old	478	0.76	489	0.79	485	0.77	485	0.76	492	0.75	471	0.74
G4	Old	493	0.79	498	0.80	504	0.80	491	0.77	485	0.74	477	0.75
G5	Old	516	0.83	508	0.82	520	0.82	496	0.78	492	0.75	497	0.78
L1	New	512		540		535		534		547		540	
L2	New	525	1.00	556	1.00	547	1.00	556	1.00	550	1.00	566	1.00
L3	Old	418	0.80	422	0.76	429	0.78	431	0.78	433	0.79	415	0.73
L4	Old	442	0.84	454	0.82	448	0.82	435	0.78	428	0.78	435	0.77
L5	Old	430	0.82	451	0.81	443	0.81	432	0.78	413	0.75	433	0.77
M1	New	577		596		618		620		640		624	
M2	New	561	1.00	586	1.00	598	1.00	603	1.00	622	1.00	604	1.00
M3	Old	488	0.87	516	0.88	516	0.86	519	0.86	525	0.84	500	0.83
M4	Old	478	0.85	518	0.88	508	0.85	506	0.84	507	0.82	505	0.84
M5	Old	502	0.90	543	0.93	526	0.88	522	0.87	524	0.84	517	0.86
Average	New (1)	634		634		644		656		674		667	
Average	New (2)	624	1.00	630	1.00	635	1.00	654	1.00	664	1.00	660	1.00
Average	Old	509	0.82	515	0.82	516	0.81	504	0.77	503	0.76	501	0.76

NOTE: No. 1 tubes supplied with city water, all others with salt water.

TABLE 7 COMPARISON OF NEW TUBES FROM STOCK WITH NEWLY DRAWN TUBES

		Water velocity, 8.9 fps						Water velocity, 5.8 fps					
Load, Btu per hr per sq ft		6000	7100	7100	9200	9200	Avge.	6000	7100	7100	9200	9200	Avge.
Steam pressure, in. Hg abs		1.24	1.32	1.53	1.64	1.98		1.23	1.32	1.54	1.65	1.96	
Steam temperature, deg F		85.6	87.7	92.4	94.6	100.8		85.5	87.6	92.5	94.7	100.6	
Location	Tube from	Heat transmittance						Heat transmittance					
B2	Stock	794	782	788	825	818	801	665	661	659	683	678	669
B3	Stock	930	862	900	909	933	907	748	708	733	741	760	738
B4	Mill	908	862	894	894	922	896	741	711	732	731	752	733
D2	Stock	940	912	908	928	931	924	765	751	736	762	757	754
D3	Stock	917	886	884	920	906	903	742	729	720	752	737	736
D4	Mill	920	886	891	912	910	904	742	736	707	746	744	735
E2	Stock	768	772	763	766	765	767	629	644	632	638	628	634
E3	Stock	822	816	826	855	851	834	656	681	676	702	704	684
E4	Mill	744	748	749	758	754	751	612	624	624	630	629	624
G2	Stock	731	724	729	746	753	737	615	615	606	626	634	619
G3	Stock	616	614	617	635	646	626	529	536	533	552	560	542
G4	Mill	663	666	666	682	678	671	562	568	561	572	580	569
L2	Stock	680	662	664	649	667	664	569	566	564	552	569	564
L3	Stock	621	608	619	625	640	623	540	538	541	552	558	546
L4	Mill	697	674	681	670	680	680	592	580	580	584	586	584
M2	Stock	722	712	717	716	728	719	592	594	600	608	608	600
M3	Stock	691	702	706	720	734	711	581	596	598	611	614	600
M4	Mill	752	737	748	742	754	747	625	628	632	626	636	629
Avge. Stock (2)		773	761	762	772	777	769	639	639	633	645	646	640
Avge. Stock (3)		766	748	759	777	785	767	633	631	634	652	656	641
Avge. Mill (4)		781	762	772	777	783	775	646	641	639	648	655	646

the fouled-tube overall heat transmittance divided by the cleanliness factor will give the value for the new tube condition.

Two convenient equations for computing the absolute pressure when guarantees are given on the heat-transmittance basis are as follows:

$$t_s = t_1 + \frac{Q}{WC\,(1 - e^{-x})} \quad \ldots\ldots\ldots\ldots\ldots [5]$$

$$t_s = t_1 + \frac{tr}{1 - e^{-x}} \quad \ldots\ldots\ldots\ldots\ldots [6]$$

$$x = \frac{US}{WC}$$

where

U = heat transmittance, Btu per hr per sq ft per deg F log m.t.d.
Q = heat transferred, Btu per hr
S = area of condensing surface, sq ft
W = quantity of circulating water, lb per hr
C = specific heat of the circulating water
e = Napierian base (2.718)
t_1 = inlet temperature of circulating water, deg F
tr = temperature rise of circulating water, deg F (this will be the same for the guarantee as for the test)
t_s = saturation temperature in deg F, from which the corresponding absolute pressure at the top of the condenser can be obtained from the steam tables.

New Tubes

New tubes received from the mills are not absolutely uniform as regards their performance, and this fact accounts for some of the apparent inconsistencies in the data obtained. Where several new tubes (one for each group) are used, however, their average has been found to be a reliable measure of new-tube performance for the conditions under which they were operating. Since the overall cleanliness factor is based on the average performance of the old and new test tubes, the accuracy of the measurement is not impaired by inequalities in the new tubes so long as a sufficient number is used to give a reliable average.

Tubes left over when a condenser is tubed are preferable ones to use for new tubes during the acceptance test. If these tubes are carefully stored, there will be little or no depreciation in their performance. The data reported in Table 7 give the results of tests made on condenser *E* to determine the comparison of newly drawn tubes with tubes that had been stored from the time the condenser was installed. The No. 2 tubes in all groups were the stock tubes used during the test reported in Table 6. The No. 3 tubes were from stock also, but appeared to be not as well taken care of as the No. 2 tubes. The No. 4 tubes were drawn less than a week prior to the test. As can be seen, the difference between the average for the newly drawn and the stock tubes is so small as to be negligible.

Tubes in the same condition as when received from the manufacturer are commercially new, and no attempt should be made to improve their performance prior to using them on a test. Some methods of cleaning are as likely to reduce the performance of a new tube as to increase it.

TABLE 8 VARIATION OF HEAT TRANSMITTANCE AND CLEANLINESS FACTOR WITH WATER VELOCITY

Load, 8500 Btu per hr per sq ft; inlet-water temperature, 55 F; steam pressure 1.18 in. Hg abs

		Water velocity, ft per sec					
		8.9	8.1	7.1	6.0	4.9	
		Heat transmittance, Btu per hr per sq ft per deg F m.t.d.					
Tube No.	Condition						Velocity exponent
A1	New	760	709	664	626	541	0.50
A2	Old	644	612	582	549	489	0.40
A3	Old	670	636	610	565	499	0.43
A4	Old	616	585	556	527	463	0.40
B1	New	903	858	809	752	648	0.48
B2	Old	442	431	417	393	361	0.31
B3	Old	459	440	429	406	369	0.32
B4	Old	408	399	393	359	351	0.29
C1	New	914	858	816	755	654	0.51
C2	Old	503	488	462	439	396	0.36
C3	Old	537	514	484	464	413	0.38
C4	Old	...	...	...	...	...	..
D1	New	784	732	685	636	568	0.54
D2	Old	561	544	523	492	430	0.34
D3	Old	530	582	559	524	442	0.30
D4	Old	409	397	391	376	345	0.20
E1	New	514	493	468	443	395	0.40
E2	Old	508	483	467	435	389	0.38
E3	Old	503	479	461	435	394	0.38
E4	Old	470	444	429	403	369	0.37
F1	New	451	517	494	466	429	0.35
F2	Old	359	338	329	315	295	0.35
F3	Old	392	361	353	330	314	0.42
F4	Old	340	323	315	296	290	0.34
Average	New	721	695	656	613	539	0.46
Average	Old	491	474	456	430	389	0.35
Cleanliness factor		0.68	0.68	0.69	0.70	0.72	—0.11[a]

[a] Since the cleanliness factor varies inversely as some power of the velocity, the exponent is negative.

Effect of Water Velocity

It will be noticed from the data reported that the cleanliness factor is different for different water velocities through the tubes. This is as it should be according to theory, and is explained by the fact that the heat transmittance varies with some power of the velocity and that the exponents for new tubes are higher than for old tubes. Since the cleanliness factor is the ratio of the heat transmittance of old to new tubes, it is readily seen that the cleanliness factor will vary with a power of the velocity equal to the exponent for the old tubes minus the exponent for the new tubes. This will give a negative exponent, which means that the cleanliness factor decreases with increasing velocity. Table 8 shows test-tube data obtained on condenser B for a wide range of water velocities, and the variation in cleanliness factor is shown graphically by the solid curve in Fig. 8. The dotted curves were plotted from calorimeter measurements on individual tubes from other condensers. The slopes of the cleanli-

ness-factor curves are not great, and therefore the velocity through the test tubes can differ slightly from the velocity for the entire condenser without seriously affecting the accuracy of the measurement. If tests are made at two or more velocities, the cleanliness factor can be corrected for velocity, as explained under "Methods of Computation."

Progressive Fouling

Some condensers maintain an approximately constant cleanliness factor for a day or two after cleaning, while others foul rapidly. If there is no apparent decrease in the cleanliness factor during the period of test, the cleanliness factors for all runs at the same speed can be averaged, thereby increasing the accuracy of this measurement. Where the cleanliness factor decreases rapidly, it is necessary to apply the measured value for each run. Fig. 9 shows the variation in the cleanliness factor with time for the test on condenser *E*. There was a sufficient number of runs in this particular case to draw a curve which fairs the points. Values taken from this curve were considered to be more nearly correct than the individual points.

When the runs are made in the descending order of loads, a slight decrease in the cleanliness factor may be due to reduced pressure drop, where the pressure at the top has been used for the

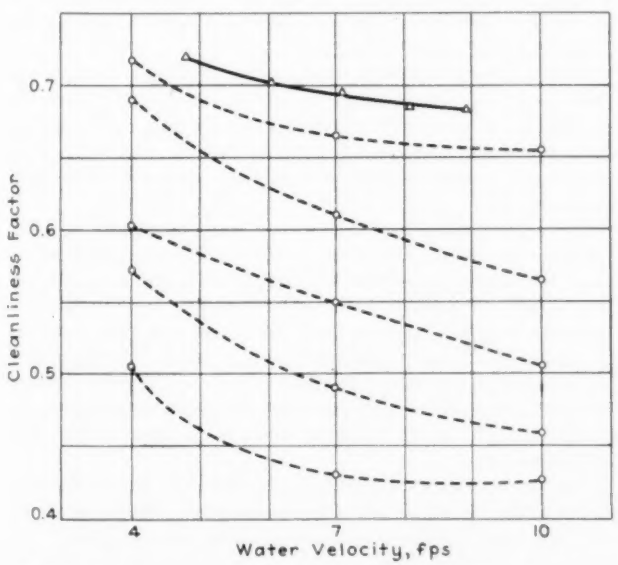


Fig. 8 Variation of the Cleanliness Factor With Velocity of Water Through the Tubes

test-tube computations, or it might be due to the inaccuracies inherent with light-load measurements. In this case those responsible for the test must decide whether an average value or the individual values are to be used.

With the method of operating test tubes described under "Test Procedure," it is assumed that the old tubes will foul at the same rate as the remainder of the condenser. The shorter the period over which it is necessary to apply this assumption, the more reliable the results will be. Every effort should be made, therefore, to start testing as soon as possible after the unit is returned to service, following the installation of test equipment, and to complete all runs as quickly as possible.

If the new tubes are operated only during the periods when tests are in progress, they are not likely to foul perceptibly before all runs are completed. Repeating the first run at the end will provide a definite check upon this condition.

Chlorination

The sterilization of circulating water with chlorine is often assumed to be so effective in eliminating fouling that the condenser maintains its new-tube performance. If this were true, it would be unnecessary to include the cleanliness-factor measurement in tests on chlorinated condensers. Experiments with chlorine on isolated test tubes in condensers *D* and *E*, however, indicate that the cleanliness factor of a chlorinated condenser is

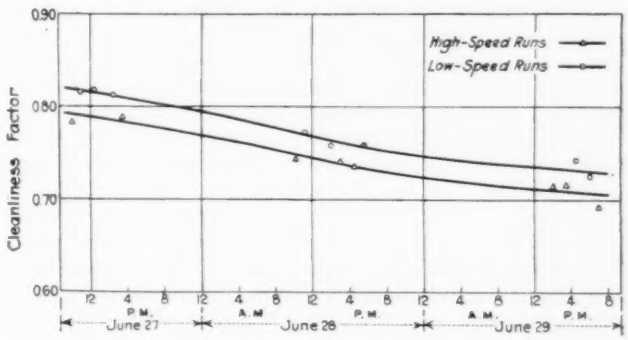


Fig. 9 Progressive Fouling During Test on Condenser *E*

probably between 0.90 and 0.95. Even these values may be optimistic, because the rate of chlorine injection during the experiments was higher than would be economically justifiable. Both old and new tubes were tested for a sufficient period for the performance of each to become constant. The tubes gave different cleanliness factors with the same dosage and frequency of application. Few of the old tubes attained as high cleanliness factors as those to which the new tubes decreased.

Conclusions

The cleanliness-factor measurement has proved its practicability and has eliminated the uncertainty of fouling in appraising the performance of the condensers on which it was applied. By eliminating the necessity for unusual methods of cleaning prior to the acceptance tests, the overall test performance was indicative of normal operation.

Those who have been dubious of, or hostile to, the test-in-place method have objected on the ground that tubes tested in the center and bottom of a condenser will give erroneous values for their cleanliness factor. In practise this has not been found to be the case. The fact that there is a greater percentage of air in the steam that flows over these tubes apparently does not affect the ratio of the heat transmittance of a fouled tube and its adjacent new tube, even though the heat transmittance of both tubes is greatly reduced. It is preferable, therefore, to test the tubes in place rather than moving them to the top of the condenser or to a calorimeter.

Acknowledgments

Mr. J. N. Landis, mechanical engineer of the Brooklyn Edison Company, Inc., was responsible for obtaining the condenser guarantee based on a cleanliness factor to be determined from isolated tube measurements made at the time of the acceptance test. This purchasing procedure led to the development of the method of test described.

The authors also wish to acknowledge the cooperation of the following manufacturers and their staffs in the tests described in this paper: Westinghouse Elec. & Mfg. Co.; Worthington Pump and Machinery Corporation; Ingersoll-Rand Company.

Discussion

D. W. R. Morgan.[5] This collection of data points the way to a measurement of condenser performance and is valuable in that it may be used as an instrument by the designer to further improvements in condenser design.

The authors state that it is just as important to determine the cleanliness factor as it is to determine circulating-water temperature and water velocity through the tube. The writer thoroughly agrees with the conclusion, but in his opinion the thermometers generally used and the means for determining the temperature rise are inadequate, particularly for determining the performance of single-pass condensers where large quantities of water are circulated and where the temperature rise is relatively small.

One may infer that it is the intention of the Power Test Code Committee to incorporate in the condenser test code the means of testing described by the authors. The writer agrees that similar data should be included in the test code; he does not think the method proposed is sufficiently accurate for the purpose. For example, certain tubes have been selected as representing the performance throughout the condenser. This assumption may or may not hold for all installations, and as a matter of fact, serious consideration should be given to the use of weighted averages of performance throughout the condenser rather than the designation of a relatively few tubes from which the performance is determined.

The selection of the tubes for test introduces a real problem, as the performance of the condenser is not only dependent upon the design of the condenser itself, but may be adversely affected by the distribution of steam issuing from the turbine exhaust. There are records of condenser performance from one station in the New York territory indicating that the performance of the condenser installed in 1914 was superior to that of the condenser installed in 1925. An examination of the data clearly indicates that the good performance of the earlier designed condenser can be attributed entirely to the uniform distribution of steam from the turbine exhaust throughout the tube length of the condenser, whereas in the more recent design the tube length was greatly in excess of the turbine exhaust opening, resulting in a non-uniform distribution of steam, carrying with it a relatively poor condenser performance. Progress in condenser design will result only through thorough study and investigation and the introduction of such elements as are necessary to secure uniform distribution of steam to the tube nest of the condenser.

The writer agrees with the authors that it is absurd to consider the removal of tubes from the condenser for test purposes for the reasons set forth by the authors. On the other hand, the choice of tubes for test must be such that the steam distribution be uniform to insure loading of the tube for its entire length. The authors state that tubes located near the top of the condenser have a greater effect on performance than those near the air offtake. There are many tests on record which prove conclusively that the tubes in the top row are not necessarily the most effective. This depends largely on the distribution of steam from the turbine exhaust, and on the other hand, by the design of the condenser itself. Further, tubes near the offtake perform just as important a function as the tubes near the outer periphery, and this should be recognized in determining the performance of the entire tube nest.

The authors describe a distributing manifold arrangement at the inlet end of the condenser. It would be well for the authors to keep in mind the possibility of error due to an accumulation of air in the manifold. It is stated that the water for the test tubes is taken from the same source as the main condenser. The authors show that the water is taken from the inlet-water box. While it may be taken from the same source, if it passes through the house service pump, there is every possibility that it will contain less debris than if introduced through the main circulating pump.

The authors suggest that in order to produce uniformity of fouling, the condenser tubes should be cleaned prior to the test. We have several instances in the testing of large condensers where the heat-transfer rates were reduced following cleaning such as proposed by the authors. In other words, the deposit on the tubes originally was non-uniform, and when followed by cleaning the tubes, it resulted in a more uniform distribution of the deposit on the tube, which actually reduced the effectiveness of the tube. Therefore this suggestion depends largely upon the nature of the debris and upon the nature of the deposit upon the tube plate itself.

The single-group arrangement would lead to an erroneous idea of the overall results, as readings would not represent the general averages.

The authors describe a method for determining the fouling of tubes through which city water is circulated. They, however, fail to include the amount of fouling that occurred in the tubes through which city water was circulated. No doubt a certain amount of fouling occurred, which demonstrates that it is impractical to use this method for comparison and determining the rapidity and amount of fouling.

What one really desires to obtain on condenser tests is the actual result, and the writer does not regard the data presented as sufficient to permit of such definite conclusions, as reached by the authors, and neither does he agree that the effect of fouling is the same on tubes that are required for cooling non-condensible vapors.

The authors suggest that the tests be made rapidly so as to minimize the effect of the fouling of the new tubes. This statement in itself suggests that the method proposed is unsatisfactory. It is also suggested that on tests of long duration the new tubes be plugged to preserve their cleanliness. This arrangement is unsatisfactory, as it produces the equivalent of a baking and cracking process used so successfully in removing scale and debris from evaporator tubes, and therefore, depending upon the nature of the deposit, would in some instances seriously affect the overall results.

The assumption as to pressure drop from the top of the condenser to the air offtake would depend largely on the design of the condenser and whether or not external air coolers were used.

Authors' Closure

Mr. Morgan's contentions regarding the accuracy of this method seem to be based in general on the fact that little is known about what occurs inside the steam space of condensers. The authors do not consider that this knowledge is necessary. The test tubes are not selected for the purpose of determining the average heat transmittance of the condenser, but rather the degree of fouling, which varies much less than the heat transmittance and is affected very little, if any, by steam distribution. The overall heat transmittance must still be determined the same as usual. The true accuracy of any test measurement is very difficult of proof. However, the authors hope to present in the future the results of an investigation to be conducted for the specific purpose of definitely determining the accuracy of the cleanliness-factor measurement.

Mr. Morgan's statement that in several instances the heat transmittance of condensers was known to be reduced by cleaning prior to the test seems to illustrate quite forcefully the necessity for a cleanliness-factor measurement so that such unusual conditions can be evaluated. As stated in the paper, the only pur-

[5] Manager, Condenser and Pump Division, Westinghouse Elec. & Mfg. Co. South Philadelphia, Pa. Assoc-Mem. A.S.M.E.

pose in cleaning just prior to the test is to make the fouling of all tubes as uniform as possible, since a comparison of test and guarantee performances can be made as well at a cleanliness factor of 0.50 as 0.90. Other relevant points raised by Mr. Morgan are fully covered in the paper.

Mr. J. F. Grace[6] has asked the cost of making the cleanliness-factor measurement. This is hard to answer because of the difference in test facilities and method of computing the cost of tests at different plants. In general, however, the labor cost of the condenser test is doubled when the cleanliness-factor measurement is included, and $800 to $1000 is required for additional equipment when the test is performed for the first time.

If this method should come into more general use, the condenser manufacturers would probably maintain the special equipment in order to have it available for acceptance tests in those plants where the cost of the equipment would make the method prohibitive.

[6] Condenser Engineer, Worthington Pump and Machinery Corporation, Harrison, N. J. Mem. A.S.M.E.

RP-55-4

The Pressure of Saturated Water Vapor in the Range 100 C to 374 C[1]

By N. S. OSBORNE,[2] H. F. STIMSON,[2] E. F. FIOCK,[2] and D. C. GINNINGS,[3] WASHINGTON, D. C.

The method and apparatus employed, and the measurements made in this determination of the vapor pressure of water are described, and the results obtained are given and formulated.

The method used was the "static method" in which the pressure is determined at the stationary boundary between the liquid and vapor phase in equilibrium at a constant temperature in a closed container.

The container for the water sample was a calorimeter specially built for high-pressure work. Pressures were measured by a precision piston gage which balances the pressure by the gravitational force on a piston loaded with weights. Temperatures were measured by platinum-resistance thermometers supplemented by thermoelements.

A total of 394 measurements were made at 38 temperatures so distributed as to facilitate formulation of the pressure-temperature relation.

The formulation was made by using an empirical equation which was fitted to the results in two temperature ranges. Values of pressure and its derivative with respect to temperature are tabulated from this formulation in standard atmospheres, centibars, and kilograms per square centimeter at each degree centigrade, and in pounds per square inch at each degree fahrenheit. This provides a mutually consistent group of vapor-pressure tables in convenient form for practical working tables of fundamental steam data.

It is estimated that the results are reliable to 3 parts in 10,000.

I—INTRODUCTION

THE relation between the temperature and the pressure of saturated water vapor is of prime importance in establishing the thermodynamic behavior of steam. This relation, commonly called the vapor-pressure relation, is a characteristic property which can be observed directly. Moreover, its derivative is an essential factor in correlating other important thermodynamic properties. It is evident that adequate knowledge of this property is indispensable to the success of any systematic effort to determine and formulate the thermal properties of steam.

The vapor-pressure measurements to be described here were made as a coordinate part of the larger and more formidable project of calorimetric measurement of the enthalpy (heat content) of saturated water and steam. Since pressure and temperature of saturated vapor are definitely related, measurements of other thermal properties may be referred to either temperature or pressure as an independent variable, depending on which is the more expedient. This characteristic was utilized in planning a new high-pressure calorimetric apparatus[4] for surveying the behavior of saturated steam. As a refinement in technique, provision was made for observing both the temperature and the pressure of a water sample in the calorimeter. The arrangements for control and measurement of temperature, and for transmission of pressure to the measuring gage, were perfected to a degree which permits the rapid and reliable measurement of saturation pressure in the range from 100 C to 374 C.

Measurements within this range have already been made in several laboratories in accordance with definitely specified, recognized standards, permitting reduction of the results to a common basis. The results of these independent determinations, differing considerably in method and technique, have been regarded as in virtual agreement for technical purposes. Nevertheless, close scrutiny disclosed discrepancies that seemed unnecessarily large, particularly in the derivative on which reliance must be placed for making thermodynamic correlations of other experimental data.

The equipment available in this laboratory provided a favorable opportunity for making vapor-pressure measurements. These were undertaken prior to the calorimetric measurements with the object of further verifying the numerical values for this property.

To provide for trustworthy correlation of thermodynamic properties, it is necessary not only that the observations of the corresponding pressures and temperatures be adequate in number, distribution, and precision, but also that a formulation be used which yields reliable values of the derivative. This latter requirement has been met by the successful application of a type of empirical equation that conforms with remarkable fidelity to the observed behavior of saturated steam.

II—METHOD AND APPARATUS

1—General Description

In the calorimetric equipment used in making the vapor-pressure measurements, provision is made for observing temperature, pressure, mass of water sample, and the energy added. The arrangements provided for controlling the state of the fluid in the calorimetric experiments are also especially appropriate for the determination of vapor pressure. In this report, which concerns only the latter, the description is intended to record those features which are essential to the vapor-pressure measurements.

A schematic diagram of the vapor-pressure measuring equipment is shown in Fig. 1. The metal calorimeter shell contains besides the water sample an electric heater and a system of radial silver plates for diffusing heat. This shell is suspended within a thermally controlled enclosure or envelope, which shields it against heat exchange with the surroundings.

The pressure in the calorimeter is transmitted directly through a tube from the bottom of the calorimeter shell to a cell with a thin, elastic metal diaphragm which readily transmits pressure, yet prevents escape of the water sample. The pressure in the calorimeter is thus transmitted through this sensitive diaphragm to a column of water, and through this column to a balancing artificial atmosphere of air. The meniscus between the water

[1] Publication approved by the Director of the Bureau of Standards of the U. S. Department of Commerce.

[2] Physicist, Bureau of Standards.

[3] Assistant Physicist, Bureau of Standards.

Contributed by the Special Research Committee on Thermal Properties of Steam and presented at the Steam Tables Session of the Annual Meeting, New York, N. Y., December 5–9, 1932, of The American Society of Mechanical Engineers.

Note: The committee personnel is given on the last page.

[4] *Mechanical Engineering*, vol. 54 (1932), no. 2, p. 118.

and air is visible in a glass capillary, and serves to indicate balance between the air pressure and the pressure in the calorimeter. The pressure of the artificial atmosphere, which is subject to very delicate manual control, is transmitted to the measuring piston gage through an oil column. The air-oil meniscus is visible in another glass capillary, and is used as an indicator to show when the piston is properly loaded.

The temperature of the water sample, which is observed after the load on the piston has been properly adjusted, is obtained by the combined use of platinum-resistance thermometers and thermoelements. Resistance thermometers located in a reference block of thick silver, yield the temperature of the block according to the International Temperature Scale. The thermoelements

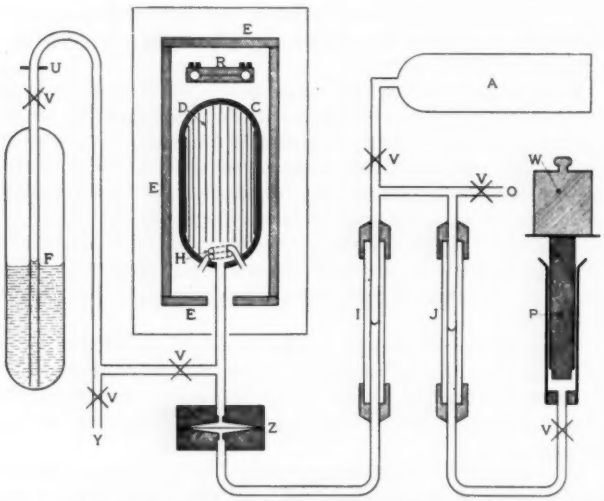


FIG. 1 SCHEMATIC DIAGRAM OF THE VAPOR-PRESSURE MEASURING EQUIPMENT

A	High-pressure air	*O*	Air vent
C	Calorimeter shell (special steel)	*P*	Piston
D	Heat-diffusing system (silver)	*R*	Reference block (silver)
E	Envelope (silver)	*U*	Union
F	Water container (silver)	*V*	Valves
H	Electric heater	*W*	Weights
I	Water indicator (glass capillary)	*Y*	Vacuum connection
J	Oil indicator (glass capillary)	*Z*	Pressure-transmission cell

with principal junctions distributed on the shell and reference junctions located on the reference block, indicate the small differences which exist between the calorimeter and the reference temperature. The thermoelements can be used in series to indicate average temperatures, or individually to indicate local temperatures. A diagrammatic scale drawing of the assembled apparatus is shown in Fig. 2. The principal parts will be described in detail forthwith.

2—THE CALORIMETER SHELL

The calorimeter shell shown at *C* in Fig. 2 provides a receptacle for the sample of water so that its thermal behavior may be observed. It has the shape of a cylinder with hemispherical ends and, as assembled with the various accessory parts in place, will hold approximately 320 cu cm. The material is a special alloy steel containing about 19 per cent chromium, 7.5 per cent nickel, 4.5 per cent tungsten, 1.3 per cent silicon, 0.5 per cent manganese, and 0.46 per cent carbon. This material was chosen because of its resistance to creep and to attack by water at temperatures up to 400 C. The shell was machined from a solid bar and was made in two similar parts held together on a thin silver gasket by the tension developed in the right- and left-threaded band which is screwed on by powerful wrenches engaging the machined lugs. This annular joint was formed by machining one member to a plane surface and the other to a blunt-angled edge, giving a contact with the silver gasket of about a half-millimeter width All parts were accurately machined and the surfaces well polished.

The shell, having a thickness of 0.125 in. (3.2 mm), was designed to stand steam pressures up to the critical without permanent deformation. Actually in hydraulic proof tests, it showed no permanent stretch for an internal pressure of 4500 lb per sq in. After carrying a charge of water at 350 C, the inner surface showed on examination no sign of attack by the water other than the formation of a very thin film of light-straw color similar to that formed on the outside where in contact with air.

3—THE CALORIMETER HEATER

The electric heater in the calorimeter shell, shown at *H1*, Fig. 2, is an insulated resistor encased in a metal tubular coil sealed to the shell. It consists of about 10 ohms of calido wire, 0.2 mm in diameter, with gold leads. This resistor was wound in helical form, 0.6 mm outside diameter, and embedded in magnesia for insulation. It was sheathed in a platinum tube drawn down tightly on the magnesia to an outside diameter of about 2 mm. This sheathed resistor was then bent into the form of a helical coil 12 mm in outside diameter, the resistor occupying about two turns. Each projecting end of this heater was put through a threaded plug of silver-palladium alloy and sealed by soldering with gold. To complete the seal, the shoulders of these plugs were drawn down tightly on thin silver gaskets by threaded nuts outside the shell.

The design of this heater was chosen with regard to influence on the calorimetric performance of the instrument. The heater surface has been made small with three objects in view, namely, to avoid unnecessary heat capacity, to permit operation with small amounts of liquid, and to avoid excessive accumulation of energy as superheat in the liquid. The characteristics of the heater have relatively little to do with the technique of pressure measurements, since it is used only for bringing the water sample to a desired temperature.

4—THE HEAT DIFFUSER

For promoting temperature equalization in the calorimeter shell, a system of heat-conducting plates is provided. This heat-diffusing system consists of 30 flat plates of silver 0.5 mm thick, shaped to conform to the vertical profile of the shell and held radially in two slotted hubs so as to penetrate and interconnect the space within the shell with a good heat conductor. The chief function of this arrangement is to hasten the equalization of temperature after a period of change. This feature is important in both the calorimetric and the pressure measurements, and to some extent compensates for the lack of positive circulation by mechanical means.

5—THE ENVELOPE

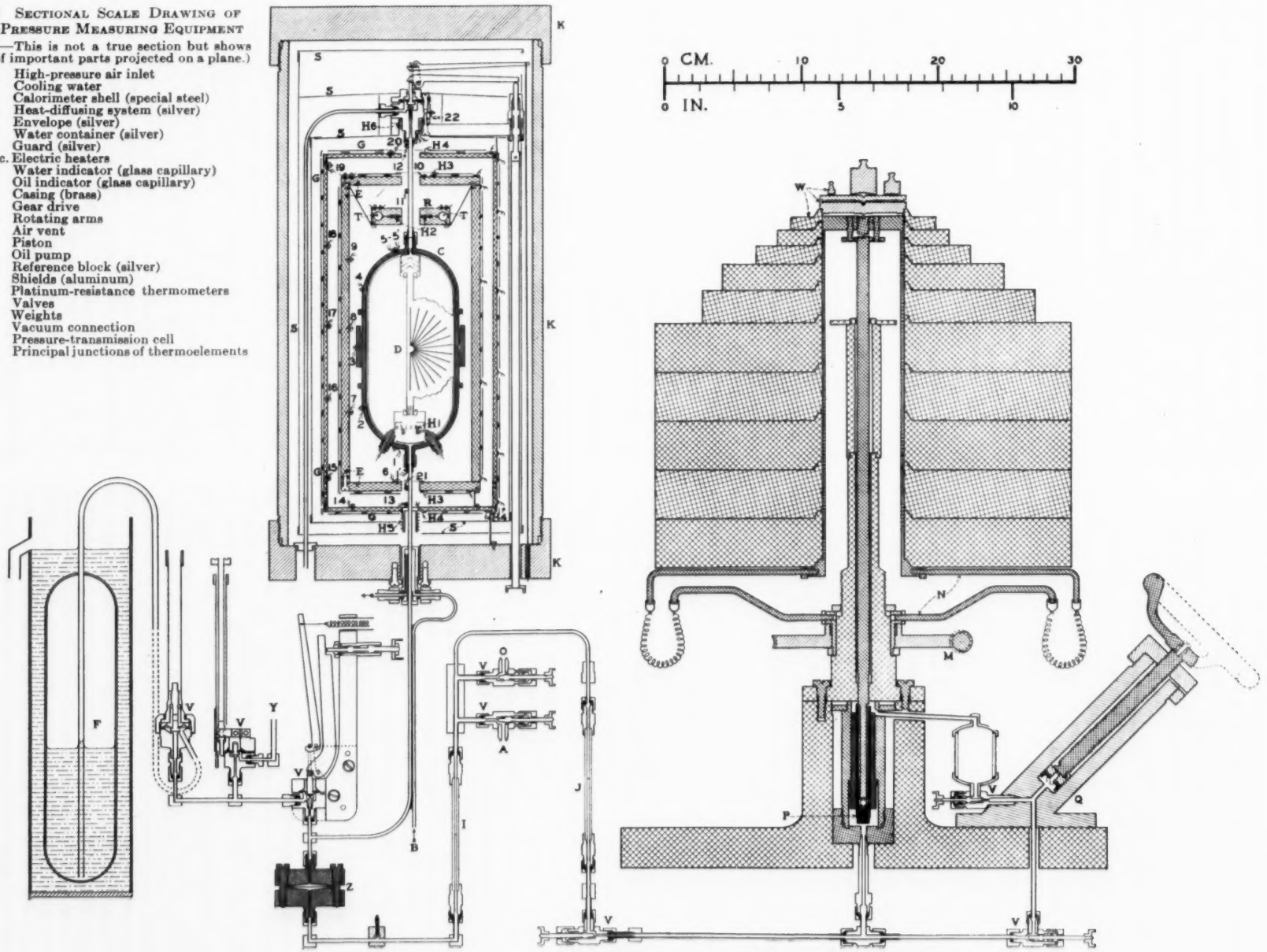
The principal purpose of the envelope that surrounds the calorimeter shell is to provide protection from fortuitous exchanges of heat. This feature, which is vital to the energy accounting in calorimetry, contributes to the suitability of the instrument for pressure measurements by favoring the attainment of thermal equilibrium of the water sample and its container.

This protecting envelope is a double-walled enclosure formed of two coaxial cylindrical silver shells with flat ends, the inner one 6.3 mm thick and the outer one 3.2 mm thick, which will be designated as "envelope" and "guard," respectively. Electric heating elements are distributed over the outer surfaces of these shells. Subdivision of these heaters provides for meeting various local thermal requirements. The heater on the envelope is used only when its temperature is being raised. The guard may also be heated at a controlled rate, or its temperature may be main-

FIG. 2 SECTIONAL SCALE DRAWING OF VAPOR-PRESSURE MEASURING EQUIPMENT

(NOTE.—This is not a true section but shows sections of important parts projected on a plane.)

A	High-pressure air inlet
B	Cooling water
C	Calorimeter shell (special steel)
D	Heat-diffusing system (silver)
E	Envelope (silver)
F	Water container (silver)
G	Guard (silver)
H_1, H_2, etc.	Electric heaters
I	Water indicator (glass capillary)
J	Oil indicator (glass capillary)
K	Casing (brass)
M	Gear drive
N	Rotating arms
O	Air vent
P	Piston
Q	Oil pump
R	Reference block (silver)
S	Shields (aluminum)
T	Platinum-resistance thermometers
V	Valves
W	Weights
Y	Vacuum connection
Z	Pressure-transmission cell
1, 2, etc.	Principal junctions of thermoelements



tained automatically at any desired value by a sensitive thermoregulator using a platinum-resistance thermometer.

At a steady temperature the guard heater supplies the heat loss to the outside while the envelope temperature remains fixed. Such conditions are favorable for the establishment and maintenance of temperature equality of the calorimeter and its contents. In the space outside the guard two thin aluminum shields are placed to impede the loss of heat by radiation and convection. These light shields, 0.05 mm thick, furnish effective thermal insulation and prevent excessive heat loss without introducing any considerable thermal lag. The whole is enclosed in a heavy brass casing which serves not only as a cover but also as protection in case of an explosive failure.

6—Connections to the Calorimeter Shell

The calorimeter shell is held in place by two tubes of silver-palladium alloy. The one at the top bears the weight of the shell and the one at the bottom centers it. Besides furnishing firm, thermally resistant support for the shell, these tubes serve also for the transfer of fluid to or from the calorimeter. The upper tube is intended for a vapor outlet to be used in calorimetric measurements of heat of vaporization, and was temporarily closed during the pressure measurements by a disk in the union at the top of the shell. This suspension tube reaches from the body of the vapor valve, which is carried on a light but firm support from the guard shell.

The lower tube is used as a connection to transfer liquid to or from the calorimeter, and also to transmit pressure from within to the auxiliary measuring equipment outside. A cylindrical brass cell for cooling water surrounds this tube where it passes through the outer brass casing. Where the tube passes through the outer silver guard shell it bears an electric heater mounted on a silver support attached to the tube. There is a free length of about 2.6 cm between this heater and the union at the shell. The purpose of the heater is to control the thermal gradient in this section of the tube, as indicated by thermoelements installed for that purpose. Similarly, the gradients on the upper tube are controlled by a heater on the vapor-valve body.

The lower tube is always filled with liquid while observing saturation behavior in the calorimeter. From the union just below the cooling cell at the bottom, connection is made through a copper-nickel tube to the liquid valve and the pressure-transmission cell. Beyond the liquid valve the line extends to a union by which the water receiver with its valve is connected. A side connection leads through a valve to a vacuum pump to permit evacuation of the calorimeter and its connecting lines.

7—Pressure Connections and Auxiliary Indicators

Pressure is transmitted from the free surface of the liquid in the calorimeter through a continuous column of liquid water to the elastic diaphragm of the pressure-transmission cell to be described forthwith. The purpose of this cell is to allow the pressure in the calorimeter system to be communicated to the pressure-measuring gage, while at the same time interposing a barrier to the escape of water from the system through the gage line. A limited amount of inward and outward movement is allowed the water column by the necessary flexibility of the transmitting device, but this movement is restricted to the small volume displacement of 0.04 cu cm required for observation. In this transmitting cell, a circular diaphragm of thin silver about 3 cm in diameter is held under slight tension between the two parts of the cell which clamp the diaphragm tightly near the edge. Small pressure changes suffice to move the diaphragm back and forth across the space between the two parts. The inner cell walls are shaped to conform approximately to the figure of the distended diaphragm. The delicate diaphragm must encounter support from the cell walls to avoid deformation beyond its elastic limit when the pressures are far out of balance.

The cell is made of two disks of stainless steel, each about 5 cm in diameter and about 1.4 cm thick, provided with 12 screws to draw them tightly together on the silver diaphragm. The two parts have nipples with unions for making connections with the lines as shown. The apertures to the inner cell are made only 0.035 cm in diameter to avoid too large an unsupported area of the thin silver diaphragm. This diaphragm is a vital part of the device, and its characteristics determine the limit of sensitivity of the transmitting cell. It is of rolled sheet silver about 0.06 mm thick. The diaphragm and cell were put together at a temperature of about 150 C in order to produce sufficient tension, when cooled, to flatten out the slight unevenness of the sheet which otherwise would interfere with the freedom of motion between the cell walls. This procedure left enough flexibility to give a satisfactory sensitivity to pressure change.

On the measuring or gage side of the diaphragm a continuous column of liquid water extends and transmits pressure to air at the meniscus boundary in a glass capillary indicator. The volume of this water column is small. The meniscus is a reliable indicator of the position of the diaphragm and is used as a null device to tell when the pressures on the two sides are balanced to bring the diaphragm to a chosen zero position, near the neutral. The air column extends to the end of an oil column in a second glass capillary indicator, whence the pressure is transmitted through the oil directly to the piston of the measuring gage. Sensitive needle valves permit fine adjustment of the pressure in the air line to balance the pressure in the calorimeter. When these pressures are balanced and the piston gage is also balanced by weights, the vapor pressure in the calorimeter may be found from the pressure measured at the piston by taking into account the fluid columns between the piston and the free surface in the calorimeter and whatever pressure differences the diaphragm may support.

The fluid columns to be accounted for are the oil, air, and water columns. The oil-column correction is constant and is easily measured. The air-column correction depends on the pressure and is almost negligible. The correction for the water column is somewhat more complicated owing to the variation of the position of each end. The position of the meniscus in the glass capillary is directly observed and easily corrected for in each measurement. From a reference point on this capillary to a point near the calorimeter there is a constant pressure correction which is computed from the difference in height. From this latter point to the free surface the pressure correction varies both with the filling in the calorimeter and with its temperature, and was computed for the conditions of each measurement.

The correction for pressure difference supported by the diaphragm is small, but arises because the chosen zero position is not necessarily the neutral or unstrained position. It was determined by making a vapor-pressure measurement near 100 C with the air column open to the atmosphere instead of being connected to the piston gage. Since the vapor pressure of water is one standard atmosphere at 100 C by definition of the International Temperature Scale, and its temperature variation there is well known, the difference between the computed pressure at the open end and the observed barometric pressure gives the desired diaphragm zero correction. This correction was determined occasionally during the progress of measurement. This method of calibration connects the scale of pressure with the standard unit at the 1-atm fixed point and in effect includes that point in the range of the observations.

A preliminary calibration of the assembled diaphragm with its capillary indicator was made at atmospheric pressure to determine its sensitivity to pressure differences. This showed the

diaphragm displacement to be nearly proportional to the pressure difference over a considerable portion of the middle range. Within this linear range a pressure difference of 0.001 atm was found to correspond to about 0.6 mm on the capillary indicator. This sensitivity calibration is merely an index of the limit of precision to be expected when the diaphragm is set to a chosen zero position and does not enter into the reduction of the observations.

8—The Pressure Gage

The pressure-measuring gage used in these determinations was one of a group of piston gages whose construction and calibration have been described previously.[5] These gages were designed and built at this Bureau to meet a need for precision pressure-measuring instruments in determining thermodynamic properties of fluids. They have been studied with great care to determine their reliability as standard instruments.

This type of gage employs a loaded rotating piston which is supported in a vertically mounted, closely fitting cylinder by oil under the pressure to be measured. A balancing load of weights on the piston is borne axially above it on a carrier which engages the piston. A horizontal couple applied to the carrier by motor-driven arms produces a slow rotation which is transmitted through it to the piston. Thus the entire load consisting of piston, carrier, and weights is rotated without introducing any appreciable vertical component of driving force. The rotation of the piston maintains a lubricating film of oil between it and the closely fitting cylinder. This provides greater freedom for vertical motion of the loaded piston in case of unbalance, by preventing direct contact between the piston and cylinder.

The gage chosen for the present work and shown in Fig. 2 has a piston area of about 1 sq cm. It was designed for the range from about 3 to 100 atm pressure. As it was desired to cover the range of steam pressures from 1 atm to 218 atm, it was necessary to make some modification to provide for this extension of the range. This was done by substituting for the original weight carrier two special ones, the first very light to permit a small load on the piston, and the second a larger carrier to accommodate the extra weights for the high pressures. This gage is provided with a set of weights in units ranging from 20 kg to 0.1 kg, specially built to stack on the carrier with proper stability. Standard laboratory weights were used for the range 100 grams to 1 gram. The weights were calibrated at the time of their use in these measurements, and corrections were applied where significant.

The effective piston area had been determined in 1928 by a series of calibrations using a multiple-column mercury manometer as the fundamental standard. Direct comparisons with this standard manometer were made at a pressure of 15 atm. In addition to these, comparisons were made at pressures up to 75 atm, using another gage to step up from the 15-atm limit of the manometer to the higher pressures. These calibrations all gave a value of 0.9961 sq cm at 20 C for the effective piston area at that time.

During the course of the present pressure measurements a comparison was made with two other similar standard piston gages, and with the vapor pressure of a standard sample of carbon dioxide as a precaution against any significant change in the effective piston area which might have occurred since its 1928 calibration. The results of these comparisons gave the same value of 0.9961 sq cm.

9—The Thermometric Installation

The thermometric installation used in the control and measurement of the temperature of the water sample includes platinum-resistance thermometers and thermoelements. Thermoelements alone suffice for the survey of temperature distribution. For determining the actual temperature of the water sample, certain of the thermoelements are used to supplement the indications of the platinum-resistance thermometers.

(*a*) *The Reference Block.* A silver reference block located in the space above the calorimeter shell serves as an isothermal union between the resistance thermometers and the reference junctions of the thermoelements. Thus the thermometers measure the temperature of the reference junctions, while the thermoelements indicate the small additional differences between this reference temperature and that of chosen points where the principal junctions are located.

The reference block is made of two similar rectangular pieces of pure silver, each 5.6 cm × 3.8 cm × 0.63 cm, held together flatwise by screws. It is suspended horizontally from the top of the envelope by four slender straps of stainless steel. A pair of electric heaters and two resistance thermometers fit in receptacles machined across the horizontal mid-section. A hole 12.7 mm in diameter in the vertical axis accommodates the small tube which suspends the calorimeter shell. This well-conducting block of silver is designed to keep the reference junctions at the same temperature as the resistance thermometers.

(*b*) *The Thermoelements.* The temperature-measuring system includes thermoelements with 38 principal junctions. These thermoelements are connected in groups to economize leads and

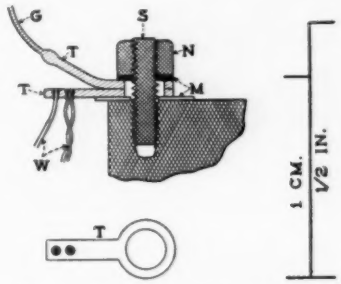


Fig. 3 Details of Thermojunction Attachment

G Gold wire
M Mica insulation
N Nut
S Stud
T Gold terminal
W Thermoelement wires

to permit combinations for surveying temperature distribution, for indicating average temperature differences of surfaces, and for determining the temperature of the water sample. These combinations are completed at the option of the observer by the use of specially built distributing switches with all-copper circuits.

The thermoelements which are used to indicate the temperature of the water are a group of five with principal junctions located on the calorimeter shell at points selected as representative. These chosen points lie in one vertical element of the shell. The five junctions, designated as 1, 2, 3, 4, and 5, are spaced as shown in Fig. 2. The indications of these five junctions may be observed individually, giving the temperature difference between any zone and the reference block, or all five may be observed in series to give a composite of the temperatures of the five zones. The single junctions may also be observed differentially against junction No. 5 at the top of the calorimeter to show directly the vertical distribution of temperature on the shell. The ability to make such a temperature survey enables the observer to follow the approach to equilibrium and make due allowance of time to assure a reliable temperature determination.

Each thermoelement is made of one wire of "Chromel P," 0.127 mm in diameter, and two wires of "Copel," 0.10 mm in diameter, the latter twisted together. For insulation and support, these wires are threaded through thin strips of mica which are assembled

[5] Meyers and Jessup, Research Paper No. 324, *Bureau of Standards Journal of Research*, vol. 6, June, 1931.

with mica separators and covers into sturdy but light and flexible bundles carrying groups of thermoelements.

Each principal junction of chromel to copel is made by silver soldering the wires to a tiny gold terminal which is used to attach the junction in the desired place. The reference junctions, two to each element, are made similarly by attaching gold terminals to the chromel and copel wires. These gold terminals make electrical connection to gold lead wires and provide for attachment of the reference junctions to the reference block.

Fig. 3 shows how the terminals are installed to secure good thermal union, electrical insulation, and electrical connection where desired. They are held firmly under nuts on threaded studs which are screwed into the metal at the desired places. The terminals are insulated electrically from above and below by mica washers, and from the stud by centering. Additional thermal attachments of the gold lead wires are made in a similar manner at the reference block and at the bottom of the envelope to prevent thermal lead conduction from directly affecting the

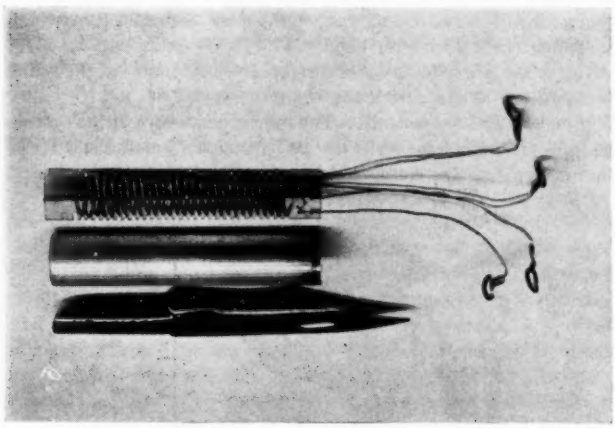


FIG. 4 PLATINUM-RESISTANCE THERMOMETER AND SHEATH, WITH PEN POINT FOR COMPARISON OF SIZE

reference junctions and thus causing erroneous temperature indications.

Gold wires, insulated with mica as described above, are brought out in a bundle from the reference junctions within the heated space to an accessible place at the temperature of the room. Here an isothermal attachment on copper blocks is provided for the junctions between these gold wires and the copper wires which lead across to the observing station.

The electromotive forces of the thermoelements, amounting in nearly all cases to less than 20 microvolts, are measured on a Wolff potentiometer designed by F. Wenner. Calibration of this instrument showed its corrections to be negligible. When the temperature of the calorimeter is being observed by means of the five thermoelements in series, a scale deflection of 1 mm corresponds to about 0.001 C.

The five thermoelements which are used in the measurement of water temperature were calibrated in place against a resistance thermometer in the reference block by the following procedure. With the block at nearly the same temperature as the calorimeter, a small emf is indicated on the thermoelements. When the block is heated the thermoelement reference junctions are heated by the same amount as the resistance thermometer. Therefore, the change in thermoelement emf is equivalent to the change in temperature as indicated by the thermometer. Calibrations were made at every temperature where pressure observations were made.

(*c*) *The Resistance Thermometers.* Two specially constructed platinum-resistance thermometers were used as working standards for the temperature measurements. These are of the four-lead potential-terminal type. They were made small and compact to fit into the receptacles in the reference block. The windings were made of highly refined platinum which showed at 100 C a resistance as high as 1.392 times that at 0 C. The platinum wire, 0.1 mm in diameter, was first wound in the form of a helix 0.45 mm in diameter, and this was then wound into a second helix 4.8 mm in diameter upon a mica cross with the edges notched to carry the winding, yet leave it free from mechanical constraint. Initial strains were relieved by annealing the completed thermometer at 660 C. Each thermometer is mounted in a cylindrical silver case which fits the receptacle in the reference block. Fig. 4 shows the construction of one of these thermometers.

This type of thermometer winding has been described by C. H. Meyers[6] and is particularly adapted to use where large size is objectionable. The question as to whether a platinum winding of this compact double-helical type will define the same temperature scale between the fixed points as a winding of the customary strain-free type, has been studied by making a direct comparison between one of the thermometers of this apparatus and one of the earlier standard thermometers of this laboratory. In the interval from 200 C to 320 C, the maximum observed difference in indication was at 270 C and amounted to 0.013 C, which is not more that attributable to the uncertainty of reproduction of the scale itself.

The two thermometers used (M-22 and M-26) were selected from a group of eight by consideration of their characteristics and behavior over a period of time. They were calibrated according to the specifications of the International Temperature Scale,[7] which uses the fixed points of ice, steam, and boiling sulphur as 0, 100, and 444.6 deg, respectively. The thermometers were calibrated before and during the progress of the pressure measurements and showed no significant changes. Thermometer No. M-22, which was used in all the experiments, had the constants $R_0 = 27.6637$, $R_{100} = 38.5158$, $\delta = 1.496$, which were used in the Callendar formula (given later) for the temperature computation. Thermometer No. M-26, which was used only occasionally to check the other thermometer, had the constants $R_0 = 27.7640$, $R_{100} = 38.6555$, $\delta = 1.496$.

The resistances of the thermometers are measured with a Mueller[8] bridge built by O. Wolff. The bridge coils are immersed in a thermostated oil bath. A separate commutator switch permits the observation of either thermometer. The bridge coils were recalibrated several times during this investigation. A bridge current of 4.5 milliamperes was used both in the calibration and in the measurements of temperature. A galvanometer-scale deflection of 1 mm corresponded approximately to 0.0001 ohm, or to about 0.001 C for the thermometers used.

III—EXPERIMENTAL PROCEDURE

The first step in a vapor-pressure determination is the introduction of a chosen amount of pure air-free water into the calorimeter. The water used in these measurements was prepared from distilled water by continuous low-pressure distillation in a special apparatus. The air was removed by pumping from the condenser a small fraction of the vapor which carried with it all but a trace of the air dissolved in the original water. This purification was found adequate to avoid any measurable partial pressure of air in the steam. A measured sample of the purified water is transferred without contact with air from the distilling apparatus to

[6] *Bureau of Standards Journal of Research*, vol. 9, Dec., 1932, R. P 508.

[7] Burgess, Research Paper No. 22, *Bureau of Standards Journal of Research*, Oct., 1928, p. 635.

[8] Mueller, *Bulletin of the Bureau of Standards*, vol. 13 (1916), p. 547.

a special container. With this container attached to the evacuated calorimeter system, the water is driven into the calorimeter shell by heating the container. Weighing the container before and after this operation checks the amount transferred to the calorimeter. This must be known in order to compute the correction for the height of the liquid at any time.

The calorimeter with its charge of water is next brought to the desired temperature by adding heat electrically. At the same time the envelope, guard, reference block, and connecting tubes are all heated in a similar manner and at about the same rate. Their temperatures are finally adjusted by successive approximations while the calorimeter with its contents approaches the desired uniform stationary temperature. As this chosen temperature is approached, the automatic temperature control of the guard shell is put into operation.

The behavior of the fluid in the calorimeter varies appreciably in the temperature range covered by these experiments. At temperatures below 350 C, thermal gradients produced by heating the fluid, diminish quickly, leaving only small persistent gradients. As the critical region is approached, however, these persistent gradients become larger. The procedure used to hasten favorable conditions for observing pressure at these high temperatures is first to heat the water above the desired temperature and then to cool by lowering the envelope temperature. This produces condensation on the upper calorimeter walls, bathing them with liquid, which tends to bring them nearer the effective saturation temperature.

As equilibrium is approached, successive approximations are made to the adjustments necessary for a pressure measurement. The calorimeter pressure is balanced by the air pressure, which in turn is balanced by the load on the piston. If the temperature and pressure in the calorimeter were stationary and the measuring apparatus adjusted to proper balance, the diaphragm of the pressure cell would be at rest in its zero position, the two capillary indicators stationary, and the loaded rotating piston of the gage neither rising nor falling. This ideal condition of absolute constancy is not necessary to a satisfactory measurement. After close control and adjustment have been obtained, they are held over a period of about two minutes while a series of temperature readings is taken. The definitive value of the load is that at the mid-point of this period. The pressure observation includes the balancing load on the piston, temperature of the piston, barometric pressure, and height of fluid columns in the capillary indicators. The balancing load could always be determined to the nearest gram, and at low pressures it was possible to estimate fractions of a gram. At the higher pressures it is unnecessary to determine the load more closely than to 1 gram in order to have the precision of the pressure measurement correspond with that of the temperature.

For determining the effective saturation temperature, four successive temperature readings are made at equal time intervals. Each temperature reading consists of simultaneous observations of resistance thermometer and thermoelements. The above-described method of reading several successive temperatures is desirable for several reasons. First, the four-lead potential-terminal resistance thermometer requires at least two observations to eliminate the lead resistance from the measurement. Second, increasing the number of readings decreases the accidental error of observation. Third, a regular schedule of readings takes account of slight drift of temperatures. Fourth, the schedule permits the observation of local temperature irregularities during the measurement.

The best selection of thermoelements to determine the temperature of the free surface of the water depends on the temperature distribution and the location of the liquid level in the calorimeter. In the range of temperature below 350 C, the water and calorimeter reach a steady state promptly. Below 200 C this steady state was a uniform temperature, as indicated by surveys. All five thermoelements were therefore taken to indicate the saturation temperature up to 200 C.

Between 200 C and 350 C, after the steady state was reached, the bottom of the shell showed a small persistent depression of temperature and the top a small elevation due to inability to control the surroundings perfectly. In this range the three intermediate thermoelements, Nos. 2, 3, and 4, were used.

Above 350 C a difference in the behavior of the fluid was observed. The steady state was reached more slowly and thermal gradients were larger over the entire shell. The saturation temperature was estimated from the indication of a single thermoelement located in the vicinity of the liquid level in the calorimeter. If the quantity of water in the calorimeter is checked as previously described, the location of the liquid level is reliably known at temperatures below 370 C. Near the critical region, a comparatively small difference in the water sample determines whether saturated vapor, superheated vapor, or compressed liquid is present. Experiments were therefore made to indicate the state of the water in the calorimeter. These experiments started with a calorimeter supposed to be full of liquid water. After the pressure was observed, a chosen small amount of liquid was withdrawn and the pressure again observed. This procedure was repeated until the observed pressure, when reduced to a given temperature, showed a constant value, indicating existence of the saturation state.

During the progress of the vapor-pressure measurements, two types of experiments were made to prove the absence of an appreciable amount of permanent gas in the calorimeter. In the first type, pressures were measured before and after the withdrawal of liquid, which increased the vapor space in the calorimeter. In the second type, the presence of gas in the calorimeter after removal of the liquid was tested by a McLeod gage. These tests gave no indication of enough gas to affect the vapor pressure results.

IV—RESULTS OF MEASUREMENTS

The results of the entire series of measurements have been assembled in Table 1, which includes each measured temperature, reduced to degrees of the International Temperature Scale of 1927[9] and the corresponding measured pressure reduced to International standard atmospheres.[10] This table also contains the reduction of the observed pressures to values corresponding to even temperatures to permit comparison of the individual determinations and to facilitate the formulation of the entire group of results.

Each measured temperature is computed from the observed data consisting of four readings of the bridge when balanced with the platinum-resistance thermometer in circuit, and the four simultaneous readings of the potentiometer when balanced against the thermoelements, as described above in III—Experimental Procedure.

Each measured pressure is computed from the following observed quantities: the load on the piston gage, temperature of gage, position of water meniscus and oil meniscus, amount of water in the calorimeter, and barometer reading, including its temperature.

The auxiliary data used in these reductions include the density

[9] Burgess, Research Paper No. 22, *Bureau of Standards Journal of Research*, Oct., 1928, p. 635.

[10] "Standard atmospheric pressure is defined as the pressure due to a column of mercury 760 mm high, having a mass of 13,5951 g/cm³, subject to a gravitational acceleration of 980.665 cm/sec² and is equal to 1,013,250 dynes/cm²," *Bureau of Standards Journal of Research*, Oct., 1928, p. 637.

of water and of oil, value of gravity at the Bureau of Standards, the relative elevation of gage and barometer from the gravity bench mark, the results of calibrations of the bridge, resistance thermometers, thermoelements, and the piston gage with its connecting lines and weights.

The mean of the four bridge readings corrected for the bridge calibration gives the resistance of the platinum thermometer (R_θ) at the mean temperature of the series. The temperature θ is computed from this resistance by use of the Callendar formula

$$\theta = 100\,(R_\theta - R_0)/(R_{100} - R_0) + 0.01\,\theta\,(0.01\,\theta - 1)\delta$$

The constants R_0, R_{100}, and δ are determined by the calibration previously described.

The mean of the four thermoelement observations is reduced from microvolts to degrees temperature difference by use of the calibration factor determined as previously described. By combining this mean temperature difference between the water and the reference block, and the mean temperature of the block as determined by the thermometer, the temperature of the water, θ_w, is obtained.

The pressure observation, made at the middle of the series of temperature readings, corresponds with the water temperature determined as above. In reducing the pressure observation, the effective weight of the entire load supported by the oil acting on the effective piston area was computed as the sum of the masses of the weights, weight carrier, and the piston, corrected for calibration and air buoyancy. From this weight the resultant pressure was computed by use of the value of gravity at this laboratory (980.09 cm per sec per sec) relative to the standard value of gravity (980.665 cm per sec per sec) and the effective piston area, corrected for the effect of thermal expansion. To this component of pressure due to the load was added the observed barometric pressure corrected for temperature, difference in level, and for gravity, thus giving the total pressure at the bottom of the piston. From this pressure at the gage, the pressure at the level of the liquid in the calorimeter was found by applying the following corrections for the intermediate fluid columns: the oil column between the piston and the oil meniscus, the air column between oil and water, the water column between the water meniscus and the water level in the calorimeter, and the correction for the diaphragm position determined by the calibration at atmospheric pressure. The pressure contributed by the liquid water column in the calorimeter was calculated from the mass of water in the calorimeter, the dimensions of the calorimeter and the specific volumes of vapor and liquid water determined by Keyes and Smith.[11]

The total correction for fluid columns in the transmission line did not exceed 0.04 atm and was estimated to 0.0001 atm. The accuracy of this estimation may have limited the precision attainable at the lowest pressures measured. At the higher pressures, it was of less importance in comparison with several other factors.

V—FORMULATION OF RESULTS

Proceeding with the mean values of vapor pressure corresponding to even temperatures assembled in Table 1 as described above, the next step was to express the aggregate result of the entire series by means of a formula in order to smooth out irregularities caused by accidental errors of observation, and to provide a trustworthy method for interpolating intermediate values and for obtaining the derivative. The results of this formulation are given in Table 2. The constants of the empirical equation were determined by the method of least squares. Since the pressure

[11] *Mechanical Engineering*, vol. 53 (1931), no. 2, p. 133.

TABLE 1 OBSERVED PRESSURE OF SATURATED WATER VAPOR

Date	Observed quantities: Temp θ_w deg C (Int.)	Pressure P_w	Reduction to even temp	Vapor pressure at even temp P	Residual, $P - P_m$
		Standard atmospheres (Int.)			
7-20-32	110.027	1.4146	−0.0013	1.4133	−0.0010
7-20-32	110.018	1.4152	−0.0009	1.4143	0.0000
7-20-32	110.003	1.4149	−0.0001	1.4148	+0.0005
7-20-32	110.019	1.4156	−0.0009	1.4147	+0.0004
7-21-32	110.011	1.4148	−0.0005	1.4143	0.0000
7-21-32	109.994	1.4138	+0.0003	1.4141	−0.0002
7-21-32	110.008	1.4146	−0.0004	1.4142	−0.0001
7-21-32	110.027	1.4156	−0.0013	1.4143	0.0000
(Even temperature 110 C; mean value of pressure at 110 C = P_m = 1.4143 atm)					
7-20-32	119.975	1.9579	+0.0015	1.9594	−0.0004
7-20-32	119.991	1.9587	+0.0006	1.9593	−0.0005
7-20-32	119.981	1.9587	+0.0012	1.9599	+0.0001
7-20-32	120.010	1.9608	−0.0006	1.9602	+0.0004
7-21-32	119.982	1.9590	+0.0011	1.9601	+0.0003
7-21-32	119.956	1.9564	+0.0027	1.9591	−0.0007
7-21-32	120.004	1.9604	−0.0003	1.9601	+0.0003
7-21-32	119.988	1.9596	+0.0007	1.9603	+0.0005
(Even temperature 120 C; mean value of pressure at 120 C = P_m = 1.9598 atm)					
4-2-32	129.965	2.6624	+0.0028	2.6652	−0.0006
4-2-32	129.965	2.6635	+0.0028	2.6663	+0.0005
4-2-32	129.965	2.6636	+0.0028	2.6664	+0.0006
4-5-32	130.014	2.6656	−0.0011	2.6645	−0.0013
4-7-32	130.008	2.6644	−0.0006	2.6638	−0.0020
4-8-32	129.996	2.6641	+0.0003	2.6644	−0.0014
7-2-32	129.990	2.6652	+0.0007	2.6659	+0.0001
7-2-32	129.971	2.6636	+0.0023	2.6659	+0.0001
7-2-32	130.000	2.6660	0.0000	2.6660	+0.0002
7-7-32	130.046	2.6692	−0.0037	2.6655	−0.0003
7-21-32	129.998	2.6665	+0.0002	2.6667	+0.0009
7-21-32	129.976	2.6649	+0.0019	2.6668	+0.0010
7-21-32	130.003	2.6670	−0.0002	2.6668	+0.0010
7-21-32	130.023	2.6686	−0.0018	2.6668	+0.0010
(Even temperature 130 C; mean value of pressure at 130 C = P_m = 2.6658 atm)					
4-2-32	140.370	3.6027	−0.0373	3.5654	−0.0007
4-2-32	140.371	3.6028	−0.0374	3.5654	−0.0007
4-2-32	140.371	3.6028	−0.0374	3.5654	−0.0007
4-5-32	140.023	3.5682	−0.0023	3.5659	−0.0002
4-7-32	139.980	3.5614	+0.0020	3.5634	−0.0027
4-8-32	139.983	3.5625	+0.0017	3.5642	−0.0019
7-2-32	139.984	3.5653	+0.0016	3.5669	+0.0008
7-2-32	139.968	3.5634	+0.0032	3.5666	+0.0005
7-2-32	140.005	3.5675	−0.0005	3.5670	+0.0009
7-7-32	140.004	3.5665	−0.0004	3.5661	0.0000
7-21-32	139.975	3.5654	+0.0025	3.5679	+0.0018
7-21-32	139.987	3.5662	+0.0012	3.5674	+0.0013
7-21-32	140.021	3.5694	−0.0021	3.5673	+0.0012
(Even temperature 140 C; mean value of pressure at 140 C = P_m = 3.5661 atm)					
3-25-32	149.879	4.6802	+0.0152	4.6954	−0.0015
3-28-32	149.911	4.6849	+0.0112	4.6961	−0.0008
3-29-32	149.868	4.6789	+0.0166	4.6955	−0.0014
3-30-32	150.032	4.7018	−0.0040	4.6978	+0.0009
4-1-32	149.998	4.6970	+0.0002	4.6972	+0.0003
4-7-32	149.984	4.6922	+0.0020	4.6942	−0.0027
4-8-32	150.004	4.6964	−0.0005	4.6959	−0.0010
7-7-32	150.035	4.7017	−0.0044	4.6973	+0.0004
7-21-32	149.985	4.6971	+0.0019	4.6990	+0.0021
7-21-32	150.004	4.6994	−0.0005	4.6989	+0.0020
7-21-32	150.025	4.7019	−0.0031	4.6988	+0.0019
(Even temperature 150 C; mean value of pressure at 150 C = P_m = 4.6969 atm)					
4-2-32	160.067	6.1101	−0.0104	6.0997	−0.0001
4-2-32	160.068	6.1102	−0.0106	6.0996	−0.0002
4-2-32	160.068	6.1102	−0.0106	6.0996	−0.0002
4-7-32	160.027	6.1028	−0.0042	6.0986	−0.0012
4-8-32	159.990	6.0972	+0.0016	6.0988	−0.0010
7-2-32	160.016	6.1031	−0.0025	6.1006	+0.0008
7-2-32	159.999	6.0996	+0.0002	6.0998	0.0000
7-2-32	160.022	6.1036	−0.0034	6.1002	+0.0004
7-7-32	160.016	6.1023	−0.0025	6.0998	0.0000
7-21-32	159.965	6.0956	+0.0054	6.1010	+0.0012
7-21-32	159.978	6.0950	+0.0034	6.0984	−0.0014
7-21-32	159.997	6.1008	+0.0005	6.1013	+0.0015
(Even temperature 160 C; mean value of pressure at 160 C = P_m = 6.0998 atm)					
4-2-32	169.994	7.8182	+0.0011	7.8193	+0.0015
4-2-32	169.993	7.8182	+0.0013	7.8195	+0.0017
4-2-32	169.994	7.8182	+0.0011	7.8193	+0.0015
4-5-32	170.016	7.8183	−0.0030	7.8153	−0.0025
4-7-32	169.993	7.8147	+0.0013	7.8160	−0.0018
4-8-32	169.992	7.8140	+0.0015	7.8155	−0.0023
7-2-32	169.991	7.8159	+0.0017	7.8176	−0.0002
7-2-32	169.981	7.8140	+0.0036	7.8176	−0.0002
7-2-32	169.996	7.8168	+0.0008	7.8176	−0.0002
7-7-32	170.002	7.8174	−0.0004	7.8170	−0.0008
7-21-32	169.978	7.8148	+0.0042	7.8190	+0.0012
7-21-32	169.996	7.8182	+0.0008	7.8190	+0.0012
7-21-32	170.014	7.8216	−0.0026	7.8190	+0.0012
(Even temperature 170 C; mean value of pressure at 170 C = P_m = 7.8178 atm)					

Date	Observed quantities: Temp ϑ_w deg C (Int.)	Pressure P_w	Reduction to even temp	Vapor pressure at even temp P	Residual, $P - P_m$
		Standard atmospheres (Int.)			
4-2-32	180.024	9.9023	−0.0055	9.8968	+0.0010
4-2-32	180.022	9.9011	−0.0050	9.8961	+0 0003
4-2-32	180.022	9.9017	−0.0050	9.8967	+0.0009
4-5-32	180.014	9.8990	−0.0032	9.8958	0.0000
4-7-32	180.001	9.8966	−0.0002	9.8964	+0.0006
4-8-32	179.999	9.8918	+0.0002	9.8920	−0.0038
7-7-32	179.982	9.8926	+0.0041	9.8967	+0.0005
7-7-32	179.969	9.8887	+0.0071	9.8958	0.0000
7-7-32	179.996	9.8953	+0.0009	9.8962	+0.0004
(Even temperature 180 C; mean value of pressure at 180 C = P_m = 9.8958 atm)					
4-2-32	190.001	12.3913	−0.0003	12.3910	+0.0023
4-2-32	190.002	12.3904	−0.0005	12.3899	+0.0012
4-2-32	190.002	12.3905	−0.0005	12.3900	+0 0013
4-5-32	190.005	12.3890	−0.0014	12.3876	−0.0011
4-7-32	190.010	12 3908	−0.0027	12.3881	−0.0006
4-8-32	190.010	12.3898	−0.0027	12.3871	−0.0016
7-7-32	190.010	12.3908	−0.0027	12.3881	−0.0006
7-7-32	189.998	12.3871	+0.0005	12.3876	−0.0011
7-7-32	190.012	12.3918	−0.0033	12.3885	−0 0002
(Even temperature 190 C; mean value of pressure at 190 C = P_m = 12.3887 atm)					
3-25-32	200.141	15.3920	−0.0453	15.3467	−0.0005
3-28-32	199.957	15.3332	+0.0138	15.3470	−0.0002
3-29-32	200.037	15.3576	−0.0119	15.3457	−0.0015
3-30-32	199.700	15.2530	+0.0963	15.3493	+0.0021
4-1-32	199.935	15.3264	+0.0209	15.3473	+0.0001
4-7-32	199.990	15.3443	+0.0032	15.3475	+0.0003
4-8-32	199.712	15.2535	+0.0924	15.3459	−0.0013
7-7-32	199.996	15.3463	+0.0013	15.3476	+0.0004
7-7-32	199.976	15.3395	+0.0077	15.3472	0.0000
7-7-32	199.987	15.3440	+0.0042	15.3482	+0.0010
(Even temperature 200 C; mean value of pressure at 200 C = P_m = 15.3472 atm)					
4-4-32	209.991	18.8256	+0.0034	18.8290	−0.0010
4-5-32	210.026	18.8393	−0.0098	18.8295	−0.0005
4-7-32	210.006	18.8314	−0.0023	18.8291	−0.0009
4-8-32	210.030	18.8424	−0.0113	18.8311	+0.0011
4-29-32	209.979	18.8245	+0.0079	18.8324	+0.0024
7-7-32	209.979	18.8213	+0 0079	18.8292	−0.0008
7-7-32	209.957	18.8133	+0.0162	18.8295	−0.0005
7-7-32	209.994	18.8277	+0.0023	18.8300	0.0000
(Even temperature 210 C; mean value of pressure at 210 C = P_m = 18.8300 atm)					
4-4-32	219.998	22.8960	+0.0009	22.8969	+0.0005
4-5-32	220.002	22.8963	−0.0009	22.8954	−0.0010
4-7-32	220.003	22.8960	−0.0013	22.8947	−0.0017
4-8-32	220.013	22.9006	−0.0057	22.8949	−0.0015
4-29-32	219.993	22.8948	+0.0031	22.8979	+0.0015
7-7-32	219.999	22.8963	+0.0004	22.8967	+0.0003
7-7-32	219.977	22.8867	+0.0101	22.8968	+0.0004
7-7-32	219.996	22.8960	+0.0018	22.8978	+0.0014
(Even temperature 220 C; mean value of pressure at 220 C = P_m = 22.8964 atm)					
4-4-32	230.007	27.6156	−0.0036	27.6120	+0.0003
4-5-32	230.019	27.6187	−0.0096	27.6091	−0.0026
4-7-32	230.015	27.6208	−0.0076	27.6132	+0.0015
4-8-32	230.010	27.6137	−0.0051	27.6086	−0.0031
4-29-32	230.006	27.6160	−0.0030	27.6130	+0.0013
7-7-32	229.986	27.6055	+0.0071	27.6126	+0.0009
7-7-32	229.951	27.5883	+0.0248	27.6131	+0 0014
7-7-32	229.999	27.6115	+0.0005	27.6120	+0.0003
(Even temperature 230 C; mean value of pressure at 230 C = P_m = 27.6117 atm)					
4-4-32	240.014	33.0550	−0.0081	33.0469	+0.0053
4-5-32	240.060	33.0706	−0.0349	33.0357	−0.0059
4-7-32	240.015	33.0474	−0.0087	33.0387	−0.0029
4-8-32	240.021	33.0499	−0.0122	33.0377	−0.0039
4-29-32	240.026	33.0575	−0.0151	33.0424	+0.0008
7-8-32	239.976	33.0307	+0.0139	33.0446	+0.0030
7-8-32	239.955	33.0188	+0.0261	33.0449	+0.0039
(Even temperature 240 C; mean value of pressure at 240 C = P_m = 33.0416 atm)					
3-25-32	250.186	39.3760	−0.1233	39.2527	−0.0039
3-28-32	250.026	39.2729	−0.0172	39.2557	−0.0009
3-29-32	249.968	39.2340	+0.0212	39.2552	−0.0014
3-30-32	249.832	39.1476	+0.1114	39.2590	+0.0024
4-1-32	250.131	39.3401	−0.0869	39.2532	−0.0034
4-7-32	250.030	39.2729	−0.0199	39.2530	−0.0036
4-8-32	250.005	39.2660	−0.0033	39.2627	+0.0061
7-8-32	249.977	39.2403	+0.0153	39.2556	−0.0010
7-8-32	249.933	39.2148	+0.0444	39.2592	+0.0026
7-8-32	249.985	39.2497	+0.0099	39.2596	+0.0030
(Even temperature 250 C; mean value of pressure at 250 C = P_m = 39.2566 atm)					
3-28-32	259.991	46.3170	+0.0068	46.3238	−0.0048
4-4-32	259.999	46.3287	+0.0008	46.3295	+0.0009
4-5-32	260.011	46.3334	−0.0083	46.3251	−0.0035
4-7-32	260.010	46.3326	−0.0075	46.3251	−0.0035
4-8-32	260.005	46.3367	−0.0038	46.3329	+0.0043
4-29-32	259.998	46.3244	+0.0015	46.3259	−0.0027
7-8-32	259.986	46.3200	+0.0105	46.3305	+0.0019
7-8-32	259.970	46.3086	+0.0226	46.3312	+0.0026
7-8-32	259.997	46.3310	+0.0023	46.3333	+0.0047
(Even temperature 260 C; mean value of pressure at 260 C = P_m = 46.3286 atm)					
3-28-32	269.996	54.3349	+0.0034	54.3383	+0.0050
4-4-32	270.000	54.3318	0.0000	54.3318	−0.0015
4-5-32	270.030	54.3533	−0.0255	54.3278	−0.0065
4-7-32	270.018	54.3454	−0.0153	54.3301	−0.0032
4-8-32	269.986	54.3193	+0.0119	54.3312	−0.0021
4-26-32	269.994	54.3305	+0.0051	54.3356	+0.0023
4-26-32	270.002	54.3337	−0.0017	54.3320	−0.0013
4-28-32	269.990	54.3245	+0.0085	54.3330	−0.0003
4-28-32	270.002	54.3325	−0.0017	54.3308	−0.0025
4-29-32	269.994	54.3303	+0.0051	54.3354	+0.0021
7-8-32	269.990	54.3286	+0.0085	54.3371	+0.0038
7-8-32	269.975	54.3138	+0.0213	54.3351	+0.0018
7-8-32	269.995	54.3302	+0.0043	54.3345	+0.0012
(Even temperature 270 C; mean value of pressure at 270 C = P_m = 54.3333 atm)					
4-4-32	275.059	58.7648	−0.0532	58.7116	−0.0009
4-5-32	275.415	59.0855	−0.3756	58.7099	−0.0026
4-7-32	275.015	58.7218	−0.0135	58.7083	−0.0042
4-8-32	275.669	59.3127	−0.6041	58.7086	−0.0039
7-8-32	274.990	58.7073	+0.0090	58.7163	+0.0038
7-8-32	274.968	58.6857	+0.0289	58.7146	+0.0021
7-8-32	274.963	58.6820	+0.0334	58.7154	+0.0029
7-8-32	274.998	58.7132	+0.0018	58.7150	+0.0025
(Even temperature 275 C; mean value of pressure at 275 C = P_m = 58.7125 atm)					
3-28-32	280.015	63.3688	−0.0143	63.3545	−0.0013
4-4-32	280.009	63.3635	−0.0086	63.3545	−0.0013
4-5-32	280.219	63.5622	−0.2093	63.3529	−0.0029
4-7-32	280.054	63.4020	−0.0516	63.3504	−0.0054
4-8-32	279.996	63.3651	+0.0038	63.3689	+0.0131
4-26-32	280.098	63.4445	−0.0937	63.3508	−0.0050
4-29-32	280.005	63.3602	−0.0048	63.3554	−0.0004
7-8-32	279.988	63.3444	+0.0115	63.3559	+0.0001
7-8-32	279.962	63.3214	+0.0363	63.3577	+0.0019
7-8-32	279.988	63.3458	+0.0115	63.3573	+0.0015
(Even temperature 280 C; mean value of pressure at 280 C = P_m = 63.3558 atm)					
3-28-32	290.406	73.9146	−0.4351	73.4795	+0.0016
4-4-32	290.027	73.5075	−0.0289	73.4786	+0.0007
4-5-32	290.027	73.5066	−0.0289	73.4777	−0.0002
4-7-32	290.001	73.4771	−0.0011	73.4760	−0.0019
4-8-32	290.009	73.4944	−0.0096	73.4848	+0.0069
4-26-32	290.014	73.4824	−0.0150	73.4674	−0.0105
4-26-32	290.023	73.4963	−0.0246	73.4717	−0.0062
4-29-32	290.040	73.5189	−0.0428	73.4761	−0.0018
7-8-32	289.986	73.4657	+0.0150	73.4807	+0.0028
7-8-32	289.949	73.4254	+0.0546	73.4800	+0.0021
7-8-32	289.996	73.4796	+0.0043	73.4839	+0.0060
(Even temperature 290 C; mean value of pressure at 290 C = P_m = 73.4779 atm)					
3-25-32	300.129	84.9464	−0.1546	84.7918	−0.0051
3-28-32	300.005	84.8075	−0.0060	84.8015	+0.0046
3-29-32	299.970	84.7592	+0.0358	84.7950	−0.0019
3-30-32	299.953	84.7475	+0.0561	84.8036	+0.0067
4-1-32	300.048	84.8578	−0.0573	84.8005	+0.0036
4-7-32	300.047	84.8492	−0.0561	84.7931	−0.0038
4-8-32	300.025	84.8232	−0.0299	84.7933	−0.0036
6-30-32	299.609	84.3244	+0.4665	84.7909	−0.0060
6-30-32	299.954	84.7441	+0.0549	84.7990	+0.0021
6-30-32	299.813	84.5766	+0.2234	84.8000	+0.0031
7-1-32	299.654	84.3803	+0.4128	84.7931	−0.0038
7-8-32	300.009	84.8095	−0.0107	84.7988	+0.0019
7-8-32	299.974	84.7671	+0.0311	84.7982	+0.0013
7-8-32	299.973	84.7662	+0.0322	84.7984	+0.0015
7-8-32	300.012	84.8111	−0.0143	84.7968	−0.0001
(Even temperature 300 C; mean value of pressure at 300 C = P_m = 84.7969 atm)					
3-28-32	310.019	97.4298	−0.0253	97.4045	−0.0017
4-4-32	310.039	97.4615	−0.0518	97.4097	+0.0035
4-5-32	310.036	97.4503	−0.0478	97.4025	−0.0037
4-7-32	309.980	97.3822	+0.0266	97.4088	+0.0026
4-8-32	309.902	97.2816	+0.1302	97.4118	+0.0056
4-25-32	310.015	97.4259	−0.0199	97.4060	−0.0002
4-25-32	310.015	97.4246	−0.0199	97.4047	−0.0015
4-25-32	310.017	97.4264	−0.0226	97.4038	−0.0024
4-29-32	310.030	97.4361	−0.0399	97.3962	−0.0100
7-1-32	309.975	97.3685	+0.0332	97.4017	−0.0035
7-11-32	309.993	97.4006	+0.0093	97.4099	+0.0037
7-11-32	309.973	97.3753	+0.0359	97.4112	+0.0050
7-11-32	309.989	97.3949	+0.0146	97.4095	+0.0033
(Even temperature 310 C; mean value of pressure at 310 C = P_m = 97.4062 atm)					
4-4-32	320.070	111.524	−0.103	111.421	+0.003
4-5-32	320.032	111.473	−0.047	111.426	+0.008
4-7-32	320.045	111.493	−0.066	111.427	+0.009
4-8-32	320.020	111.460	−0.030	111.430	+0.012
4-29-32	320.017	111.447	−0.025	111.422	+0.004
7-1-32	320.007	111.426	−0.010	111.416	−0.002
7-1-32	320.007	111.422	−0.010	111.412	−0.006
7-1-32	319.957	111.333	+0.063	111.396	−0.022
7-1-32	320.012	111.418	−0.018	111.400	−0.018
7-11-32	320.017	111.446	−0.025	111.421	+0.003
7-11-32	319.994	111.415	+0.009	111.424	+0.006
7-11-32	320.003	111.425	−0.004	111.421	+0.003
(Even temperature 320 C; mean value of pressure at 320 C = P_m = 111.418 atm)					

Date	Observed quantities: Temp θ_w deg C (Int.)	Pressure P_w	Reduction to even temp	Vapor pressure at even temp P	Residual, $P - P_m$
		Standard atmospheres (Int.)			
4-4-32	325.025	119.033	−0.039	118.994	+0.006
4-5-32	325.033	119.043	−0.051	118.992	+0.004
4-7-32	325.004	118.994	−0.006	118.988	0.000
4-8-32	325.004	118.998	−0.006	118.992	+0.004
7-1-32	324.968	118.920	+0.050	118.970	−0.018
7-1-32	324.923	118.854	+0.120	118.974	−0.014
7-11-32	325.005	119.004	−0.008	118.996	+0.008
7-11-32	324.997	119.986	+0.005	118.991	+0.003
7-11-32	325.038	119.052	−0.060	118.992	+0.004
(Even temperature 325 C; mean value of pressure at 325 C = P_m = 118.988 atm)					
4-4-32	330.020	126.994	−0.033	126.961	+0.001
4-5-32	330.010	126.980	−0.016	126.964	+0.004
4-7-32	330.009	126.983	−0.015	126.968	+0.008
4-8-32	330.012	126.988	−0.020	126.968	+0.008
4-26-32	330.020	126.990	−0.033	126.957	−0.003
4-26-32	330.033	127.008	−0.054	126.954	−0.006
4-26-32	330.038	127.012	−0.062	126.950	−0.010
4-29-32	330.043	127.030	−0.070	126.960	0.000
7-11-32	330.026	127.002	−0.042	126.960	0.000
7-11-32	330.014	127.982	−0.023	126.959	−0.001
7-11-32	330.032	127.005	−0.052	126.953	−0.007
7-11-32	330.025	127.004	−0.041	126.963	+0.003
(Even temperature 330 C; mean value of pressure at 330 C = P_m = 126.960 atm)					
4-4-32	340.035	144.229	−0.063	144.166	−0.001
4-5-32	340.040	144.243	−0.072	144.171	+0.004
4-7-32	340.041	144.231	−0.074	144.157	−0.010
4-8-32	340.015	144.197	−0.027	144.170	+0.003
4-28-32	340.021	144.208	−0.038	144.170	+0.003
4-28-32	340.025	144.204	−0.045	144.159	−0.008
4-29-32	340.028	144.213	−0.051	144.162	−0.005
7-11-32	340.026	144.216	−0.047	144.169	+0.002
7-11-32	340.023	144.211	−0.042	144.169	+0.002
7-11-32	339.994	144.161	+0.011	144.172	+0.005
7-11-32	340.028	144.222	−0.051	144.171	+0.004
(Even temperature 340 C; mean value of pressure at 340 C = P_m = 144.167 atm)					
3-29-32	349.998	163.208	+0.004	163.212	+0.007
3-29-32	349.996	163.209	+0.008	163.217	+0.012
3-30-32	349.989	163.183	+0.022	163.205	0.000
3-30-32	350.110	163.421	−0.220	163.201	−0.004
4-1-32	350.089	163.382	−0.178	163.204	−0.001
4-5-32	350.057	163.324	−0.114	163.210	+0.005
4-7-32	349.999	163.198	+0.002	163.200	−0.005
4-8-32	350.026	163.254	−0.052	163.202	−0.003
4-8-32	350.043	163.282	−0.086	163.196	−0.009
4-8-32	350.060	163.332	−0.120	163.212	+0.007
4-27-32	350.001	163.215	−0.002	163.213	+0.008
4-27-32	350.018	163.225	−0.036	163.189	−0.016
7-11-32	350.027	163.262	−0.054	163.208	+0.003
7-11-32	350.009	163.218	−0.018	163.200	−0.005
7-11-32	350.020	163.246	−0.040	163.206	+0.001
7-12-32	350.025	163.254	−0.050	163.204	−0.001
7-12-32	349.995	163.197	+0.010	163.207	+0.002
7-12-32	350.006	163.214	−0.012	163.202	−0.003
(Even temperature 350 C; mean value of pressure at 350 C = P_m = 163.205 atm)					
7-12-32	355.037	173.552	−0.078	173.474	+0.001
7-12-32	355.006	173.475	−0.013	173.462	−0.011
7-12-32	355.022	173.521	−0.046	173.475	+0.002
7-13-32	354.988	173.449	+0.025	173.474	+0.001
7-13-32	354.966	173.402	+0.072	173.474	+0.001
7-13-32	355.013	173.507	−0.027	173.480	+0.007
(Even temperature 355 C; mean value of pressure at 355 C = P_m = 173.473 atm)					
4-29-32	360.008	184.311	−0.018	184.293	−0.004
7-12-32	360.029	184.365	−0.064	184.301	+0.004
7-12-32	359.996	184.291	+0.009	184.300	+0.003
7-12-32	360.004	184.308	−0.009	184.299	+0.002
7-13-32	359.976	184.241	+0.053	184.294	−0.003
7-13-32	359.959	184.208	+0.091	184.299	+0.002
7-13-32	359.996	184.285	+0.009	184.294	−0.003
(Even temperature 360 C; mean value of pressure at 360 C = P_m = 184.297 atm)					
7-12-32	361.993	188.770	+0.016	188.786	−0.001
7-12-32	361.990	188.765	+0.023	188.788	+0.001
7-12-32	361.973	188.729	+0.061	188.790	+0.003
7-12-32	362.007	188.803	−0.016	188.787	0.000
7-13-32	361.981	188.741	+0.043	188.784	−0.003
7-13-32	361.965	188.708	+0.079	188.787	0.000
7-13-32	361.992	188.770	+0.018	188.788	+0.001
(Even temperature 362 C; mean value of pressure at 362 C = P_m = 188.787 atm)					
7-12-32	363.996	193.367	+0.009	193.376	−0.002
7-12-32	363.978	193.323	+0.051	193.374	−0.004
7-12-32	364.009	193.401	−0.021	193.380	+0.002
7-13-32	363.972	193.314	+0.065	193.379	+0.001
7-13-32	363.958	193.282	+0.097	193.379	+0.001
7-13-32	363.981	193.338	+0.044	193.382	+0.004
(Even temperature 364 C; mean value of pressure at 364 C = P_m = 193.378 atm)					
7-12-32	366.019	198.110	−0.045	198.065	+0.001
7-12-32	365.992	198.038	+0.019	198.057	−0.007
7-12-32	366.032	198.139	−0.076	198.063	−0.001
7-12-32	366.034	198.143	−0.081	198.062	−0.002
7-12-32	366.022	198.119	−0.052	198.067	+0.003
7-13-32	365.951	197.949	+0.116	198.065	+0.001
7-13-32	365.938	197.917	+0.147	198.064	0.000
7-13-32	366.004	198.078	−0.009	198.069	+0.005
(Even temperature 366 C; mean value of pressure at 366 C = P_m = 198.064 atm)					
7-13-32	368.009	202.876	−0.022	202.854	−0.003
7-13-32	367.988	202.831	+0.029	202.860	+0.003
7-13-32	368.030	202.938	−0.073	202.865	+0.008
7-14-32	368.047	202.972	−0.114	202.858	+0.001
7-14-32	368.040	202.947	−0.097	202.850	−0.007
7-14-32	368.018	202.899	−0.044	202.855	−0.002
(Even temperature 368 C; mean value of pressure at 368 C = P_m = 202.857 atm)					
4-29-32	370.014	207.801	−0.035	207.766	−0.005
4-29-32	370.009	207.788	−0.022	207.766	−0.005
4-29-32	370.010	207.792	−0.025	207.767	−0.004
4-29-32	370.020	207.820	−0.049	207.771	0.000
4-29-32	370.035	207.857	−0.087	207.770	−0.001
7-13-32	369.959	207.667	+0.101	207.768	−0.003
7-13-32	370.017	207.829	−0.042	207.787	+0.016
7-13-32	370.034	207.862	−0.084	207.778	+0.007
7-13-32	370.013	207.810	−0.032	207.778	+0.007
7-13-32	370.056	207.924	−0.138	207.786	+0.015
7-13-32	370.041	207.870	−0.101	207.769	−0.002
7-13-32	370.012	207.804	−0.030	207.774	+0.003
7-13-32	370.006	207.795	−0.015	207.780	+0.009
7-14-32	369.983	207.723	+0.042	207.765	−0.006
7-14-32	370.016	207.812	−0.040	207.772	+0.001
7-14-32	370.010	207.789	−0.025	207.764	−0.007
7-14-32	369.986	207.726	+0.035	207.761	−0.010
7-14-32	370.017	207.808	−0.042	207.766	−0.005
7-14-32	370.001	207.766	−0.002	207.764	−0.007
(Even temperature 370 C; mean value of pressure at 370 C = P_m = 207.771 atm)					
7-14-32	370.952	210.147	+0.120	210.267	−0.003
7-14-32	370.966	210.182	+0.085	210.267	−0.003
7-14-32	371.091	210.511	−0.227	210.284	+0.014
7-14-32	371.082	210.480	−0.205	210.275	+0.005
7-14-32	371.046	210.389	−0.115	210.274	+0.004
7-14-32	371.038	210.360	−0.095	210.265	−0.005
7-14-32	371.015	210.315	−0.037	210.278	+0.008
7-14-32	371.011	210.297	−0.027	210.270	0.000
7-15-32	370.953	210.149	+0.117	210.266	−0.004
7-15-32	370.914	210.047	+0.215	210.262	−0.008
7-15-32	370.900	210.022	+0.250	210.272	+0.002
7-15-32	370.892	209.987	+0.270	210.257	−0.013
(Even temperature 371 C; mean value of pressure at 371 C = P_m = 210.270 atm)					
7-15-32	371.993	212.780	+0.018	212.798	+0.003
7-15-32	372.004	212.806	−0.010	212.796	+0.001
7-15-32	372.027	212.866	−0.068	212.798	+0.003
7-15-32	372.014	212.831	−0.035	212.796	+0.001
7-15-32	372.020	212.848	−0.051	212.797	+0.002
7-15-32	372.016	212.846	−0.040	212.806	+0.011
7-22-32	372.070	212.966	−0.177	212.789	−0.006
7-22-32	372.072	213.074	−0.182	212.792	−0.003
7-22-32	372.098	213.033	−0.248	212.785	−0.010
(Even temperature 372 C; mean value of pressure at 372 C = P_m = 212.795 atm)					
7-15-32	373.012	215.393	−0.031	215.362	−0.001
7-15-32	373.012	215.391	−0.031	215.360	−0.003
7-15-32	373.003	215.373	−0.008	215.365	+0.002
7-15-32	373.005	215.367	−0.013	215.354	−0.009
7-15-32	372.987	215.330	+0.033	215.363	0.000
7-15-32	372.983	215.318	+0.043	215.361	−0.002
7-15-32	372.974	215.294	+0.066	215.360	−0.003
7-15-32	372.974	215.299	+0.066	215.365	+0.002
7-15-32	372.996	215.361	+0.010	215.371	+0.008
7-22-32	373.059	215.515	−0.151	215.364	+0.001
7-22-32	373.029	215.447	−0.074	215.373	+0.010
7-22-32	373.008	215.381	−0.020	215.361	−0.002
(Even temperature 373 C; mean value of pressure at 373 C = P_m = 215.363 atm)					
7-22-32	374.001	217.985	−0.003	217.982	−0.003
7-22-32	373.998	217.980	+0.005	217.985	0.000
7-22-32	373.992	217.956	+0.021	217.977	−0.008
7-22-32	373.999	217.971	+0.003	217.974	−0.011
7-22-32	373.978	217.913	+0.057	217.970	−0.015
7-22-32	373.957	217.859	+0.111	217.970	−0.015
7-25-32	374.039	218.091	−0.101	217.990	+0.005
7-25-32	373.971	217.912	+0.078	217.990	+0.005
7-25-32	373.948	217.855	+0.134	217.989	+0.004
7-25-32	374.037	218.090	−0.096	217.994	+0.009
7-25-32	374.052	218.135	−0.134	218.001	+0.016
7-25-32	374.044	218.106	−0.114	217.992	+0.007
7-25-32	374.069	218.169	−0.178	217.991	+0.006
(Even temperature 374 C; mean value of pressure at 374 C = P_m = 217.985 atm)					

TABLE 2 SATURATION PRESSURE OF WATER VAPOR

(Results and formulation of observations)

Temp, θ deg C	Saturation pressure: Mean observed P_m Standard atmospheres	Saturation pressure: Calculated[1] P atmospheres	Deviations: $P_m - P$ (Int.)	Deviations: $\frac{P_m - P}{P} \times 10^4$ Parts in 10,000	Derivative[2]: Calculated $dP/d\theta$ Atm per deg C
100	.....	.....	.....	...	0.03569
110	1.4143	1.4138	+0.0005	+3.5	0.04750
120	1.9598	1.9593	+0.0005	+2.5	0.06208
130	2.6658	2.6658	0.0000	.0	0.07978
140	3.5661	3.5664	−0.0003	−0.8	0.10095
150	4.6969	4.6977	−0.0008	−1.7	0.12598
160	6.0998	6.1000	−0.0002	−0.3	0.15521
170	7.8178	7.8171	+0.0007	+0.9	0.18899
180	9.8958	9.8962	−0.0004	−0.4	0.22768
190	12.3887	12.3881	+0.0006	+0.5	0.27161
200	15.3472	15.3468	+0.0004	+0.3	0.32110
210	18.8300	18.8296	+0.0004	+0.2	0.37647
220	22.8964	22.8969	−0.0005	−0.2	0.43805
230	27.6117	27.6122	−0.0005	−0.2	0.5061
240	33.0416	33.0421	−0.0005	−0.2	0.5810
250	39.2566	39.2563	+0.0003	+0.1	0.6630
260	46.3286	46.3280	+0.0006	+0.1	0.7526
270	54.3333	54.3339	−0.0006	−0.1	0.8500
275	58.7125	58.7122	+0.0003	+0.1	0.9016
280	63.3558	63.3529	+0.0029	+0.5	0.9553
290	73.4779	73.4723	+0.0056	+0.8	1.0701
300	84.7969	84.7881	+0.0088	+1.0	1.1947
310	97.4062	97.4015	+0.0047	+0.5	1.3297
320	111.418	111.420	−0.002	−0.2	1.4760
325	118.988	118.994	−0.006	−0.5	1.5538
330	126.960	126.964	−0.004	−0.3	1.6349
340	144.167	144.168	−0.001	−0.1	1.8086
350	163.205	163.200	+0.005	+0.3	2.0016
355	173.473	173.470	+0.003	+0.2	2.1076
360	184.297	184.290	+0.007	+0.4	2.2220
362	188.787	188.782	+0.005	+0.3	2.2705
364	193.378	193.373	+0.005	+0.3	2.3208
366	198.064	198.067	−0.003	−0.2	2.3732
368	202.857	202.867	−0.010	−0.5	2.4277
370	207.771	207.781	−0.010	−0.5	2.4846
371	210.270	210.279	−0.009	−0.4	2.5141
372	212.795	212.808	−0.013	−0.6	2.5442
373	215.363	215.367	−0.004	−0.2	2.5751
374	217.985	217.958	+0.027	+1.2	2.6068

[1] Calculated from the equation

$$\Theta \log_{10} P = a\Theta + b + cx^3 + dx^5 + ex^8$$

where $\Theta = (273.1 + \theta)$, $x = \left(\frac{\Theta^2}{298{,}000} - 1\right)$, and a, b, c, d, and e are constants given in the table below for the two temperature ranges.

[2] Calculated from the equation

$$dP/d\Theta = 2.302585\, P \left(\frac{a - \log_{10} P}{\Theta} + 6cx^2 + 10dx^4 + 12ex^8\right)$$

which is the derivative of the above equation, using the same notation and constants.

TABLE OF CONSTANTS

	Range 100 C–275 C	Range 275 C–374 C
a =	+5.4247285	+5.4231165
b =	−2003.853	−2002.971
c =	+87.880	+109.54
d =	+107.35	−608.22
e =	−96.252	+1399.0

at the 100-deg point is fixed by definition, the formula was made to give the value of exactly 1 atm at this point. It was found necessary to apply the formula over two overlapping temperature ranges in order to secure a satisfactory fit. The agreement of the formula with observation is shown by the differences in columns 4 and 5 and graphically in Fig. 5. Since these differences in no case exceed the amount which experimental errors might cause, the formula is taken as a reliable representation of the aggregate results of the complete series of measurements. This formulation is the basis of a mutually consistent group of tables suitable for use as actual working tables. They are expressed in various appropriate units and arranged in convenient form to provide for intercomparison of current steam tables. The number of significant figures retained may be more than corresponds to the absolute accuracy of measurement, yet it was determined by considerations of consistency and precision of formulation, calculation, conversion of units, and comparisons. The derivative is included because this factor is important for making thermodynamic correlations and for interpolating intermediate values.

For the units which involve the value of the intensity of gravity the internationally accepted value of 980.665 cm per sec per sec or its equivalent in the English system (32.174 ft per sec per sec) has been used. Obviously it would be undesirable to complicate the tabulation further by taking into account the difference of gravity in different localities. It may be regarded as unfortunate that existing engineering practice still retains pressure units which involve local values of gravity, particularly since the difference is usually so small that the change to standard gravity could be made painlessly.

It should not be overlooked that the temperatures given in this group of tables are expressed either on the International centigrade scale or on the fahrenheit scale derived from it. For correlations involving the second law of thermodynamics, the departure of this temperature scale from the absolute or thermodynamic scale must be taken into account if it should be found to be significant.

Table 3 contains the values of the pressure of saturated water vapor and the derivative with respect to temperature at each degree centigrade from 100 deg to 374 deg. Values are given in each of the three units of pressure which are ordinarily used with the centigrade scale. The standard atmosphere is the standard international unit which has been used as the basis of reduction

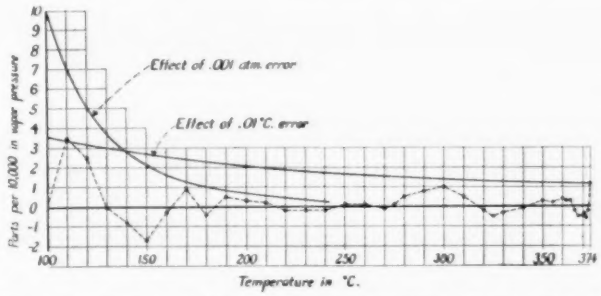


FIG. 5 DEVIATION OF MEAN OBSERVED PRESSURES FROM THE B. S. FORMULATION IN PROPORTIONAL PARTS

of the measurements. The centibar is a decimal subdivision of an internationally recognized unit of pressure.[12] This unit possesses several advantages for use as a practical working standard. It is derived directly from the fundamental units of length, mass, and time, independently of the properties of any substance or of the intensity of gravity. It may therefore be used in thermodynamic calculations with other current cgs units without requiring a conversion factor, a property possessed by no system of thermodynamic units used in current steam tables. This unit is of convenient size for practical use and has a convenient and unequivocal name, which is a unique combination.

The kilogram per square centimeter has a wide usage, particularly abroad and in scientific circles. Its chief disadvantage is that the intensity of gravity must be specified to make the unit definite. A second is that, lacking a name, it has frequently been confused with the atmosphere, having approximately the same value. The three units are used coordinately in this table to facilitate their use interchangeably and give the user a choice of unit appropriate for his purpose.

Table 4 contains values of the pressure of saturated steam in pounds per square inch at each degree fahrenheit. This pressure unit, like the kilogram per square centimeter, is indefinite unless the intensity of gravity is specified. Table 5 contains values of the derivative in the corresponding units.

Tables 6 and 7 are reciprocal to Tables 3 and 4 and express the same relation of vapor pressure to temperature arranged to indicate temperatures corresponding to integral values of saturation pressures.

[12] The "bar," equal to 10^6 dynes per sq cm, was approved as a unit of pressure by the International Meteorological Commission, Rome, 1913.

TABLE 3 PRESSURE OF SATURATED WATER VAPOR

(1 standard atmosphere = 101.325 centibars* = 1.033228 kg per sq cm†)

Temp, θ deg C (Int.)	Pressure P std atm	Deriv. $dP/d\theta$ std atm per deg C	Pressure P centibars	Deriv. $dP/d\theta$ centibars per deg C	Pressure P kg per sq cm	Deriv. $dP/d\theta$ kg per sq cm per deg C	Temp, θ deg C (Int.)
100	1.0000	0.03569	101.325	3.616	1.03323	0.03688	100
101	1.0362	0.03676	104.99	3.725	1.0706	0.03798	101
102	1.0735	0.03785	108.77	3.835	1.1092	0.03911	102
103	1.1119	0.03896	112.66	3.948	1.1488	0.04025	103
104	1.1515	0.04011	116.68	4.064	1.1898	0.04144	104
105	1.1922	0.04128	120.80	4.183	1.2318	0.04265	105
106	1.2341	0.04247	125.05	4.303	1.2751	0.04388	106
107	1.2771	0.04369	129.40	4.427	1.3195	0.04514	107
108	1.3214	0.04493	133.89	4.553	1.3653	0.04642	108
109	1.3670	0.04621	138.51	4.682	1.4124	0.04775	109
110	1.4138	0.04750	143.25	4.813	1.4608	0.04908	110
111	1.4620	0.04883	148.14	4.948	1.5106	0.05045	111
112	1.5115	0.05019	153.15	5.086	1.5617	0.05186	112
113	1.5624	0.05157	158.31	5.225	1.6143	0.05328	113
114	1.6146	0.05298	163.60	5.368	1.6682	0.05474	114
115	1.6683	0.05442	169.04	5.514	1.7237	0.05623	115
116	1.7235	0.05590	174.63	5.664	1.7808	0.05776	116
117	1.7802	0.05740	180.38	5.816	1.8394	0.05931	117
118	1.8383	0.05893	186.27	5.971	1.8994	0.06089	118
119	1.8980	0.06049	192.31	6.129	1.9611	0.06250	119
120	1.9593	0.06208	198.53	6.290	2.0244	0.06414	120
121	2.0222	0.06370	204.90	6.454	2.0894	0.06582	121
122	2.0867	0.06536	211.43	6.623	2.1560	0.06753	122
123	2.1529	0.06704	218.14	6.793	2.2244	0.06927	123
124	2.2208	0.06876	225.02	6.967	2.2946	0.07104	124
125	2.2904	0.07051	232.07	7.144	2.3665	0.07285	125
126	2.3619	0.07230	239.32	7.326	2.4404	0.07470	126
127	2.4351	0.07412	246.74	7.510	2.5160	0.07658	127
128	2.5101	0.07597	254.34	7.698	2.5935	0.07849	128
129	2.5870	0.07786	262.13	7.889	2.6730	0.08045	129
130	2.6658	0.07978	270.11	8.084	2.7544	0.08243	130
131	2.7466	0.08173	278.30	8.281	2.8379	0.08445	131
132	2.8293	0.08372	286.68	8.483	2.9233	0.08650	132
133	2.9140	0.08575	295.26	8.689	3.0108	0.08860	133
134	3.0008	0.08781	304.06	8.897	3.1005	0.09073	134
135	3.0897	0.08991	313.06	9.110	3.1924	0.09290	135
136	3.1806	0.09204	322.27	9.326	3.2863	0.09510	136
137	3.2738	0.09421	331.72	9.546	3.3826	0.09734	137
138	3.3691	0.09642	341.37	9.770	3.4810	0.09962	138
139	3.4666	0.09867	351.25	9.998	3.5818	0.10195	139
140	3.5664	0.10095	361.37	10.229	3.6849	0.10430	140
141	3.6686	0.10328	371.72	10.465	3.7905	0.10671	141
142	3.7730	0.10564	382.30	10.704	3.8984	0.10915	142
143	3.8798	0.10804	393.12	10.947	4.0087	0.11163	143
144	3.9891	0.11048	404.20	11.194	4.1216	0.11415	144
145	4.1008	0.11296	415.51	11.446	4.2371	0.11671	145
146	4.2150	0.11548	427.08	11.701	4.3551	0.11932	146
147	4.3318	0.11805	438.92	11.961	4.4757	0.12197	147
148	4.4511	0.12065	451.01	12.225	4.5990	0.12466	148
149	4.5731	0.12329	463.37	12.492	4.7251	0.12739	149
150	4.6977	0.12598	475.99	12.765	4.8538	0.13017	150
151	4.8251	0.12871	488.90	13.042	4.9854	0.13299	151
152	4.9552	0.13148	502.09	13.322	5.1199	0.13585	152
153	5.0880	0.13429	515.54	13.607	5.2571	0.13875	153
154	5.2238	0.13714	529.30	13.896	5.3974	0.14170	154
155	5.3623	0.14004	543.34	14.190	5.5405	0.14469	155
156	5.5039	0.14298	557.68	14.487	5.6868	0.14773	156
157	5.6483	0.14597	572.31	14.790	5.8360	0.15082	157
158	5.7959	0.14901	587.27	15.098	5.9885	0.15396	158
159	5.9464	0.15209	602.52	15.411	6.1440	0.15714	159
160	6.1000	0.15521	618.08	15.727	6.3027	0.16037	160
161	6.2568	0.15837	633.97	16.047	6.4647	0.16363	161
162	6.4168	0.16159	650.18	16.373	6.6300	0.16696	162
163	6.5800	0.16485	666.72	16.703	6.7986	0.17033	163
164	6.7465	0.16816	683.59	17.039	6.9707	0.17375	164
165	6.9163	0.17151	700.79	17.378	7.1461	0.17721	165
166	7.0895	0.17491	718.34	17.723	7.3251	0.18072	166
167	7.2661	0.17836	736.24	18.072	7.5075	0.18429	167
168	7.4462	0.18185	754.49	18.426	7.6936	0.18789	168
169	7.6298	0.18540	773.09	18.786	7.8833	0.19156	169
170	7.8171	0.18899	792.07	19.149	8.0768	0.19527	170
171	8.0079	0.19263	811.40	19.518	8.2740	0.19903	171
172	8.2023	0.19632	831.10	19.892	8.4748	0.20284	172
173	8.4005	0.20007	851.18	20.272	8.6796	0.20672	173
174	8.6025	0.20386	871.65	20.656	8.8883	0.21063	174
175	8.8083	0.20770	892.50	21.045	9.1010	0.21460	175
176	9.0179	0.21159	913.74	21.439	9.3175	0.21862	176
177	9.2315	0.21554	935.38	21.840	9.5382	0.22270	177
178	9.4490	0.21953	957.42	22.244	9.7630	0.22682	178
179	9.6705	0.22357	979.86	22.653	9.9918	0.23100	179
180	9.8962	0.22768	1002.7	23.070	10.225	0.23525	180
181	10.126	0.23183	1026.0	23.490	10.462	0.23953	181
182	10.360	0.23604	1049.7	23.917	10.704	0.24388	182
183	10.598	0.24029	1073.8	24.347	10.950	0.24827	183
184	10.840	0.24460	1098.4	24.784	11.200	0.25273	184
185	11.087	0.24896	1123.4	25.226	11.455	0.25723	185
186	11.338	0.25338	1148.8	25.674	11.715	0.26180	186
187	11.594	0.25786	1174.8	26.128	11.979	0.26643	187
188	11.854	0.26238	1201.1	26.586	12.248	0.27110	188
189	12.119	0.26697	1228.0	27.051	12.522	0.27584	189
190	12.388	0.27161	1255.2	27.521	12.800	0.28064	190
191	12.662	0.27630	1283.0	27.996	13.083	0.28548	191
192	12.941	0.28105	1311.2	28.477	13.371	0.29039	192
193	13.224	0.28585	1339.9	28.964	13.663	0.29535	193
194	13.512	0.29071	1369.1	29.456	13.961	0.30037	194
195	13.806	0.29564	1398.9	29.956	14.265	0.30546	195
196	14.104	0.30061	1429.1	30.459	14.573	0.31060	196
197	14.407	0.30564	1459.8	30.969	14.886	0.31580	197
198	14.715	0.31074	1491.0	31.486	15.204	0.32107	198
199	15.028	0.31589	1522.7	32.008	15.527	0.32639	199
200	15.347	0.32110	1555.0	32.535	15.857	0.33177	200
201	15.671	0.32637	1587.9	33.069	16.192	0.33721	201
202	16.000	0.33169	1621.2	33.608	16.532	0.34271	202
203	16.334	0.33708	1655.0	34.155	16.877	0.34828	203
204	16.674	0.34252	1689.5	34.706	17.228	0.35390	204
205	17.019	0.34803	1724.5	35.264	17.585	0.35959	205
206	17.370	0.35360	1760.0	35.829	17.947	0.36535	206
207	17.726	0.35922	1796.1	36.398	18.315	0.37116	207
208	18.088	0.36491	1832.8	36.975	18.689	0.37704	208
209	18.456	0.37066	1870.1	37.557	19.069	0.38298	209
210	18.830	0.37647	1907.9	38.146	19.456	0.38898	210
211	19.209	0.38235	1946.4	38.742	19.847	0.39505	211
212	19.594	0.38828	1985.4	39.342	20.245	0.40118	212
213	19.986	0.39428	2025.1	39.950	20.650	0.40738	213
214	20.383	0.40034	2065.3	40.564	21.060	0.41364	214
215	20.786	0.40646	2106.1	41.185	21.477	0.41997	215
216	21.196	0.41265	2147.7	41.812	21.900	0.42636	216
217	21.612	0.41890	2189.8	42.445	22.330	0.43282	217
218	22.034	0.42522	2232.6	43.085	22.766	0.43935	218
219	22.462	0.43160	2276.0	43.732	23.208	0.44594	219
220	22.897	0.43805	2320.0	44.385	23.658	0.45261	220
221	23.338	0.44455	2364.7	45.044	24.113	0.45932	221
222	23.786	0.45113	2410.1	45.711	24.576	0.46612	222
223	24.240	0.45777	2456.1	46.384	25.045	0.47298	223
224	24.702	0.46448	2502.9	47.063	25.523	0.47991	224
225	25.170	0.47125	2550.4	47.749	26.006	0.48691	225
226	25.644	0.47809	2598.4	48.442	26.496	0.49398	226
227	26.126	0.48500	2647.2	49.143	26.994	0.5011	227
228	26.614	0.49197	2696.7	49.849	27.498	0.5083	228
229	27.110	0.49901	2746.9	50.56	28.011	0.5156	229
230	27.612	0.5061	2797.8	51.28	28.529	0.5229	230
231	28.122	0.5133	2849.5	52.01	29.056	0.5304	231
232	28.639	0.5205	2901.8	52.74	29.591	0.5378	232
233	29.163	0.5279	2954.9	53.48	30.132	0.5454	233
234	29.695	0.5352	3008.8	54.23	30.682	0.5530	234

* 1 centibar = 10,000 dynes per sq cm.
† Standard gravity = 980.665 cm per sec per sec.

Temp, θ deg C (Int.)	Pressure P std atm	Deriv. $dP/d\theta$ std atm per deg C	Pressure P centibars	Deriv. $dP/d\theta$ centibars per deg C	Pressure P kg per sq cm	Deriv. $dP/d\theta$ kg per sq cm per deg C	Temp, θ deg C (Int.)
235	30.234	0.5427	3063.5	54.99	31.239	0.5607	235
236	30.780	0.5502	3118.8	55.75	31.803	0.5685	236
237	31.334	0.5578	3174.9	56.52	32.375	0.5763	237
238	31.896	0.5655	3231.9	57.30	32.956	0.5843	238
239	32.465	0.5732	3289.5	58.08	33.544	0.5923	239
240	33.042	0.5810	3348.0	58.87	34.140	0.6003	240
241	33.627	0.5889	3407.3	59.67	34.744	0.6085	241
242	34.220	0.5968	3467.3	60.47	35.357	0.6167	242
243	34.821	0.6049	3528.2	61.29	35.978	0.6250	243
244	35.430	0.6130	3589.9	62.11	36.607	0.6333	244
245	36.047	0.6211	3652.5	62.93	37.245	0.6418	245
246	36.672	0.6293	3715.8	63.77	37.891	0.6503	246
247	37.305	0.6377	3779.9	64.61	38.545	0.6589	247
248	37.947	0.6460	3845.0	65.46	39.208	0.6675	248
249	38.598	0.6545	3910.9	66.32	39.881	0.6763	249
250	39.256	0.6630	3977.6	67.18	40.560	0.6851	250
251	39.924	0.6717	4045.3	68.06	41.251	0.6940	251
252	40.600	0.6803	4113.8	68.93	41.949	0.7029	252
253	41.284	0.6891	4183.1	69.82	42.656	0.7120	253
254	41.978	0.6979	4253.4	70.72	43.373	0.7211	254
255	42.680	0.7068	4324.6	71.62	44.098	0.7303	255
256	43.392	0.7158	4396.7	72.53	44.834	0.7396	256
257	44.112	0.7249	4469.6	73.45	45.578	0.7490	257
258	44.841	0.7340	4543.5	74.38	46.331	0.7584	258
259	45.580	0.7433	4618.4	75.31	47.095	0.7680	259
260	46.328	0.7526	4694.2	76.25	47.867	0.7776	260
261	47.085	0.7619	4770.9	77.20	48.650	0.7873	261
262	47.852	0.7714	4848.6	78.16	49.442	0.7970	262
263	48.628	0.7809	4927.2	79.13	50.244	0.8069	263
264	49.414	0.7906	5006.9	80.10	51.056	0.8168	264
265	50.209	0.8002	5087.4	81.08	51.877	0.8268	265
266	51.014	0.8100	5169.0	82.08	52.709	0.8369	266
267	51.829	0.8199	5251.6	83.08	53.551	0.8471	267
268	52.654	0.8298	5335.2	84.08	54.404	0.8574	268
269	53.489	0.8399	5419.8	85.10	55.266	0.8678	269
270	54.334	0.8500	5505.4	86.12	56.139	0.8782	270
271	55.189	0.8602	5592.0	87.16	57.023	0.8887	271
272	56.054	0.8704	5679.7	88.20	57.917	0.8994	272
273	56.930	0.8808	5768.4	89.25	58.822	0.9101	273
274	57.816	0.8912	5858.2	90.31	59.737	0.9209	274
275	58.712	0.9018	5949.0	91.37	60.663	0.9317	275
276	59.619	0.9120	6040.9	92.41	61.600	0.9423	276
277	60.536	0.9227	6133.8	93.49	62.547	0.9533	277
278	61.464	0.9335	6227.8	94.58	63.506	0.9645	278
279	62.403	0.9444	6323.0	95.69	64.477	0.9757	279
280	63.353	0.9553	6419.2	96.80	65.458	0.9871	280
281	64.314	0.9664	6516.6	97.92	66.451	0.9985	281
282	65.286	0.9775	6615.1	99.05	67.455	1.0100	282
283	66.269	0.9888	6714.7	100.19	68.471	1.0216	283
284	67.263	1.0001	6815.4	101.34	69.498	1.0333	284
285	68.269	1.0116	6917.4	102.50	70.537	1.0452	285
286	69.286	1.0231	7020.4	103.67	71.588	1.0571	286
287	70.315	1.0347	7124.7	104.84	72.651	1.0691	287
288	71.356	1.0464	7230.1	106.03	73.727	1.0812	288
289	72.408	1.0582	7336.7	107.22	74.814	1.0934	289
290	73.472	1.0701	7444.6	108.43	75.913	1.1057	290
291	74.548	1.0821	7553.6	109.64	77.025	1.1181	291
292	75.637	1.0942	7663.9	110.87	78.150	1.1306	292
293	76.737	1.1064	7775.4	112.11	79.287	1.1432	293
294	77.850	1.1188	7888.2	113.36	80.437	1.1560	294
295	78.974	1.1312	8002.0	114.62	81.598	1.1688	295
296	80.112	1.1437	8117.3	115.89	82.774	1.1817	296
297	81.262	1.1563	8233.9	117.16	83.962	1.1947	297
298	82.424	1.1690	8351.6	118.45	85.163	1.2078	298
299	83.600	1.1818	8470.8	119.75	86.378	1.2211	299
300	84.788	1.1947	8591.1	121.05	87.605	1.2344	300
301	85.989	1.2077	8712.8	122.37	88.846	1.2478	301
302	87.204	1.2209	8835.9	123.71	90.102	1.2615	302
303	88.431	1.2341	8960.3	125.05	91.369	1.2751	303
304	89.672	1.2474	9086.0	126.39	92.652	1.2888	304

Temp, θ deg C (Int.)	Pressure P std atm	Deriv. $dP/d\theta$ std atm per deg C	Pressure P centibars	Deriv. $dP/d\theta$ centibars per deg C	Pressure P kg per sq cm	Deriv. $dP/d\theta$ kg per sq cm per deg C	Temp, θ deg C (Int.)
305	90.926	1.2609	9213.1	127.76	93.947	1.3028	305
306	92.194	1.2744	9341.6	129.13	95.257	1.3167	306
307	93.475	1.2881	9471.4	130.52	96.581	1.3309	307
308	94.770	1.3019	9602.6	131.92	97.919	1.3452	308
309	96.079	1.3157	9735.2	133.31	99.272	1.3594	309
310	97.402	1.3297	9869.3	134.73	100.64	1.3739	310
311	98.738	1.3438	10,005	136.16	102.02	1.3885	311
312	100.09	1.3581	10,142	137.61	103.42	1.4032	312
313	101.45	1.3724	10,279	139.06	104.82	1.4180	313
314	102.83	1.3868	10,419	140.52	106.24	1.4329	314
315	104.23	1.4014	10,561	142.00	107.69	1.4480	315
316	105.64	1.4161	10,704	143.49	109.15	1.4632	316
317	107.06	1.4309	10,848	144.99	110.62	1.4784	317
318	108.50	1.4458	10,994	146.50	112.11	1.4938	318
319	109.95	1.4609	11,141	148.03	113.60	1.5094	319
320	111.42	1.4760	11,290	149.56	115.12	1.5250	320
321	112.90	1.4913	11,440	151.11	116.65	1.5409	321
322	114.40	1.5068	11,592	152.68	118.20	1.5569	322
323	115.92	1.5223	11,746	154.25	119.77	1.5729	323
324	117.45	1.5380	11,901	155.84	121.35	1.5891	324
325	118.99	1.5538	12,057	157.44	122.94	1.6054	325
326	120.55	1.5697	12,215	159.05	124.56	1.6219	326
327	122.13	1.5858	12,375	160.68	126.19	1.6385	327
328	123.73	1.6020	12,537	162.32	127.84	1.6552	328
329	125.34	1.6184	12,700	163.98	129.50	1.6722	329
330	126.96	1.6349	12,864	165.66	131.18	1.6892	330
331	128.61	1.6516	13,031	167.35	132.88	1.7065	331
332	130.27	1.6684	13,200	169.05	134.60	1.7238	332
333	131.94	1.6853	13,369	170.76	136.32	1.7413	333
334	133.64	1.7024	13,541	172.50	138.08	1.7590	334
335	135.35	1.7197	13,714	174.25	139.85	1.7768	335
336	137.08	1.7371	13,890	176.01	141.63	1.7948	336
337	138.82	1.7547	14,066	177.80	143.43	1.8130	337
338	140.59	1.7724	14,245	179.59	145.26	1.8313	338
339	142.37	1.7905	14,426	181.42	147.10	1.8500	339
340	144.17	1.8086	14,608	183.26	148.96	1.8687	340
341	145.98	1.8269	14,791	185.11	150.83	1.8876	341
342	147.82	1.8455	14,978	187.00	152.73	1.9068	342
343	149.68	1.8642	15,166	188.89	154.65	1.9261	343
344	151.55	1.8831	15,356	190.81	156.59	1.9457	344
345	153.44	1.9023	15,547	192.75	158.54	1.9655	345
346	155.36	1.9217	15,742	194.72	160.52	1.9856	346
347	157.29	1.9413	15,937	196.70	162.52	2.0058	347
348	159.24	1.9611	16,135	198.71	164.53	2.0263	348
349	161.21	1.9812	16,335	200.75	166.57	2.0470	349
350	163.20	2.0016	16,536	202.81	168.62	2.0681	350
351	165.21	2.0222	16,740	204.90	170.70	2.0894	351
352	167.24	2.0431	16,946	207.02	172.80	2.1110	352
353	169.30	2.0643	17,154	209.17	174.93	2.1329	353
354	171.37	2.0858	17,364	211.34	177.06	2.1551	354
355	173.47	2.1076	17,577	213.55	179.23	2.1776	355
356	175.59	2.1298	17,792	215.80	181.42	2.2006	356
357	177.73	2.1523	18,008	218.08	183.64	2.2238	357
358	179.89	2.1751	18,227	220.39	185.87	2.2474	358
359	182.08	2.1984	18,449	222.75	188.13	2.2714	359
360	184.29	2.2220	18,673	225.14	190.41	2.2958	360
361	186.52	2.2460	18,899	227.58	192.72	2.3206	361
362	188.78	2.2705	19,128	230.06	195.05	2.3459	362
363	191.06	2.2954	19,359	232.58	197.41	2.3717	363
364	193.37	2.3208	19,593	235.16	199.80	2.3979	364
365	195.71	2.3467	19,830	237.78	202.21	2.4247	365
366	198.07	2.3732	20,069	240.46	204.65	2.4521	366
367	200.45	2.4001	20,311	243.19	207.11	2.4799	367
368	202.87	2.4277	20,556	245.99	209.61	2.5084	368
369	205.31	2.4558	20,803	248.83	212.13	2.5374	369
370	207.78	2.4846	21,052	251.75	214.68	2.5672	370
371	210.28	2.5141	21,307	254.74	217.27	2.5976	371
372	212.81	2.5442	21,563	257.79	219.88	2.6287	372
373	215.37	2.5751	21,822	260.92	222.53	2.6607	373
374	217.96	2.6068	22,085	264.13	225.20	2.6934	374

TABLE 4 PRESSURE OF SATURATED WATER VAPOR

(Pounds* per square inch)

Temp θ deg F	0	1	2	3	4	5	6	7	8	9
	Pressure (lb per sq in.)									
210	....	....	14.696	14.990	15.288	15.592	15.900	16.213	16.532	16.857
220	17.186	17.521	17.862	18.206	18.556	18.911	19.273	19.640	20.014	20.393
230	20.777	21.167	21.565	21.969	22.377	22.793	23.215	23.642	24.076	24.518
240	24.966	25.420	25.882	26.349	26.824	27.305	27.795	28.290	28.794	29.305
250	29.823	30.348	30.877	31.420	31.969	32.525	33.088	33.660	34.243	34.829
260	35.425	36.028	36.641	37.262	37.892	38.530	39.177	39.834	40.498	41.172
270	41.853	42.545	43.246	43.957	44.677	45.406	46.144	46.892	47.651	48.419
280	49.198	49.986	50.784	51.593	52.412	53.243	54.083	54.933	55.793	56.665
290	57.549	58.444	59.349	60.266	61.194	62.133	63.085	64.046	65.020	66.006
300	67.006	68.015	69.038	70.074	71.120	72.181	73.251	74.336	75.433	76.546
310	77.669	78.805	79.957	81.119	82.296	83.484	84.691	85.909	87.141	88.386
320	89.646	90.922	92.209	93.514	94.829	96.162	97.510	98.873	100.25	101.64
330	103.05	104.47	105.91	107.36	108.84	110.32	111.83	113.34	114.88	116.43
340	118.00	119.58	121.18	122.80	124.44	126.09	127.76	129.45	131.15	132.87
350	134.62	136.37	138.15	139.94	141.75	143.58	145.44	147.31	149.19	151.10
360	153.02	154.97	156.93	158.91	160.91	162.94	164.98	167.04	169.13	171.23
370	173.35	175.50	177.67	179.85	182.06	184.29	186.53	188.81	191.10	193.41
380	195.74	198.10	200.48	202.89	205.32	207.76	210.24	212.72	215.24	217.78
390	220.34	222.92	225.54	228.18	230.84	233.50	236.21	238.95	241.70	244.48
400	247.29	250.11	252.97	255.85	258.75	261.67	264.63	267.61	270.63	273.66
410	276.73	279.81	282.92	286.05	289.22	292.43	295.65	298.90	302.16	305.47
420	308.82	312.17	315.56	318.98	322.43	325.89	329.40	332.93	336.50	340.09
430	343.70	347.35	351.03	354.74	358.47	362.25	366.06	369.90	373.76	377.65
440	381.58	385.53	389.52	393.53	397.60	401.68	405.79	409.94	414.12	418.33
450	422.57	426.86	431.17	435.53	439.90	444.32	448.77	453.24	457.76	462.31
460	466.90	471.51	476.17	480.86	485.59	490.35	495.15	499.98	504.85	509.76
470	514.69	519.68	524.69	529.76	534.84	539.96	545.12	550.32	555.57	560.86
480	566.17	571.52	576.91	582.36	587.83	593.33	598.88	604.46	610.08	615.77
490	621.48	627.23	633.03	638.86	644.72	650.63	656.59	662.60	668.64	674.72
500	680.84	687.01	693.21	699.46	705.76	712.09	718.47	724.90	731.36	737.88
510	744.44	751.03	757.67	764.35	771.10	777.87	784.71	791.57	798.50	805.46
520	812.46	819.52	826.62	833.77	840.96	848.20	855.50	862.84	870.23	877.66
530	885.12	892.65	900.23	907.85	915.54	923.26	931.04	938.87	946.74	954.68
540	962.64	970.67	978.74	986.87	995.04	1003.3	1011.6	1019.9	1028.3	1036.7
550	1045.2	1053.8	1062.4	1071.0	1079.7	1088.5	1097.3	1106.2	1115.1	1124.1
560	1133.2	1142.3	1151.4	1160.6	1169.9	1179.2	1188.6	1198.0	1207.5	1217.0
570	1226.6	1236.3	1246.0	1255.8	1265.7	1275.6	1285.5	1295.5	1305.6	1315.8
580	1326.0	1336.3	1346.6	1357.0	1367.4	1377.9	1388.5	1399.1	1409.8	1420.6
590	1431.4	1442.3	1453.3	1464.3	1475.3	1486.4	1497.6	1508.9	1520.3	1531.8
600	1543.3	1554.8	1566.4	1578.1	1589.8	1601.6	1613.4	1625.4	1637.4	1649.5
610	1661.6	1673.8	1686.1	1698.6	1711.1	1723.6	1736.2	1748.8	1761.4	1774.2
620	1787.0	1800.0	1813.1	1826.2	1839.4	1852.6	1865.9	1879.3	1892.8	1906.3
630	1919.9	1933.5	1947.3	1961.2	1975.1	1989.1	2003.2	2017.3	2031.5	2045.8
640	2060.2	2074.7	2089.3	2104.0	2118.7	2133.4	2148.3	2163.4	2178.4	2193.6
650	2208.8	2224.1	2239.5	2255.0	2270.6	2286.3	2302.0	2317.8	2333.8	2349.8
660	2366.0	2382.2	2398.4	2414.7	2431.2	2447.8	2464.5	2481.3	2498.1	2515.1
670	2532.1	2549.3	2566.6	2584.0	2601.4	2618.9	2636.6	2654.3	2672.3	2690.3
680	2708.3	2726.5	2744.8	2763.2	2781.7	2800.3	2819.0	2838.0	2857.0	2876.2
690	2895.5	2914.8	2934.1	2953.6	2973.4	2993.3	3013.3	3033.4	3053.6	3073.9
700	3094.4	3115.0	3135.8	3156.7	3177.7	3198.9	....	....	....	....

* Standard gravity = 32.174 ft per sec per sec.

TABLE 6 PRESSURE OF SATURATED WATER VAPOR

(Even pressures in pounds* per square inch with corresponding temperatures in degrees fahrenheit)

Pressure P lb per sq in.	Temp θ deg F	Pressure P lb per sq in.	Temp θ deg F	Pressure P lb per sq in.	Temp θ deg F	Pressure P lb per sq in.	Temp θ deg F
15	213.034	95	324.129	325	424.742	950	538.412
16	216.321	96	324.879	330	426.170	960	539.670
17	219.436	97	325.623	335	427.580	970	540.917
18	222.405	98	326.361	340	428.975	980	542.155
19	225.247	99	327.093	345	430.357	990	543.383
20	227.963	100	327.819	350	431.721	1000	544.600
21	230.572	102	329.257	355	433.070	1020	547.012
22	233.076	104	330.671	360	434.404	1040	549.386
23	235.493	106	332.062	365	435.722	1060	551.726
24	237.826	108	333.434	370	437.026	1080	554.033
25	240.075	110	334.786	375	438.319	1100	556.305
26	242.253	112	336.118	380	439.599	1120	558.544
27	244.367	114	337.431	385	440.866	1140	560.753
28	246.415	116	338.725	390	442.120	1160	562.935
29	248.405	118	340.002	395	443.362	1180	565.086
30	250.340	120	341.263	400	444.590	1200	567.211
31	252.224	122	342.507	405	445.808	1220	569.309
32	254.056	124	343.734	410	447.014	1240	571.381
33	255.841	126	344.946	415	448.209	1260	573.428
34	257.584	128	346.143	420	449.394	1280	575.449
35	259.288	130	347.326	425	450.566	1300	577.444
36	260.954	132	348.495	430	451.729	1320	579.414
37	262.580	134	349.649	435	452.881	1340	581.362
38	264.170	136	350.790	440	454.023	1360	583.290
39	265.728	138	351.917	445	455.153	1380	585.197
40	267.251	140	353.033	450	456.275	1400	587.082
41	268.746	142	354.137	455	457.388	1420	588.946
42	270.214	144	355.228	460	458.493	1440	590.789
43	271.650	146	356.304	465	459.588	1460	592.615
44	273.060	148	357.370	470	460.674	1480	594.421
45	274.444	150	358.426	475	461.749	1500	596.208
46	275.806	152	359.469	480	462.817	1520	597.972
47	277.143	154	360.504	485	463.875	1540	599.718
48	278.456	156	361.528	490	464.927	1560	601.446
49	279.747	158	362.542	495	465.969	1580	603.162
50	281.018	160	363.545	500	467.004	1600	604.863
51	282.268	162	364.538	510	469.049	1620	606.546
52	283.499	164	365.521	520	471.064	1640	608.215
53	284.708	166	366.496	530	473.049	1660	609.866
54	285.902	168	367.462	540	475.008	1680	611.499
55	287.078	170	368.417	550	476.939	1700	613.114
56	288.238	172	369.364	560	478.841	1720	614.716
57	289.381	174	370.303	570	480.717	1740	616.307
58	290.505	176	371.232	580	482.566	1760	617.885
59	291.616	178	372.153	590	484.395	1780	619.450
60	292.711	180	373.067	600	486.201	1800	621.000
61	293.792	182	373.973	610	487.983	1820	622.529
62	294.859	184	374.872	620	489.742	1840	624.047
63	295.911	186	375.763	630	491.479	1860	625.556
64	296.953	188	376.647	640	493.195	1880	627.052
65	297.980	190	377.522	650	494.892	1900	628.535
66	298.994	192	378.390	660	496.568	1920	630.007
67	299.994	194	379.253	670	498.224	1940	631.468
68	300.985	196	380.110	680	499.863	1960	632.918
69	301.963	198	380.958	690	501.482	1980	634.351
70	302.929	200	381.799	700	503.086	2000	635.773
71	303.886	205	383.868	710	504.671	2050	639.286
72	304.830	210	385.903	720	506.239	2100	642.730
73	305.767	215	387.905	730	507.790	2150	646.108
74	306.692	220	389.868	740	509.324	2200	649.423
75	307.607	225	391.794	750	510.844	2250	652.678
76	308.509	230	393.687	760	512.349	2300	655.873
77	309.405	235	395.549	770	513.837	2350	659.013
78	310.293	240	397.382	780	515.313	2400	662.098
79	311.170	245	399.186	790	516.772	2450	665.132
80	312.037	250	400.961	800	518.216	2500	668.112
81	312.898	255	402.708	810	519.650	2550	671.041
82	313.749	260	404.428	820	521.068	2600	673.922
83	314.595	265	406.123	830	522.474	2650	676.755
84	315.429	270	407.793	840	523.867	2700	679.542

(Continued on following page)

TABLE 5 DERIVATIVE OF THE PRESSURE OF SATURATED WATER VAPOR ($dP/d\theta$)

(Pounds* per square inch per degree fahrenheit)

Temp θ deg F	0	1	2	3	4	5	6	7	8	9
	Derivative (lb per sq in. per deg F)									
210	....	....	0.2914	0.2962	0.3011	0.3061	0.3111	0.3161	0.3212	0.3264
220	0.3317	0.3370	0.3424	0.3478	0.3534	0.3590	0.3646	0.3703	0.3761	0.3819
230	0.3878	0.3938	0.3999	0.4061	0.4123	0.4186	0.4249	0.4313	0.4378	0.4443
240	0.4511	0.4578	0.4646	0.4714	0.4783	0.4854	0.4924	0.4996	0.5068	0.5142
250	0.5216	0.5291	0.5367	0.5443	0.5520	0.5598	0.5677	0.5757	0.5833	0.5919
260	0.6002	0.6085	0.6169	0.6254	0.6339	0.6427	0.6514	0.6602	0.6691	0.6781
270	0.6872	0.6964	0.7057	0.7151	0.7246	0.7341	0.7437	0.7534	0.7633	0.7732
280	0.7832	0.7933	0.8036	0.8139	0.8242	0.8348	0.8454	0.8561	0.8668	0.8778
290	0.8887	0.8998	0.9110	0.9223	0.9337	0.9452	0.9568	0.9686	0.9803	0.9922
300	1.0042	1.0163	1.0286	1.0409	1.0533	1.0659	1.0786	1.0913	1.1042	1.1171
310	1.1302	1.1433	1.1568	1.1702	1.1837	1.1973	1.2111	1.2250	1.2389	1.2531
320	1.2672	1.2816	1.2959	1.3105	1.3252	1.3400	1.3549	1.3699	1.3851	1.4003
330	1.4157	1.4312	1.4468	1.4626	1.4784	1.4944	1.5105	1.5267	1.5430	1.5595
340	1.5761	1.5928	1.6096	1.6267	1.6437	1.6609	1.6783	1.6958	1.7134	1.7311
350	1.7491	1.7670	1.7851	1.8033	1.8217	1.8402	1.8589	1.8777	1.8966	1.9157
360	1.9349	1.9541	1.9736	1.9932	2.0128	2.0326	2.0527	2.0728	2.0931	2.1135
370	2.1340	2.1547	2.1755	2.1965	2.2176	2.2388	2.2601	2.2817	2.3033	2.3251
380	2.3471	2.3691	2.3914	2.4138	2.4363	2.4589	2.4817	2.5046	2.5278	2.5511
390	2.5744	2.5980	2.6216	2.6456	2.6695	2.6936	2.7178	2.7423	2.7669	2.7916
400	2.8165	2.8415	2.8668	2.8921	2.9176	2.9432	2.9690	2.9949	3.0211	3.0473
410	3.0737	3.1004	3.1271	3.1540	3.1810	3.2082	3.2355	3.2631	3.2908	3.3186
420	3.3466	3.3747	3.4031	3.4316	3.4603	3.4891	3.5180	3.5472	3.5764	3.6060
430	3.6355	3.6654	3.6953	3.7254	3.7557	3.7862	3.8168	3.8475	3.8786	3.9097
440	3.9410	3.9724	4.0041	4.0358	4.0678	4.0999	4.1322	4.1648	4.1974	4.2303
450	4.2633	4.2965	4.3298	4.3633	4.3969	4.4308	4.4649	4.4992	4.5336	4.5681
460	4.6030	4.6379	4.6730	4.7083	4.7437	4.7794	4.8152	4.8513	4.8874	4.9238
470	4.9604	4.9971	5.034	5.071	5.108	5.146	5.184	5.221	5.259	5.298
480	5.336	5.375	5.413	5.453	5.492	5.531	5.571	5.610	5.650	5.691
490	5.731	5.771	5.812	5.853	5.894	5.935	5.977	6.018	6.060	6.102
500	6.144	6.187	6.230	6.273	6.316	6.359	6.402	6.446	6.489	6.534
510	6.578	6.622	6.667	6.712	6.757	6.803	6.848	6.894	6.939	6.986
520	7.032	7.079	7.126	7.172	7.219	7.267	7.315	7.363	7.411	7.458
530	7.506	7.554	7.602	7.651	7.700	7.750	7.800	7.850	7.900	7.951
540	8.002	8.053	8.104	8.155	8.207	8.259	8.312	8.364	8.416	8.469
550	8.522	8.575	8.629	8.683	8.737	8.791	8.846	8.901	8.956	9.011
560	9.067	9.123	9.179	9.236	9.292	9.349	9.407	9.464	9.521	9.579
570	9.637	9.695	9.754	9.813	9.872	9.932	9.992	10.052	10.112	10.172
580	10.233	10.294	10.355	10.417	10.479	10.541	10.604	10.667	10.730	10.793
590	10.856	10.920	10.984	11.049	11.114	11.179	11.244	11.310	11.376	11.442
600	11.508	11.575	11.642	11.710	11.778	11.846	11.914	11.982	12.051	12.120
610	12.190	12.260	12.330	12.401	12.472	12.543	12.614	12.686	12.758	12.830
620	12.903	12.976	13.050	13.124	13.198	13.273	13.348	13.424	13.500	13.576
630	13.652	13.729	13.806	13.884	13.962	14.041	14.120	14.199	14.278	14.358
640	14.439	14.520	14.602	14.684	14.766	14.849	14.933	15.017	15.102	15.187
650	15.272	15.357	15.444	15.531	15.619	15.707	15.796	15.886	15.976	16.066
660	16.157	16.249	16.342	16.435	16.529	16.624	16.719	16.815	16.912	17.010
670	17.109	17.208	17.308	17.409	17.511	17.613	17.717	17.822	17.928	18.034
680	18.142	18.251	18.361	18.471	18.582	18.696	18.810	18.925	19.042	19.159
690	19.280	19.401	19.522	19.646	19.771	19.898	20.025	20.155	20.286	20.419
700	20.553	20.690	20.828	20.968	21.111	21.254	....	....	....	....

* Standard gravity = 32.174 ft per sec per sec.

85	316.255	275	409.439	850	525.248	2750	682.283
86	317.074	280	411.061	860	526.614	2800	684.981
87	317.886	285	412.662	870	527.969	2850	687.632
88	318.691	290	414.240	880	529.313	2900	690.241
89	319.488	295	415.799	890	530.649	2950	692.810
90	320.279	300	417.337	900	531.970	3000	695.336
91	321.061	305	418.855	910	533.281	3050	697.822
92	321.838	310	420.354	920	534.579	3100	700.272
93	322.607	315	421.835	930	535.867	3150	702.679
94	323.370	320	423.297	940	537.144	3200	705.052

* Standard gravity = 32.174 ft per sec per sec.

TABLE 7 PRESSURE OF SATURATED WATER VAPOR

(Even pressures in kilograms* per square centimeter with corresponding temperatures in degrees centigrade)

Pressure P kg per sq cm	Temp θ deg C	Pressure P kg per sq cm	Temp θ deg C	Pressure P kg per sq cm	Temp θ deg C	Pressure P kg per sq cm	Temp θ deg C
1.0	99.072	8.0	169.605	55	268.693	120	323.148
1.1	101.764	8.2	170.626	56	269.842	122	324.408
1.2	104.246	8.4	171.629	57	270.974	124	325.654
1.3	106.565	8.6	172.613	58	272.092	126	326.884
1.4	108.739	8.8	173.579	59	273.195	128	328.098
1.5	110.789	9.0	174.526	60	274.285	130	329.298
1.6	112.730	9.2	175.458	61	275.361	132	330.484
1.7	114.575	9.4	176.375	62	276.423	134	331.657
1.8	116.331	9.6	177.276	63	277.472	136	332.816
1.9	118.010	9.8	178.163	64	278.509	138	333.958
2.0	119.617	10	179.035	65	279.534	140	335.088
2.1	121.161	11	183.201	66	280.547	142	336.206
2.2	122.646	12	187.079	67	281.548	144	337.312
2.3	124.076	13	190.708	68	282.537	146	338.406
2.4	125.457	14	194.130	69	283.517	148	339.488
2.5	126.790	15	197.360	70	284.485	150	340.559
2.6	128.083	16	200.429	71	285.441	152	341.616
2.7	129.335	17	203.351	72	286.388	154	342.661
2.8	130.549	18	206.145	73	287.326	156	343.696
2.9	131.730	19	208.819	74	288.252	158	344.722
3.0	132.878	20	211.385	75	289.170	160	345.737
3.1	133.994	21	213.855	76	290.079	162	346.742
3.2	135.082	22	216.234	77	290.978	164	347.738
3.3	136.143	23	218.530	78	291.867	166	348.723
3.4	137.178	24	220.753	79	292.748	168	349.699
3.5	138.188	25	222.905	80	293.621	170	350.664
3.6	139.178	26	224.988	81	294.486	172	351.620
3.7	140.144	27	227.012	82	295.343	174	352.565
3.8	141.089	28	228.979	83	296.191	176	353.502
3.9	142.015	29	230.894	84	297.032	178	354.432
4.0	142.922	30	232.757	85	297.865	180	355.353
4.1	143.810	31	234.572	86	298.690	182	356.263
4.2	144.680	32	236.345	87	299.508	184	357.165
4.3	145.534	33	238.075	88	300.319	186	358.058
4.4	146.374	34	239.766	89	301.123	188	358.943
4.5	147.200	35	241.419	90	301.919	190	359.821
4.6	148.008	36	243.035	91	302.710	192	360.689
4.7	148.802	37	244.617	92	303.493	194	361.551
4.8	149.584	38	246.167	93	304.270	196	362.404
4.9	150.354	39	247.687	94	305.041	198	363.248
5.0	151.110	40	249.176	95	305.804	200	364.084
5.2	152.586	41	250.637	96	306.562	202	364.913
5.4	154.018	42	252.073	97	307.314	204	365.734
5.6	155.409	43	253.483	98	308.060	206	366.550
5.8	156.760	44	254.866	99	308.799	208	367.358
6.0	158.075	45	256.224	100	309.533	210	368.157
6.2	159.354	46	257.562	102	310.986	212	368.949
6.4	160.602	47	258.876	104	312.418	214	369.734
6.6	161.820	48	260.171	106	313.828	216	370.511
6.8	163.008	49	261.443	108	315.214	218	371.280
7.0	164.168	50	262.697	110	316.580	220	372.042
7.2	165.303	51	263.931	112	317.930	222	372.800
7.4	166.412	52	265.149	114	319.264	224	373.553
7.6	167.497	53	266.347	116	320.577	225	373.926
7.8	168.563	54	267.527	118	321.871	...	....

* Standard gravity = 980.665 cm per sec per sec.

VI—ESTIMATION OF ACCURACY

Numerical values of physical quantities derived from measurements are never in exact accord with the true values. No matter how skilful and diligent the experimenter, there is a limit to the precision attained in each separate element of measurement and in the final result. The practical use of physical data is similarly subject to limitations of accuracy. The compiler of tables of properties of steam for practical use is obliged to choose from available experimental sources the definitive values which are to be used as a basis for formulation. This selection may be aided by the experimenter if, in addition to the record of methods, standards, and units used in his measurements, he includes also a judicious appraisal of the accuracy of the results.

By careful study of all the factors which enter into the measurements, an estimate may be made of the magnitude of the systematic error which may still remain in each factor after all known corrections for standards and calibrations have been applied. Having made these preliminary estimates, they may be used in a final estimate of the amount by which the results of the measurements might differ from the truth. Such an analysis has been made for the results of the present investigation by considering every apparent source of error, systematic or accidental.

The sources of systematic error which have been found significant include the calibration of the piston gage, the determination of the pressure corrections for the connecting line between the water sample and the gage, the difference of the scale of the thermometer used from the ideal international standard temperature scale, and the determination of the actual temperature of the free surface of the water sample relative to the thermometer. Each of these factors has been discussed earlier in its proper place. The magnitude of the accidental errors of measurement was deduced from the actual differences in the individual results.

Obviously the final appraisal of accuracy can be only approximate. If the facts were known on which an exact estimate of the systematic errors could be based, corrections could be applied for them and their effect eliminated. The element of judgment must enter in the figure which is deduced to indicate the uncertainty remaining after all known corrections have been applied. As such an approximate appraisal, it is believed that the values formulated for the pressure of saturated water vapor do not differ from the truth by more than 3 parts in 10,000, with the possible exception of the region near the critical where the rapidly changing properties of water make the measurements somewhat less trustworthy.

VII—ACKNOWLEDGMENT

The work presented in this report marks a further step in the effort to establish reliability and accord in steam tables, a project which has been promoted by The American Society of Mechanical Engineers through the material support of the steam-power industries. It is hoped that the results here given may prove an aid in attaining that object.

COMMITTEE PERSONNEL

The members of the Special Research Committee on the Thermal Properties of Steam are as follows:

Alex Dow, Detroit, Mich., Chairman; William L. Abbott, Chicago, Ill., Vice-Chairman; Reginald J. S. Pigott, Pittsburgh, Pa., Secretary; George L. Bourne, New York, N. Y.; Harvey N. Davis, Hoboken, N. J.; Hobart C. Dickinson, Washington, D. C.; Frank Dollin, Philadelphia, Pa.; Arthur M. Greene, Jr., Princeton, N. J.; Robert C. H. Heck, New Brunswick, N. J.; David S. Jacobus, New York, N. Y.; Joseph H. Keenan, Hoboken, N. J.; Frederick G. Keyes, Cambridge, Mass.; Lionel S. Marks, Cambridge, Mass.; Geo. A. Orrok, New York, N. Y.; Ernest L. Robinson, Schenectady, N. Y.

RP-55-5

Strength of Roll Necks

Report of the Special Research Committee on Heavy-Duty Anti-Friction Bearings

By W. TRINKS[1] and J. H. HITCHCOCK,[2] PITTSBURGH, PA.

The increasing use of anti-friction bearings for roll necks, which frequently necessitates a change in the customary proportions of roll body and roll neck, has made the question of roll-neck strength an important one. At the juncture between roll neck and roll body, stresses of high intensity occur which cannot be evaluated by the familiar formula of mechanics, and regarding which few directly applicable data were available. In order to evaluate the concentration of stress at this point, and to indicate the manner in which the stress concentration is affected by varying proportions of roll body, and roll neck, rupture tests were made with model rolls of gypsum plaster, having various proportions and several different sizes and shapes of fillet. The test conditions were made to correspond as closely as possible to those of actual rolls in operation. The experimental results are shown, and are compared with previous data. The effect of stress concentration in rolls made of various materials is also considered.

THE increasing use of anti-friction bearings[3] on the roll necks of rolling mills has given increased importance to the subject of roll-neck strength. Except in four-high and cluster mills, the application of anti-friction bearings frequently necessitates either an increase in roll-body diameter (compared to ordinary mills) or a decrease in roll-neck diameter, or both, in order to provide sufficient space for bearings of adequate capacity. This circumstance is particularly common in the case of two-high and three-high bar mills, where the available space between vertically adjacent roll necks, which is the significant dimension, is usually quite limited. The determining factor in such cases is the extent to which the roll-neck diameter can be decreased, in comparison with the diameter of roll body, without making the roll necks unduly weak.

The fact is generally recognized that a discontinuity of cross-section, such as that which exists at the juncture between roll body and roll neck, produces a concentration of stress at that point which cannot be evaluated by the familiar formula of elementary mechanics. Investigation has shown that the concentration of stress in such cases depends upon the extent and abruptness of the change of cross-section. In other words, the relative proportions of body diameter, neck diameter, and radius of fillet are involved in determining the actual intensity of stress which occurs in the roll neck. The relative length of roll body may also be suspected of having some influence in this connection. The significant factors may be expressed in terms of ratios, as follows:

1 The factor of stress concentration, which is defined as the ratio of the actual maximum stress at the roll-neck fillet, determined experimentally, to the apparent stress at the same point, as computed from the familiar formula of mechanics
2 The ratio of body diameter to neck diameter
3 The ratio of fillet radius to neck diameter
4 The ratio of body length to body diameter.

Because of the fact that the rolls of rolling mills are subjected to reversed bending, in which each element of material undergoes alternate tension and compression, the question of fatigue is also involved. Previous investigations of fatigue failure have shown that the effect of stress concentration in bodies subjected to alternating stress depends not only upon the proportions of the body but also to some extent upon the characteristics of the material. Seely[4] attributes this circumstance to the variable homogeneity of materials. Thus, a given external discontinuity of cross-section is found to have less effect in cast iron, which contains many minute internal discontinuities, than in a relatively homogeneous material such as alloy steel. In determining the strength of roll necks, it is accordingly necessary to consider not only the concentration of stress, but also the capacity of the roll material to withstand repeated alternations of concentrated stress. The factor of stress concentration and the actual maximum stress are considered as being independent of the roll material, while the properties of the material are taken into consideration by using a permissible stress (fatigue strength) which is considerably lower than that used for a static condition of loading.

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[2] Carnegie Institute of Technology. Mr. Hitchcock graduated from Case School of Applied Science, Cleveland, Ohio, in 1925. Since January, 1930, he has served as research associate of the Special Research Committee on Heavy-Duty Anti-Friction Bearings, conducting an investigation into the applicability of anti-friction bearings to roll necks. Mr. Hitchcock is a junior member of the A.S.M.E., and is also secretary of the Executive Committee, Iron and Steel Division.

[3] Anti-friction bearings as applied to roll necks are considered as constituting a class in which the coefficient of friction is less than some relatively low maximum value, perhaps a value of 0.007.

The members of the Special Research Committee on Heavy-Duty Anti-Friction Bearings are:

W. Trinks (Chairman), Professor of Mechanical Engineering, Carnegie Institute of Technology, Pittsburgh, Pa.

H. E. Brunner, Chief Engineer, SKF Industries, Inc., New York, N. Y.

W. R. Clark, Works Manager, Mill Products Division, Bridgeport Brass Company, Bridgeport, Conn.

G. C. Farkell, National Tube Company, Lorain, Ohio.

S. G. Koon, Associate Editor, *The Iron Age*, New York, N. Y.

H. H. Talbot, Chief Engineer, United Engineering & Foundry Co., Pittsburgh, Pa.

S. M. Weckstein, Industrial Equipment Engineer, Timken Roller Bearing Company, Canton, Ohio.

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Contributed by the A.S.M.E. Special Research Committee on Heavy-Duty Anti-Friction Bearings and presented at the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

[4] Seely, *Mech. Eng.*, Sept., 1930.

The stress-concentration factor can be evaluated for certain conditions from the results of numerous fatigue tests, in which cylindrical bodies of various materials, both with and without concentrations of stress, have been tested to failure in reversed bending. The results of these tests, however, are still too few, in comparison with the number of variables involved, to provide conclusive relations. The results of photoelastic observations in parallel-walled specimens have previously been applied to calculations of roll-neck strength, but there is no certainty that the ratio of stress intensity in cylindrical bodies is the same as that observed in specimens of rectangular cross-section. Because of these circumstances, the need for an experimental study of the concentration of stress in roll necks is evident. Tests were accordingly undertaken by the Committee,

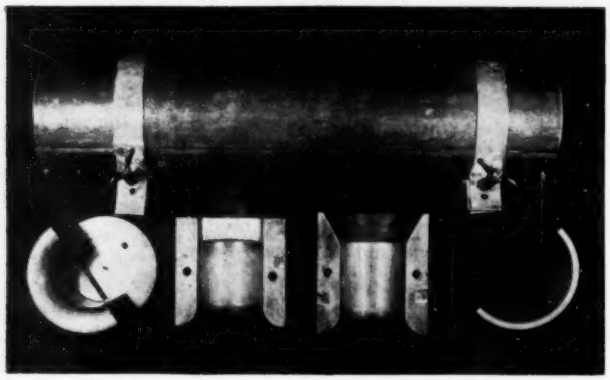


FIG. 1 STEEL MOLD USED FOR CASTING PLASTER MODEL ROLLS

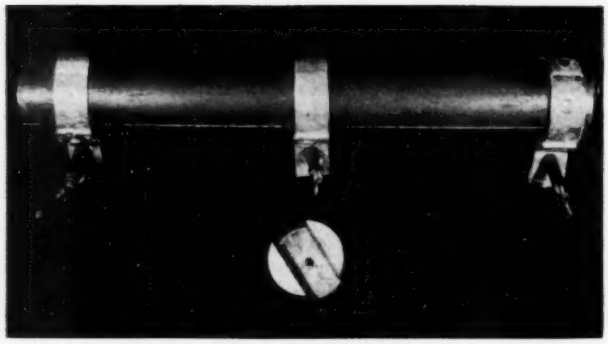


FIG. 2 CYLINDRICAL STEEL MOLD IN WHICH PLASTER COMPARISON SPECIMENS WERE CAST

in order to evaluate the stress-concentration factor for the common proportions of roll body, roll neck, and fillet radius which are encountered in practice. The influence of fillets of special design upon the concentration of stress was also investigated. The results of these tests were then compared with the results of previous fatigue tests, in order that the characteristics of different materials might be taken into consideration.

EXPERIMENTAL METHOD

The failure of a ductile material under a condition of alternating stress takes place in exactly the same manner as the failure of a brittle material under a single application of stress. Rupture tests of models made of a brittle material may accordingly be used to indicate the strength of ductile materials which are subjected to alternating stress, in so far as the strength is a function of the shape and proportions. This method of investigation has previously been used with marked success, a notable instance being the series of tests made by Seely and James[5] on models of curved beams. The conventional procedure is to construct two specimens from the same brittle material; one having the more or less complicated shape in which the stress intensity is to be determined, and the other having a simple shape in which the stress can be computed directly. Seely and James describe the ideal material for this purpose as one in which the strength in compression and shear are very high in comparison to the strength in tension. Pottery plaster was recommended as fulfilling this qualification, and this material was accordingly selected for the tests which are to be described.

PREPARATION OF TEST SPECIMENS

The customary limiting proportions of rolls for bar mills, in which the question of stress concentration is of greatest importance, are as follows:

	Minimum	Maximum
Ratio of body diameter to neck diameter...	1.65	2.00
Ratio of fillet radius to neck diameter.....	0.05	0.20
Ratio of body length to body diameter.....	1.45	2.50

Based upon these proportions, the dimensions of the plaster-model rolls were selected as follows:

Body diameter, inches.....	$3\frac{1}{2}$
Neck diameter, inches.....	2, $1\frac{3}{4}$
Fillet radius, inches.....	$\frac{3}{4}$, $\frac{1}{2}$, $\frac{1}{4}$, $\frac{1}{8}$, $\frac{1}{16}$
Length of roll body, inches.....	$8\frac{1}{2}$, 7, 6, 5

In order that the plaster model rolls might have smooth, uniform, and seamless surfaces, they were cast slightly oversize and machined to the proper dimensions after casting. A steel mold was constructed with a body diameter of $3\frac{5}{8}$ in., neck diameter of $2\frac{1}{8}$ in., and a nominal body length of $8\frac{1}{2}$ in. The mold was provided with $\frac{3}{4}$-in. fillets at the roll necks, and with facilities for adjusting the length of roll body. A photograph of the parts of the mold is reproduced in Fig. 1. A smaller cylindrical mold, $2\frac{1}{8}$ in. in diameter, shown in Fig. 2, was constructed for the comparison specimens of plaster.

Previous investigators[5, 6, 7] have found that the strength of pottery plaster is affected to a very great extent by the proportions of water and plaster used in preparing the castings, and by the details of treatment both before and after casting. For the preparation and seasoning of the model rolls, the following procedure, based upon previous information, was used throughout. Sufficient plaster and water for one model roll and one comparison specimen were carefully weighed in separate containers, the proportion selected being 40 per cent water and 60 per cent plaster. The plaster was poured into the water rapidly, but with care to avoid entrapping air, and the mixture was allowed to stand untouched for a blending period of ten minutes. After blending, the mixture was stirred by hand for five minutes, to mix the ingredients thoroughly and to remove the small quantities of air which were carried in with the plaster. The mixture was then poured into the molds, which had previously been coated with a thin layer of a water solution of soft soap, and the molds were jarred repeatedly for a few minutes to assist the escape of air bubbles. Agitation of the molds was stopped at the first sign of solidification of the plaster, and they were left motionless until solidification had become complete. After the plaster had set, the castings were removed from the molds and placed in a moist chamber, where they were stored

[5] Fred B. Seely and Richard V. James, "Plaster Model Method of Determining Stress," Univ. of Ill. Eng. Exp. Sta., Bull. No. 195, 1929.

[6] "Fundamental Factors in the Use of Pottery Plaster," U. S. Gypsum Co., *Ceramic Industry*, March, 1925.

[7] Heath, "Plaster and Mold Making," Trans. Ceramic Society, May, 1930.

for forty-eight hours. The castings were then dried for a minimum of ten days in air which was circulated by means of a small blower. The containers for plaster and water were rigorously cleaned after the preparation of each pair of castings, in order to prevent the inclusion of small particles of hardened plaster and other foreign substances. The castings produced in this manner were dense, hard, and apparently homogeneous.

The castings were machined in an ordinary lathe with special tools which had been ground to the required radii. The tolerances in machining, with very few exceptions, were held to ± 0.010 in. in body diameter and body length, and ± 0.005 in. in neck diameter. In addition to the five previously mentioned sizes of plain-radius fillets, several fillets of special design were used. The complete dimensions of these fillets, a majority of which represent miniature reproductions of actual bar-mill rolls, are shown later in Fig. 20. In general, six identical models and six comparison specimens were constructed and tested for each set of dimensions, in order to avoid the effect of possible differences in the density and strength of different specimens. This precaution, as has been pointed out by Seely and James,[5] is essential if reliable results are to be obtained with this method of investigation.

Method of Loading

Effort was made to duplicate actual rolling-mill conditions as nearly as possible in the tests of these plaster models. The rolls and roll necks of rolling mills are subjected to reversed bending stress, as previously noted. One roll neck is also subjected to torsion, the torque being transmitted through the roll neck and part of the roll body to the point of application of the separating force. The opposite roll neck is free from torsion, if the torque which is due to roll-neck friction be neglected. Calculations have indicated that the lever arm of the separating force in bar mills varies from 0.05 to 0.15 times the roll diameter. The roll diameter in this case being $3^1/_2$ in., arrangement was made to apply the force eccentrically at a distance of $^3/_8$ in. (0.375 ÷ 3.5 = 0.11) from the vertical center line through the roll. This eccentricity can be seen in Fig. 3, which illustrates the loading arrangement for the model rolls. One roll neck of each test model was provided with a flat extension to fit a metal yoke which was attached to one roll-neck bearing, and through which the torque was transmitted. The flat extension also served the purpose of driving the casting during machining. The opposite roll-neck bearing was generously lubricated to prevent the application of torsional stress to the corresponding roll neck. By means of this arrangement, a close approximation to average rolling-mill conditions was achieved. The efficacy of the loading arrangement was attested by the fact that nearly every model broke at the neck which was subjected to torsion. The simple arrangement which was used for loading the cylindrical comparison specimens is shown in Fig. 4.

The castings were broken in a small testing machine in the Materials Testing Laboratory at Carnegie Institute of Technology. The stress at which rupture occurred in each comparison specimen was calculated by use of the familiar formula of mechanics involving the breaking load, the span, and the diameter of the specimen. The apparent stress at which rupture occurred in each model roll was calculated in the same manner, except that a graphical analysis was required to locate the point of rupture. The outline of each roll neck and fillet was drawn to a large scale, and the outline of a beam of constant strength having the same length of span as the roll was then drawn tangent to the fillet outline, as shown in Fig. 5. Obviously, the point of tangency in such a drawing represents the point at which the roll neck is apparently weakest, and because the drawings were made on a large scale, the distance from the bearing center line to that point could be determined quite accurately. This distance, of course, was slightly different for each value of fillet radius and neck diameter, and was used to compute the bending moment and the apparent stress at the point of rupture.

Typical drawings of this type are shown in Fig. 5 for plain fillets, and in Fig. 6 for fillets of special design. An interesting example of the effect of stress concentration is provided in the case of fillet No. 8, which has a tapered section leading to a cylindrical shoulder, and which is illustrated in the center section of Fig. 6. The models which were constructed with this fillet broke at the root of the cylindrical shoulder, rather than at the

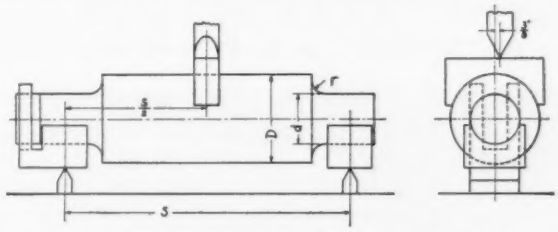


Fig. 3 Arrangement for Loading Model Rolls

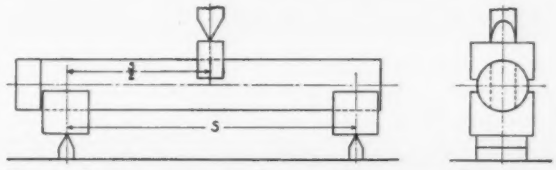


Fig. 4 Arrangement for Loading Comparison Specimens

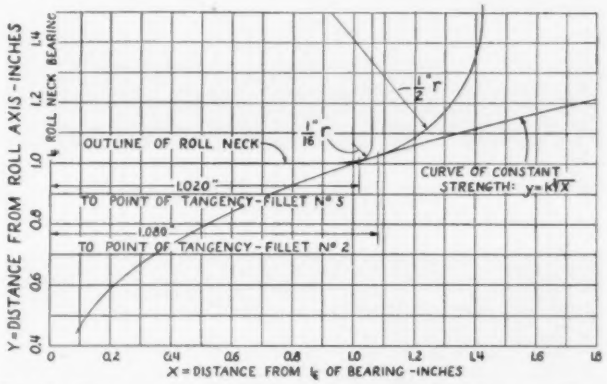


Fig. 5 Graphical Analysis to Locate Weakest Point in Roll Necks Having Plain-Radius Fillets

point of tangency where the roll neck is apparently weakest. This circumstance will be considered in detail later.

The torsional stress in the necks of the model rolls was not considered in determining the apparent stress at which rupture of the necks occurred. The torsional stress here, as in the necks of actual rolling-mill rolls, was very small in comparison with the bending stress, so that the combined stress differed in magnitude from the bending stress by only a very few per cent. The factor of stress concentration, as previously defined, was computed for each model roll as the ratio of the bending stress in the corresponding comparison specimen to the apparent bending stress in the neck of the model roll. The values of this factor in each group of identical models were then averaged. In general, the results obtained with identical models were quite consistent, although widely discrepant results were found in a few cases.

It is interesting to note that every model broke, as nearly as

could be determined, in accordance with the analyses shown in Figs. 5 and 6. These analyses indicate, in the case of plain-radius fillets, that the point of rupture must be located relatively far from the cylindrical portion of the roll neck when the fillet has a large radius, and very near the cylindrical portion when the fillet has a small radius. This contrast could easily be seen by comparison of the ruptured specimens, and may be visible in the accompanying illustrations. Fig. 7 shows a model roll and a comparison specimen, both unbroken; and Fig. 8 an identical pair of castings after rupture. Fig. 9 illustrates a broken roll which had a fillet radius of $^3/_4$ in., while broken rolls which had smaller fillets are illustrated in Figs. 10 and 11. Comparison of these photographs shows that rupture always occurs very near the cylindrical portion of the roll neck, on the side

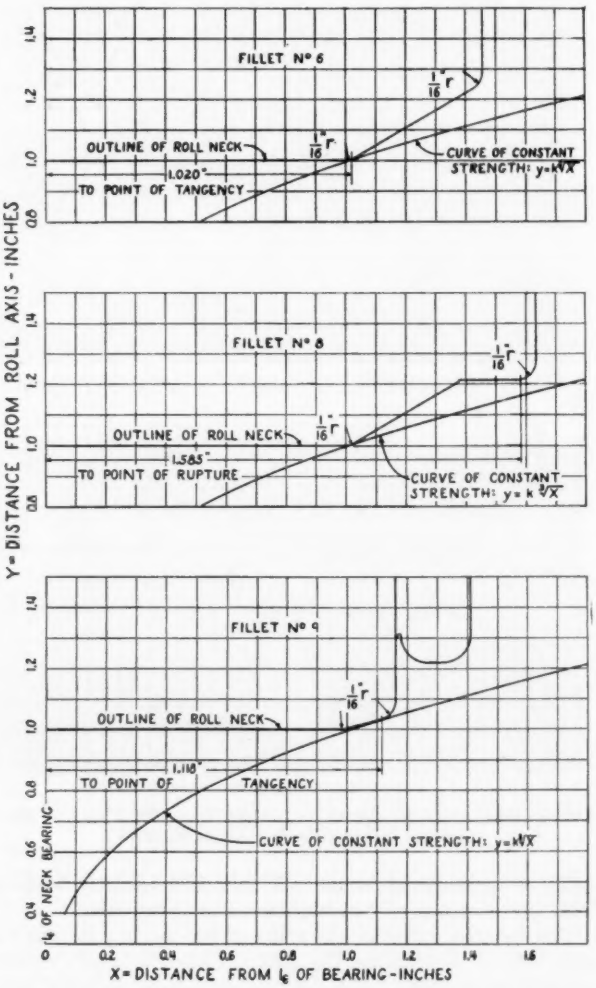


FIG. 6 GRAPHICAL ANALYSIS TO LOCATE WEAKEST POINT IN ROLL NECKS HAVING FILLETS OF SPECIAL DESIGN

which is subjected to tension, and that the point of rupture lies slightly farther from the cylindrical portion of the roll neck when the fillet has a large radius. Typical broken rolls which were constructed with special fillets are shown in Figs. 12–15. Fig. 13 is of particular interest, inasmuch as it confirms the previous statement regarding the location of the point of rupture in fillet No. 8. It is also interesting to note the agreement between Fig. 14 and the lower section of Fig. 6, with respect to the location of the point of rupture in special fillet No. 9.

TEST RESULTS

Study of the test results is facilitated by considering first those which apply to varying length of roll body. These results are shown graphically in Fig. 16; and indicate the somewhat surprising relation, shown by dotted lines, that the factor of stress concentration decreases slightly as the ratio of body length to body diameter increases. It will be observed, however, that the indicated variations of the stress-concentration factor with varying length of roll body are quite small in comparison with the discrepancies between different values in each group. The correctness of the indicated relations must therefore remain open to question. Each solid line of this illustration represents the weighted mean of all observed values in that particular group of tests. The maximum deviation of the indicated relations from the mean values is seen to be only a few per cent. It appears reasonable to assume, therefore, that within the range of the experiments with respect to the roll proportions, the factor of stress concentration is independent of the ratio of body length to body diameter. Subsequent analysis of the results is based upon this assumption.

All of the test results which were obtained with model rolls having plain radius fillets are shown graphically, in averaged form, in Fig. 17. The points marked by circles indicate the observed values of the stress concentration factor for rolls in which the ratio of body diameter to neck diameter was equal to 1.75, while the model rolls in which this ratio was equal to 2.00 provided the points marked by crosses. In accordance with the previous assumption, the small indicated effect of varying length of roll body is neglected, and the curves of this illustration are drawn through the points which correspond to the mean values shown in Fig. 16. With the exception of the points to which question marks are attached, and which will be dealt with later, rather definite relations are indicated between the value of the stress-concentration factor and the ratio of fillet radius to neck diameter. These relations may be expressed empirically by means of the following equation, which has been derived from the experimental results:

$$\eta = 1 + 3.5\ [0.01\ d/r]^{[1.968\ -\ 0.636\ D/d]}$$

In this equation, η represents the stress-concentration factor, d/r the ratio of neck diameter to fillet radius, and D/d the ratio of body diameter to neck diameter. Values of the stress-concentration factor which have been computed from this equation are shown in Fig. 17 by dotted curves. These values coincide with the experimental values for intermediate values of r/d, which is the region of greatest interest, and differ slightly from the latter curves for high and low values of r/d.

The points in Fig. 17 to which question marks are attached represent the results of tests on the first groups of castings which were prepared. Because of the obvious disagreement between these and later results, these tests were repeated with two additional groups of castings having the same dimensions. The supplementary tests provided reasonably consistent results, probably because of improved technic in the preparation of the castings.

For the sake of convenience, the experimental curves shown in Fig. 17 have been replotted in Fig. 18, in terms of the ratio of body diameter to neck diameter. The unbroken portions of the curves of Fig. 18 represent the results within the experimental range, while the dotted portions indicate the probable relations outside the experimental range. It is probable that each of these curves passes through the origin of coordinates, and their slopes indicate that such is the case; nevertheless, the fact should be emphasized that the dotted curves are purely hypothetical, and that the values which they indicate should be used in practice only with caution, if at all. It is improbable,

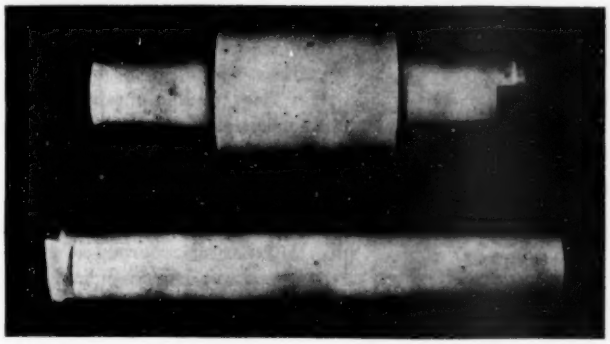


FIG. 7 PLASTER MODEL ROLL AND COMPARISON SPECIMEN AFTER MACHINING

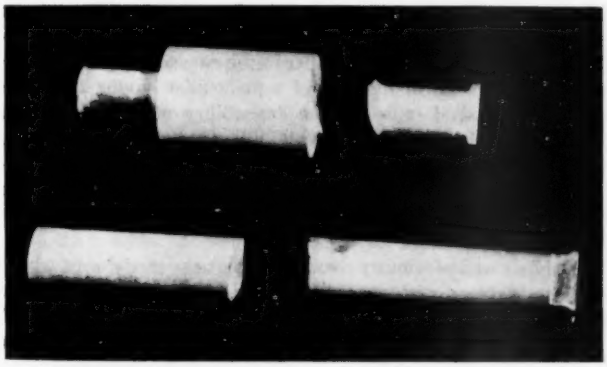


FIG. 8 TYPICAL MODEL ROLL AND COMPARISON SPECIMEN AFTER RUPTURE

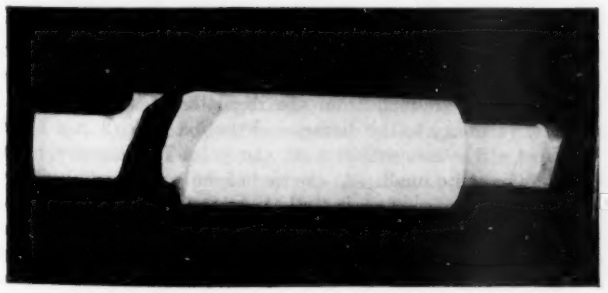


FIG. 9 MODEL ROLL WITH 3/4-IN. FILLET, AFTER RUPTURE

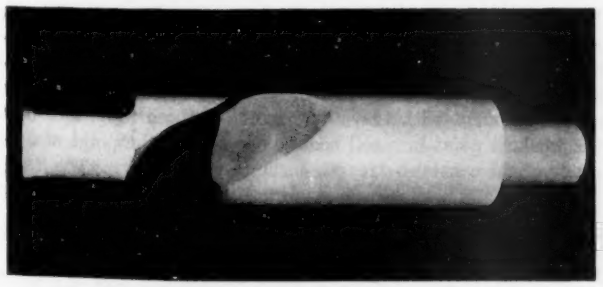


FIG. 10 BROKEN ROLL HAVING 1/2-IN. FILLET

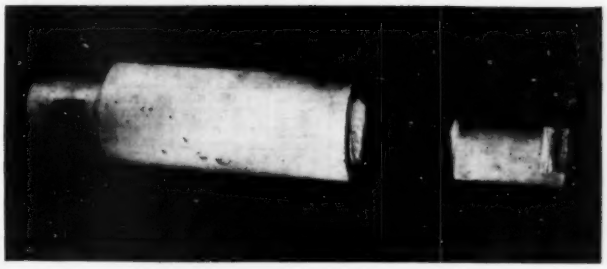


FIG. 11 MODEL ROLL WITH 1/16-IN. FILLET, AFTER RUPTURE



FIG. 12 MODEL ROLL HAVING SPECIAL FILLET NO. 6, AFTER RUPTURE



FIG. 13 MODEL ROLL HAVING SPECIAL FILLET NO. 8, AFTER RUPTURE



FIG. 14 MODEL ROLL HAVING SPECIAL FILLET NO. 9, AFTER RUPTURE

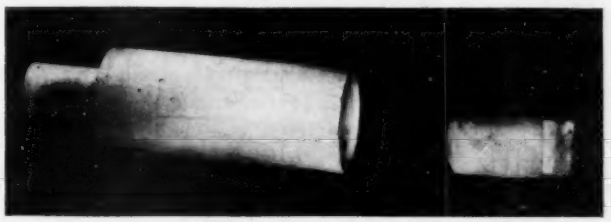


FIG. 15 MODEL ROLL HAVING SPECIAL FILLET NO. 10, AFTER RUPTURE

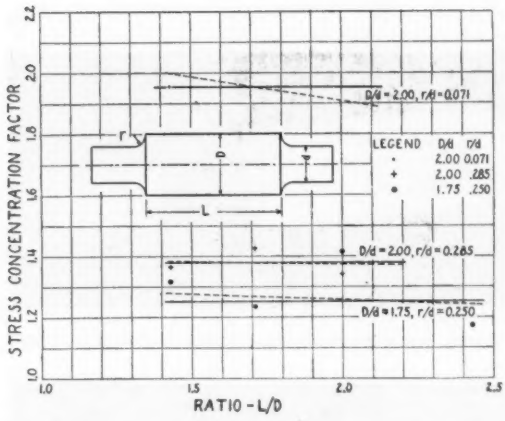


FIG. 16 GRAPHICAL REPRESENTATION OF TEST RESULTS OBTAINED WITH MODEL ROLLS OF DIFFERENT LENGTH
(Dotted lines represent the indicated relations, solid lines represent the mean value of all observations in each group.)

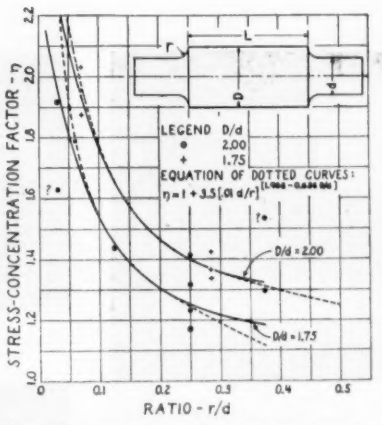


FIG. 17 GRAPHICAL REPRESENTATION OF THE AVERAGED RESULTS OF TESTS MADE WITH MODEL ROLLS HAVING PLAIN-RADIUS FILLETS

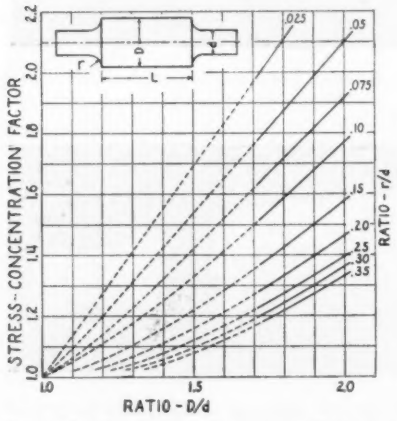


FIG. 18 PROBABLE RELATION BETWEEN STRESS-CONCENTRATION FACTOR AND RATIO OF BODY DIAMETER TO NECK DIAMETER (D/d), FOR VARIOUS VALUES OF RATIO OF FILLET RADIUS TO NECK DIAMETER (r/d)
(Solid curves represent test results; dotted curves indicate probable relations outside experimental range. Use of values shown by dotted curves is not recommended.)

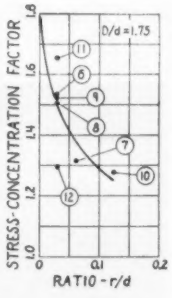


FIG. 19 TEST RESULTS OBTAINED WITH MODEL ROLLS HAVING FILLETS OF SPECIAL DESIGN
(The number at each point represents the type of fillet—see Fig. 20. All model rolls having special fillets were constructed with L/D equal to 2.43, D/d equal to 1.75.)

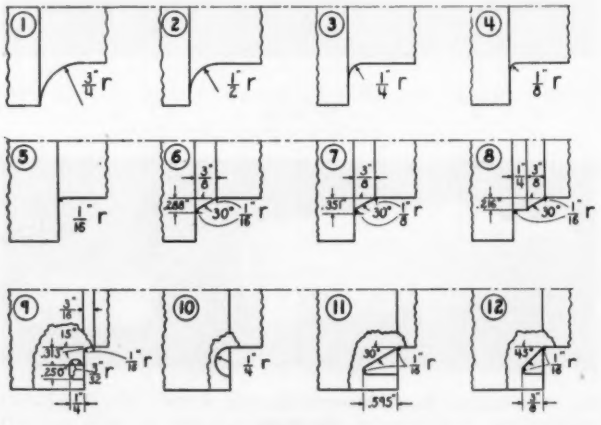


FIG. 20 SHAPES AND DIMENSIONS OF FILLETS USED ON PLASTER MODEL ROLLS

of course, that the proportions of roll bodies and roll necks in practice would require the use of values outside the range of these experiments.

The results obtained with model rolls having fillets of special design are shown graphically in Fig. 19. The number at each point of this illustration represents the type of fillet used on that particular group of specimens, and corresponds to the same number in Fig. 20. All model rolls having special fillets were constructed with the ratio of body length to body diameter equal to 2.43, and with the nominal ratio of body diameter to neck diameter equal to 1.76. The factor of stress concentration, it will be observed, is smaller in the case of any one of these special fillets than in the case of a plain fillet having the same radius. This fact indicates the desirability of using undercut or tapered fillets whenever possible. The use of undercut fillets, such as No. 10 or 12 of Fig. 20, is particularly advantageous because it allows the concentration of stress to be reduced without requiring the roll-neck bearing to be located far from the roll body.

The fact was previously mentioned that the model rolls which were constructed with special fillet No. 8 (see Fig. 20) broke at the root of the cylindrical shoulder rather than at the point of tangency of the constant-strength curve, where the roll neck is apparently weakest. This fact, far from being astonishing, is exactly what should be expected in the light of the other results of these tests. It has already been shown that the value of the stress-concentration factor is dependent upon the extent and abruptness of the change of cross-section. Considering first the point of tangency of the constant-strength curve in fillet No. 8, together with the dimensions of the fillet as given in Fig. 20, it is evident that the change of cross-section at this point is quite gradual, from the diameter of the cylindrical shoulder (2.43 in.) to the diameter of the roll neck (2 in.). The factor of stress concentration at the point of tangency must therefore be quite small. At the root of the cylindrical shoulder, however, the change of cross-section is not only greater in magnitude, but also much more abrupt, so that a higher value of the stress-concentration factor should apply at this point. The ratio of diameters at this section is 3.5/2.43, or 1.44; while the ratio of fillet radius to the smaller diameter is 0.0625/2.43, or 0.026. Applying these values to the curves of Fig. 18, it is observed that the probable value of the stress-concentration factor at this point is 1.58. According to the results shown in Fig. 19, a stress-concentration factor of 1.505 was determined experimentally for fillet No. 8. It is evident, therefore, that the experimental results obtained with rolls having fillet No. 8 are in close agreement with the other results of these tests, and that the weakest point in a roll neck is not always the point of tangency of the constant-strength curve.

COMPARISON OF RESULTS

Because of the fact that rupture of the plaster models was caused in these tests by combined torsion and bending stress, the foregoing results are not precisely comparable with previous data which involve only bending stress. Nevertheless, a comparison with the results of fatigue tests is essential, if the

varying characteristics of different materials are to be taken into consideration, and comparison with previous data obtained by other methods will be interesting. Such a comparison is shown graphically in Fig. 21. The curves A_1 and A_2 of this illustration are taken directly from Fig. 17, and indicate the experimental values which have just been discussed. Curve C indicates the results obtained by Peterson[8] in rupture tests of plaster cylinders in pure bending. Curve B represents the results of photoelastic observations made by Timoshenko and Dietz.[9] The remaining points of this illustration refer to the results of tests made by numerous other investigators with various materials. Data regarding the materials used and the proportions of the specimens in all of these tests are listed in Table 1.

Letter	Test Method	Material	Shape (Side view, End view)	Ratios D/d	Ratios r/d	Ratios L/D	Stress Concentration Factor	Investigator
A_1, A_2	Rupture tests	Pottery plaster		1.75–2.00	0.03–0.37	1.4–2.4	1.2–2.2	Bearings Committee
B	Photo-elastic tests	Celluloid		2.00	0.05–0.50	3.00	1.4–2.4	Timoshenko & Dietz[9]
C	Rupture tests	Pottery plaster		2.00	0.0 –0.50	--	1.2–1.9	Peterson[8]
a	Fatigue tests	Steel, SAE 6130		1.07	0.0259	4.17	2.20	Moore[16]
b	" "	" " "		1.47	0.155	--	1.53	"
c	" "	" " "		1.76	0.307	--	1.31	"
d	Photo-elastic tests	Celluloid		1.154	0.077	--	2.13	Preuss[17]
e	" "	"		1.667	0.333	--	1.77	"
f	" "	"		1.333	0.168	--	2.16	Coker[11]
g	" "	"		2.025	0.271	1.0	1.44	"
h	" "	"		2.025	0.137	1.0	1.58	"
i	Fatigue tests	Steel, 0.49% C.		2.00	0.00	10	2.04	Moore & Kommers[13]
k	" "	" 0.02% C.		2.00	0.00	10	1.92	
m	" "	" 0.45% C.		2.00	0.002	10	1.44	Stanton & Bairstow[12]
n	" "	" 0.33% C.		2.00	0.002	10	1.59–1.80	
p	" "	" 0.33% C.		2.00	0.08	0.4	1.45	Timoshenko & Lessels[15]
q	" "	" "		2.00	0.25	0.4	1.11	
s	" "	" , alloy		2.00	0.25	1.0	1.48	Moore & Jasper[14]
t	" "	" 0.52% C.		2.00	0.25	1.0	1.03	"
t	" "	" 0.37% C.		2.00	0.25	1.0	1.02	"

TABLE 1 RESULTS OF PREVIOUS TESTS ON CONCENTRATION OF STRESS

Examination of Fig. 21 and Table 1 reveals the fact that fatigue tests of high-carbon and alloy steel (points b, c, and s) show approximately the same concentration of stress as the Committee's tests on plaster-model rolls, while the results of fatigue tests on low-carbon steel show somewhat less concentration of stress. It is evident also that photoelastic observations in parallel-walled specimens show greater concentrations of stress than are indicated by other methods. The lack of complete agreement between curves A_2 and C of Fig. 21, which appear to represent closely comparable conditions, may be due to several factors. Values on the former curve were obtained during the rupture of plaster specimens by combined torsion and bending stress, while the latter curve represents values obtained by rupture in pure bending. An equally significant difference is indicated by a comparison of the shapes of the specimens. Those used in tests C were supported and loaded through the portions of large diameter at the ends (see Table 1) so that bending stress only was applied to the specimens at the point of rupture. The model rolls to which curve A_2 refers, on the other hand, were supported by bearings at the roll necks and loaded at the center of the roll body, as shown in Fig. 3, so that the point of rupture was quite near the region of support.

As Frocht[10] has shown by means of photoelastic observations, the stress conditions near a region of loading or support are more or less indeterminate by ordinary methods, depending upon the degree of distribution of the load. Consequently, the conditions of stress at the point of rupture in the model rolls were undoubtedly not the same as those at the point of rupture of the plaster cylinders used in test C, and may have differed somewhat from the assumed conditions upon which curves A_1 and A_2 are based. This latter circumstance leads to the important point that the results of the tests made by the Committee, although perfectly applicable to the rolls of rolling mills, are not precisely applicable to other cases.

Mention should be made of the fact that the absolute dimensions of the roll-neck fillet, quite apart from the relative proportions, may have an influence upon the stress concentration. Earlier experiments with nicked or notched specimens have indicated that a very small notch in a small specimen is more injurious than a larger notch in a larger specimen, the relative proportions being the same in both cases. While this suspected effect cannot be evaluated numerically at present, it is probably safe to assume that the effect is not large in the case of rolls for rolling mills, where the absolute dimensions are usually quite large.

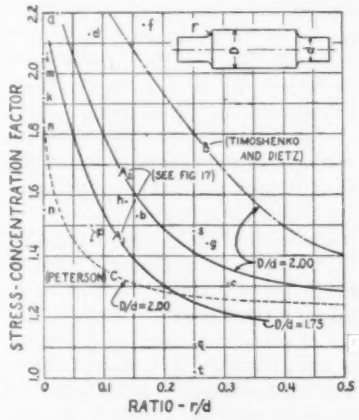


FIG. 21 GRAPHICAL COMPARISON OF RESULTS OBTAINED BY MEANS OF VARIOUS EXPERIMENTAL METHODS

(Data regarding test methods, materials, and proportions of specimens are given in Table 1.)

CONCLUSIONS

The results of these tests have confirmed and extended the conclusions of previous investigators with respect to this experimental method. Rupture tests of plaster models can be made with comparatively small expense; and provide consistent and

[8] R. E. Peterson, "Investigation of Stress Concentration by Means of Plaster Specimens," *Mech. Eng.*, Dec., 1926.

[9] Timoshenko and Dietz, Trans. A.S.M.E., 1925.

[10] Dr. M. M. Frocht, "Kinematography in Photoelasticity;" also "Recent Advances in Photoelasticity," Trans. A.S.M.E., 1931.

[11] Coker, Proc. B.A.A.S., 1924.

[12] Stanton and Bairstow, Proc. Inst. C.E. (British), 1905–6.

[13] Moore and Kommers, Univ. of Ill. Eng. Exp. Sta., Bull. No. 124.

[14] Moore and Jasper, Univ. of Ill. Eng. Exp. Sta., Bull. Nos. 136, 142, 152.

[15] Timoshenko and Lessells, "Applied Elasticity."

[16] R. R. Moore, Proc. A.S.T.M., pt. 2, 1926.

[17] Preuss, *Z.V.D.I.*, p. 1782, 1912; p. 664, 1913.

reliable results if the models are carefully prepared, and if the average of numerous observations on identical models is used. Single observations made with this method may be quite inaccurate.

The relation between the factor of stress concentration in plain fillets and the relative dimensions of roll body and roll neck has been clearly shown, and needs no further comment. With respect to the design of special roll-neck fillets, however, the following additional conclusion may be drawn. When such a fillet contains a tapered portion, the maximum strength for given dimensions of roll body and roll neck can be obtained by making the angle of taper equal to or slightly greater than the slope of the constant-strength curve at the point of tangency (see Fig. 6). Depending upon the proportions of roll body and roll neck, this slope may vary from 15 to 20 deg.

The intensity of stress in roll necks can now be computed, by use of the proper values of the stress-concentration factor. The intensity of stress, as previously stated, is considered as being independent of the characteristics of the roll material. The effect of concentrated and alternating stress, on the other hand, is a function of the fortuitous distribution of weak spots in the roll material. This effect must be taken into consideration in determining the permissible stress for each roll material. The available data which have been derived from fatigue tests are not sufficiently complete to provide final conclusions with respect to all roll materials, but some fairly definite tendencies can nevertheless be pointed out. From the results of previous fatigue tests, represented by the scattered points of Fig. 21, it appears that the permissible stress can safely be made equal to the fatigue strength in the case of alloy- or high-carbon steel rolls. For rolls of a more heterogeneous material such as cast iron, however, in which the effect of concentrated stress is less severe, the permissible stress might exceed the fatigue strength slightly, perhaps by 10 to 20 per cent. More definite conclusions in this respect are still needed, and will doubtless be provided by subsequent fatigue tests.

Concentrations of stress may occur at points in the roll body as well as in the roll-neck fillets, as a result of the grooves which are provided for rolling shapes. As Frocht[10] has shown by means of photoelastic tests, the presence of several notches or grooves close together in a stressed body, such as the numerous grooves in the roll body, introduces only mild stress concentrations in comparison with the concentration of stress caused by a single notch. For this reason, the concentrations of stress in the roll body are usually much less severe than those in the roll neck, but they may be appreciable and should be considered if the strength of the roll body is in question. The ratios of diameters at discontinuities of cross-section in the roll body are usually outside the range of these experiments. In the lack of adequate data which are directly applicable to such cases, prudent use of the extended curves of Fig. 18 may serve for rough approximation. The use of these extended curves, however, is not recommended in general.

The fact was previously mentioned, and should now be emphasized, that the data which have been shown are directly applicable only under certain specific conditions, particularly with regard to the relation between torsion and bending stress. This relation varies in the rolls of rolling mills, and although the variation is usually of very little significance from the standpoint of roll-neck strength, there are some cases, such as the small working rolls of four-high mills, in which torsion should be considered.

Grateful acknowledgment is made to the members of the Committee for their valuable advice in planning and executing the work. Acknowledgment is made also to the Department of Civil Engineering, Carnegie Institute of Technology, for their kind provision of facilities for preparing and testing the plaster specimens.

Discussion

W. R. Clark.[18] Roll designers have known for a long time that a roll having a large fillet where the roll neck joins the body would break less frequently at this point. Many rolls were therefore designed with large fillets, and neck diameters were made large enough, whenever possible, to prevent roll breakage outside the body of the rolls. The recent success of roller bearings on backed-up mills has encouraged attempts to apply roller bearings to two-high mills. This has made it necessary to reduce neck diameters to the point where neck stresses will balance permissible bearing loads, and in cold rolling mills, using chill iron rolls, the load-carrying capacity of the rolls has to be materially reduced or their diameters increased, in order to permit of increased bearing space necessary to carry the original loads. Generally, neck length must be increased, which in turn increases the neck stress near the roll body.

The problem can sometimes be solved by using an alloy-steel roll of the original diameter in place of the chill roll, and sometimes it is possible to apply roller bearings to existing alloy-steel rolls by redesign of the roll neck.

The data which the authors have presented will be of great assistance to roll designers in designing new rolls for roller bearings and in changing over existing rolls. The opportunities to save power and to roll more accurately with reliable anti-friction bearings are great. Recent developments of pressure-measuring gages, which can be readily inserted under rolling-mill screws, make it possible to accurately determine loads imposed on the bearings for the desired rolling operations.

The curves presented in this paper can be used to accurately design the roll necks and bearings for maximum safe loading possible in the space between the rolls, with balanced factors of safety allowed in necks and bearings.

About three years ago we changed over a 12-in. mill, having steel rolls, to roller bearings. In order to carry the same loads as before, it became necessary to reduce the roll neck diameter from 9 in. to 6.3 in., to increase the roll diameter to $12^{7}/_{8}$ in., and to increase the neck length from 7 in. to 11 in., in order to provide for proper bearings and bearing seals. It was found possible to incorporate these seals on a tapered section of the neck 3 in. long, 6.3 in. in diameter at the bearing, and $9^{1}/_{4}$ in. in diameter at the roll body. This was stressed less at the roll body than at the journal end and effectively prevented a weakness of the neck at the roll, due to stress concentration.

This change-over permitted a speed increase of 100 per cent; while the power drawn from the motor was generally less, so that the power cost per ton was cut in half.

S. M. Weckstein.[19] The information in the paper should help considerably in the design of suitable rolls. It would be interesting to obtain information on the same factors as related to built-up rolls. Quite a number of these special rolls are now being used, and information to check their design will be of value.

Authors' Closure

The only discussion calling for a reply is that of Mr. Weckstein, referring to the strength of built-up rolls. The stresses in that type of roll are so complex that their consideration in an appendix to the present paper is out of question. The built-up roll should be treated in a separate paper.

[18] General Works Manager, Bridgeport Brass Company, Bridgeport, Conn. Mem. A.S.M.E.

[19] Industrial Equipment Engineer, Timken Roller Bearing Company, Canton, Ohio. Jun. A.S.M.E.

RP-55-6

An Analysis of Crushing and Pulverizing Processes

By LINCOLN T. WORK,[1] NEW YORK, N. Y.

Different types of crushing and pulverizing mills are mentioned to show the diversity in operating characteristics. It is clear that different types of mills have been commercially adapted to similar products, and only rarely is one type of mill used to the exclusion of all others. The principles laid down illustrate the importance of grindability. A consideration of energy requirements shows that there is a large wastage of power over that needed to produce reduction in size. The earlier work of the author is confirmed on the difference in fineness-distribution curves produced under different grinding conditions and by different types of mills. Proper choice of mill, proper loading, and the application of separators offer the opportunity to produce material of specific size characteristics or to produce material below a given size at a minimum of cost.



Introduction

THERE have been brought together under the title or heading of crushing and pulverization a number of processes for the reduction in size of materials to constitute a basic operation useful to the engineering industries. This title represents the entire range of breaking of materials by machinery from the size of rocks leaving the quarry to the finest powders that are practically colloidal in nature. There are three terms which may be applied specifically to this group of operations; namely, crushing, pulverization, and disintegration. Crushing has been largely used as the terminology of the mining industries and may be used in the general sense of reduction in size of large pieces of material; it produces, essentially, coarser than sieve sizes. Pulverization is the term which may be applied to fine-grinding operations where particles are being broken from relatively fine sizes to powders or dusts. Both of these terms imply the actual fracture of a particle across surfaces which are tightly bonded. Disintegration is a term which may be applied to the reduction in size of particulate groups held together by relatively little force, and it may be said to imply that there is no actual change in the size of the inherent particles in the mass.

Crushing has been developed largely for the processing of minerals. Ores are broken to expose surface for roasting and oxidation and to minimize the amount of penetration needed in leaching, or to isolate individual particles of each kind for separating by flotation. Coal is crushed and sieved to various mesh sizes to render it suitable for firing on grates. Cement rock is crushed as preliminary to its final pulverization. Stone aggregates are crushed to various sizes for use as concrete aggregates or as material for macadam road. Pulverization is also applicable to minerals, but it is equally applicable to other substances which must be reduced in size for any special purpose. Rock minerals are pulverized to make them useful as fillers in rubber and in paints, and to create homogeneity of mixtures for ready reaction in the manufacture of cements, porcelain, and glass. Coal is pulverized to render it suitable for firing as a dust mixed with air. The fused clinker in cement manufacture is pulverized to render it reactive with water. Many other materials are reduced to powder to prepare them for further treatment or to prepare them in a suitable form for handling as a finished product. Disintegration may enter as a part of these crushing operations. In its purest form it may be illustrated by the breaking down of filter cakes such as dyes or pigments. This is accomplished with the same general type of machinery as is used in pulverization, but it is a special case in that the particle shape and size have already been established and the mill is used merely to separate particles from each other. Within recent years there has come into the field of fine sizes the colloid mill for the production of emulsions or of very finely subdivided solids.

The operations of crushing, pulverization, and disintegration utilize equipment of the same general character, specially designed of course to meet the diversified range of product and the varied range of particle sizes. These operations overlap and cannot be strictly differentiated. The nomenclature already defined will be used because different problems are presented, and yet all of these operations can be treated together and they represent problems of a somewhat similar nature. The objective in this reduction of materials may vary. In some cases it is desired to break large pieces to pass through a certain aperture. In other cases it is desired to crush for a maximum amount of a given sieve size. The same general requirements hold in pulverization. Sometimes a product that is uniformly of the same size is desired. In other cases it is desired to eliminate the coarsest particles, merely making the powder finer than a given specified size. In still different cases the best results are secured by a proper blending of coarse and fine sizes. Disintegration usually has for its objective the complete elimination of clusters or agglomerates and the production of individual particles of their own normal size. Thus it is not possible to select any one objective as the aim of grinding operations, but yet there stands out conspicuously from the others the production of surface as a primary objective, particularly in fine sizes.

These operations of crushing and pulverizing involve mechanical loading of a particle to reduce it in size. It is well to consider alternatives which may attain a similar objective. Dean and Gross[2] have investigated expandable substances which could be

[1] Assistant Professor in Chemical Engineering, Columbia University. Lincoln T. Work was Instructor in Chemical Engineering, Columbia University, from 1921 to 1929, since which time he has been in his present position. He has the degrees from Columbia University of A.B., 1918; Ch.E., 1921; A.M., 1924; and Ph.D., 1929. Dr. Work has been a consulting chemical engineer since 1921, his major fields being portland cement, active carbon, pigments, petroleum products, and grinding and particle size. During the summer of 1929 he was with the du Pont company in charge of the formation of the division of fundamental chemical engineering research. He was national secretary-treasurer of Phi Lambda Upsilon, honorary chemical society, from 1923 to 1929, and has been the national president since that time.

Presented at the National Process Meeting, Buffalo, N. Y., June 6 to 8, 1932, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Dean, R. S., and Gross, J., "Explosive Shattering of Minerals as a Substitute for Crushing Preparatory to Ore Dressing," U. S. Bureau of Mines Report of Investigations No. 3118, February, 1932.

placed within the pores of a solid and which would disrupt the body upon rapid expansion. These agents fall into the classes of detonators and of superheated liquids. Their study has involved only the latter. Operating with superheated water at various pressures, they have broken galena of 4 to 6 mesh to produce predominantly an undersize in the range of 6 to 14 mesh with from 0 to 5 per cent under 200 mesh. A number of investigators have used the thermal expansion of the solid itself to produce disintegration. Rapid quenching of a very hot object will produce the stress needed to shatter the larger pieces. In this case, as in the former, the action is not predominantly on the fines, and these methods are likely to produce coarse materials. With high temperatures and a very rapid quenching, the second method may be extended into the sub-sieve range. For the production of very fine materials—in fact, usually finer than any ground product—solution and precipitation methods are most commonly used. For certain purposes spray drying of pulps, rather than mass drying with subsequent grinding, has been utilized to produce certain desired results and properties. With this brief consideration of the alternative methods of reducing particle size, the discussion of crushing will be resumed.

On account of its very complexity, crushing has not been amenable to fundamental scientific analysis. Exact methods of measurement have been lacking until quite recently, and practically every early approach to the subject has been limited by the fact that fineness below sieve size was not accurately evaluated. Furthermore, until recently it has been impossible to evaluate the resistance of any material to grinding, although it has been known that the performance of milling machinery varied with the grindability of the material. In fact, some mills which ground one material very well could not be used to grind another. Machines for crushing have been developed largely by empirical methods and keen observation. There is little tendency to standardize them, although they may be classified roughly into types for purposes of this discussion. The author has offered such a classification[3] into the five following types:

1 Direct-pressure mills—such as jaw crushers, gyratory crushers, stamp mill, and Symmonds cone crusher.

2 Roller mills—such as single- or double-roll crushers, differential rolls, Chilean mill, centrifugal roll mill, ball plate or roller plate mills.

3 Tube mills—such as ball or pebble mill and rod mill.

4 Beater mills—such as impact or hammer type, and cage mills.

5 Revolving plate mills—such as disk mills, buhr stones, and certain colloid mills.

The manner in which the mills apply load and the rate at which the particle is loaded differ between the types and even within the types. Then, too, these mills may be associated with liquid or air separators to remove a certain size of ground material from the finished product, and to return this to the mills for further grinding. While these differences between mills are apparent and will undoubtedly be emphasized in the discussion, it is the primary purpose of this paper to consider the grindability characteristics of materials, basic energy requirements for grinding, and the character of the finished product as it is affected by the entire processing.

Grindability of Materials

The properties of individual materials affect the rate and extent of their pulverization. With the increase in exactness of measurement and with the demands for greater fineness, measurement of the resistance of materials to pulverization has become significant. It is self-evident that a material like talc will grind differently from a material like quartz and that a mill which will grind talc to a certain fineness may not be capable of applying the necessary force to grind silica to the same fineness, or the mill itself may suffer from a prohibitive amount of wear in handling silica which would not be encountered in the grinding of talc. Then there is mica, which due to its plate-like nature offers very special problems in reducing it to thin particles rather than to rounded ones. Grindability of these materials has been largely estimated in the terms of the hardness scale or tested roughly by dropping a ball from progressive heights until the material breaks. A method has been devised for the estimation of the friability of coarse coal by dropping it to a surface of concrete and measuring the fineness of the resulting pieces.[4] Special materials like mica present a case where experience is necessary in the selection of the proper kind of mill to deliver a suitable product. These methods are not adequate. The industry recognizes this and is now calling for mills by number or by dimension rather than in terms of a general capacity figure.

In addition to the differences between different kinds of materials, the problem of grindability is significant within a group of materials. Take coal as an example, and the differences between anthracite and bituminous are sufficient to create a serious problem in the selection of a mill or in the estimation of its capacity. The differences are so pronounced that a mill which will adequately meet the demands with bituminous coal may not even produce a combustible powder when grinding anthracite. In one case, the capacity of the mill was actually reduced more than half by a change from bituminous to anthracite coal. Noticeable differences in mill capacity and in power consumed per ton of product are to be observed with different samples of bituminous coal. It is therefore necessary to evaluate grindability in some quantitative terms. First, it must be known quantitatively whether the material can be ground by a given milling action and at what approximate rate and power consumption, and then it must be known what is the nature of the resulting fineness obtained from this mill action.

Some effort has been made to evaluate grindability by measuring the change in the fineness of a material after a definite mill action. Hardgrove[5] accepts the Martin proof of Rittinger's law that the work done in grinding is proportional to the reciprocal of the particle diameter, and uses a mechanical mortar and pestle or a revolving ball crusher to create surface as a measure of energy. The original material is prepared in the range of 16 to 30 mesh, and the material after crushing is compared by sieving. Many coal and coke samples are compared as to grindabilities, and the values for coal range from the low 20's for anthracite to 119 for bituminous. He has discussed the bearing of grindability on mill capacity, and has indicated that with complete scavenging of fines from the grinding parts, mill capacity is very nearly proportional to grindability.

The author has used a laboratory jar mill for a comparison of this same property in abrasives. In this case it was important to know not only the change in surface, but also how rapidly the coarse material is eliminated and how much fine dust is produced.

TABLE 1 GRINDABILITY DATA ON ABRASIVES, IN PERCENTAGES

Mesh	Sample 1		Sample 2		Sample 3	
	Before	After	Before	After	Before	After
On 16	9.27	3.43	10.55	0.66	12.53	3.46
16–20	89.23	68.18	87.81	21.15	67.12	28.18
20–24	1.23	10.84	1.24	42.24	18.36	31.15
24–35	0.15	6.18	0.17	16.95	1.19	15.50
35–50	0.06	2.46	0.17	5.29	0.48	6.25
Minus 50	0.06	8.91	0.06	13.71	0.32	15.46

[3] Work, L. T., "Crushing and Pulverization," *Industrial and Engineering Chemistry*, vol. 21, p. 498, May, 1929.

[4] Smith, Cloyde M., "An Investigation of the Friability of Different Coals," University of Illinois Engineering Experiment Station Bulletin No. 196, November, 1929.

[5] Hardgrove, R. M., "Grindability of Coal," Trans. A.S.M.E., vol. 54 (1932), paper no. FSP-54-5.

This may be illustrated by data on three samples of abrasive (see Table 1).

The initial materials are essentially of the same size, being chiefly in the 16–20-mesh interval. Sample 2 shows much more pronounced elimination of the coarse material than samples 1 and 3. Yet after grinding it contained less material finer than 50 mesh than did sample 3, which had resisted breaking. Such a method of test might be used to compare several samples as to their probable power consumption in a mill. It would not determine whether a special mill type was suitable for the original material. In a general way, as has been noted here, the differences in friability where coarse particles are broken through the grain and in abrasion where fine grits or impalpable powder is produced may be estimated by an observation of the shape of the final fineness curve. An exception may be noted where sharp materials are being compared with rounded materials as to abrasion resistance. The initial roughness may be significant and the particles may have to be subjected to a preliminary trial grinding to bring them to the same surface condition.

It is not alone in the manner of disintegration that grindability is significant. Mill capacity, power consumption per ton, and the quality of the product may be affected by moisture, the plastic nature of the material, and its susceptibility to heat. These factors may necessitate special types of mills or special processing procedures if they would not altogether prevent the reduction of the materials. Surface moisture increases mill power. For example, a certain pulverizing mill when fed coal of 6 per cent moisture requires an energy consumption of 15 kwhr per ton; when fed coal of 9 per cent moisture, uses 16.5 kwhr per ton; and when fed coal of 12 per cent moisture, uses 18.3 kwhr per ton. Thus moisture correspondingly decreases mill capacity, and it also limits the fineness to which grinding may be carried before there is cloggage of the mill through agglomeration of the particles. Drying is often effective in correcting these troubles. It is frequently done in a preliminary drier, but may also be done by the passage of warm air through the grinding mill. Often it is desirable to add more water and to grind wet. Some materials become plastic with increase in temperature, more particularly those materials which are near their melting points, such as sulphur and asphalt, and their grindability will change rapidly with mill temperature. Salts containing water of crystallization may be decomposed on heating, and the water which is evolved acts to impede crushing. Organic dyes may be injured or decomposed by elevation of temperature during the milling procedure. These difficulties are usually met by the practical common sense of the mill operator, and there is no quantitative method of estimating when they will occur or of evaluating their magnitude. The mill may be chosen of such a type to deliver the desired fineness with a minimum of heat. Refrigeration of the mill has been proposed to increase the grindability and to lessen the effect of heat. The material may be passed through the mill rapidly, so that it does not heat up. Then it may be further cooled, and there may be removed from it the portion which is sufficiently fine, using sieves or air separation. The cooled product or the oversize may be returned to the mill for further reduction.

It may thus be seen that the problem of grindability is one which has not been treated quantitatively, save with the possible exception of mill tests for evaluating this property. There is need for a thorough study and the actual correlation of this with crushing data. Such a study is particularly necessary in those fields where milling costs are a large proportion of the total cost, such as pulverization of fuel and the manufacture of portland cement. Through the present lack of knowledge on that score, purchasers have not been able to specify proper sizes of pulverizers and have depended on the guarantees of the mill manufacturers. As these guarantees have been largely based on experience, there has been considerable uncertainty and grief which might be avoided by quantitative evaluation of the grindability of the material to be processed.

Energy Requirements

The actual energy required for crushing or pulverization depends upon the ease with which the material may be broken, the fineness of the ultimate product, and the effectiveness of the mill in converting its energy input into grinding effort. With respect to the first two of these factors, the inherent nature of the material has been discussed under grindability. In any consideration of crushing, the basic energy required to produce the disintegration is of considerable interest.

For a long period, during which fineness measurements were largely in the range of sieve sizes, a dispute raged as to whether the Rittinger or Kick concepts of breaking energy were correct. Rittinger developed the idea that each new unit of surface produced in grinding required a fixed amount of energy to separate like material from like and to expose this new surface. From this concept he showed mathematically that the work of crushing is proportional to reduction in diameter. While the treatment is illustrated in terms of a cube, irregularities in actual surface of ground material may be taken care of by a surface factor. On the other hand, Kick's concept may be stated as follows: The energy required for producing analogous changes of configuration of geometrically similar bodies of equal technological state varies as the volumes or weights of these bodies. In other words, for any unit weight of particles the energy required to produce a desired reduction in volume of all particles in the mass is constant, no matter what may be the original size of the particles. This relation is based on the idea that elastic deformation of the particle takes place under loading and that the actual energy required to produce sufficient deformation for breakage is the theoretical energy. Stadler[6] and Taggart[7] have supported Kick, while Gates[8] and Bell[9] have supported Rittinger. Gates has developed the crushing-surface diagram to show graphically the relative efficiencies of crushing. The cumulative per cent undersize is plotted against the reciprocal of diameter for both the feed into the mill and the discharge from it. The area between the two curves is a relative measure of the energy expended in crushing, based on developed surface.

With respect to the Kick versus Rittinger dispute, the consensus of opinion among mining men has been formulated to the effect that the Rittinger law applies very closely to the actual performance of pulverizing mills. Kick's law has appeared to measure the performance in coarse grinding. The solution to this question has been regarded as uncertain because the measurements were practically all on sieve sizes, and the assumptions which were made with respect to sub-sieve size were open to great variation. With new tests available, the question has again been revived, and several contributions have recently been made. Martin[10] has compared the energy consumption in a ball mill with the new surface developed. An 18-in. by 18-in. mill with 1-in. steel balls and batch charges of silica sand was used for test-

[6] Stadler, H., "Grinding Analyses and Their Application," Trans. Inst. Mining and Metallurgy (London), vol. 19, p. 471 (1909–1910).

[7] Taggart, A. F., "The Work of Crushing," Trans. Am. Inst. Min. Engrs., vol. 48, p. 153 (1914).

[8] Gates, A. O., "Kick vs. Rittinger, an Experimental Investigation in Rock Crushing, Performed at Purdue University," Trans. Am. Inst. Min. Engrs., vol. 52, p. 875 (1915).

[9] Bell, J. W., "An Investigation on Rock Crushing Made at McGill University," Trans. Am. Inst. Min. Engrs., vol. 57, p. 133 (1917).

[10] Martin, G., Bowes, E. A., and Turner, F. B., "Connection Between the Surface Area Produced and the Work Done in Tube-Mill Grinding of Quartz Sand," Trans. of Ceramic Soc., vol. 25, p. 63 (1925–1926).

ing. A method was devised to measure the rate of solution of the silica in hydrofluoric acid. The energy required for driving the mill with and without a charge of sand was considered as energy input. When energy input and these surface values were compared, he concluded that "in ordinary tube mill grinding the surface produced is accurately proportional to the work done." It has generally been conceded that the actual work of grinding may be but a small portion of the work required of a mill, and to avoid this possible error Gross and Zimmerley[11] have utilized the energy of a freely falling body to crush sand. They have modified the hydrofluoric-acid solution method to compensate for change of surface with time, and they too have found that the new surface produced varies with the energy input. Their conclusion is that, "when crushing quartz, the new surface produced is in direct proportion to the work input in accordance with the Rittinger law of crushing." It should be borne in mind that these crushing methods measure the overall energy accompanying disintegration and that the dissolution test has not yet been rigidly proved to measure surface. It is to be noted that in the exact measurement of surface the same essential method was used by both Martin and by Gross and Zimmerley. It has been suggested by some who have discussed these results that the rate of etching of sand by hydrofluoric acid may vary with the conditions of stress to which the particles have been subjected. These theoretical considerations, therefore, throw some doubt upon the apparent conclusiveness of the Kick versus Rittinger dispute, although a great amount of valuable information has been added in these investigations.

In connection with some recent work, effort has been made to determine the actual energy needed to develop new surface. The assumption is made that the theoretical minimum is the surface energy of the material in its given state of pulverization. Values of surface energy calculated by Edser at 920 ergs per sq cm and by Martin at 310 ergs per sq cm are somewhat divergent. But if these values of surface energy be considered as an order of magnitude, the calculated crushing efficiency is still exceedingly low.

TABLE 2 SURFACE ENERGY AND CRUSHING EFFICIENCY

Observer	Mill	Surface energy — Calculated by	Surface energy — Value, ergs per sq cm	Crushing efficiency, per cent
Gaudin	Rolls	Edser	920	1.3
Martin	Ball	Martin	310	0.06
Gross and Zimmerley	Stamp	Edser	920	3.0
Gross and Zimmerley	Stamp	Martin	310	1.0

The data in Table 2 are by Gross and Zimmerley,[12] and their comments are quoted briefly: "As the values for the surface energy of solids are unsatisfactory at present, some doubt is cast on the value of theoretical efficiencies. The foregoing efficiencies indicate that a large portion of the energy input in the crushing operation is dissipated in the form of heat. . . ."

Fahrenwald, Hammar, Lee, and Staley[13] have discussed the calculation of crushing efficiency and have given a critical treatment of the derivation of surface energy, introducing surface wetting as an important factor in grinding energy. They have developed a thermal method and carried out experiments which indicate mill efficiencies from 6 per cent to 25 per cent, approximately ten times those derived by surface-energy calculations: Even these values still leave from 75 per cent to 94 per cent of the energy input of the mill to be accounted for.

The investigations upon the energy consumption in grinding have shown, first, that overall energy input in crushing quartz is proportional to its rate of dissolution and probably to the new surface developed and, second, that the actual energy needed to develop new surface is a small proportion of that which is applied to the mill to obtain the crushing.

Fineness Characteristics of Ground Materials

Other investigators have studied the fineness of the ultimate product and have measured the fineness below sieve ranges by microscopic or elutriation methods. Thus, in addition to calculations of surface, the nature of the fineness-distribution curve is also known. These investigations have made possible further study of the nature of the crushing action in any mill. Heretofore such terms as compression, beam action, shear, impact, abrasion, attrition, and others have been used to define mill action in cases where no marked characteristic of fineness can be seen in the ultimate product. The fineness-distribution curves which have been obtained after careful measurement are not amenable to simple mathematical analysis, but rather may be treated with probability or statistical formulas. Martin, Blythe, and Tongue[14] have measured with the microscope quartz ground in a ball mill and have proposed an exponential relation of the type $N = ae^{-bx}$, where N equals the number of particles in a unit interval of average diameter x, and a and b are constants for each condition of grinding. Gaudin[15] has reported data on jaw crusher, rolls, rod mill, and ball mill, using quartz, galena and granite, in terms of graded sieve apertures and the percentages included in the sieve intervals. The sieve apertures have followed the Tyler scale, using either the fourth root of 2 or the square root of 2 as the constant interval. Below sieve size he has used sedimentation and has added those values to the sieve values. When per cent in a sieve interval and sieve aperture were plotted on a logarithmic scale, he found a constant paraboloid loop in the coarse range and a straight-line relation in the fine. He has expressed only the latter relation mathematically, as he has used it for calculations into the finer ranges of size:

$$y = Cx^k, \qquad dz = (C/p - 1)x^{k-1}dx$$

In these formulas y is the percentage by weight in the sieve interval x to px mm, and dz represents the amount in a uniform interval from x to $(x + dx)$. C and k are constants. He has noted a consistent exception to this straight-line relation when the materials are not homogeneous. In such a case breakage takes place preponderantly at crystal size, a point which he has checked by measurement of that crystal size in the original rock.

The author has investigated the situation by an experimental study in laboratory equipment and by an analysis of data already in the literature. Tables 3 and 4 present the test data obtained on dry grindings and on grindings in different fluids in a laboratory jar mill. These values were determined by combination sieving, elutriation, and microscopic sizing, and the smoothed distribution curves are presented in Figs. 1, 2, 3, and 4. As these small millings represent qualitatively the general action of a ball mill, there are some observations to be made as to the progress of grinding in such a mill. This mill action involves the gradual

[11] Gross, J., and Zimmerley, S. R., "Crushing and Grinding, II. Relation of Measured Surface of Crushed Quartz to Sieve Sizes," Am. Inst. Min. Engrs., "Milling Methods" (1930), p. 27.

[12] Gross, J., and Zimmerley, S., "Crushing and Grinding, III. Relation of Work Input to Surface Produced in Crushing Quartz," Am. Inst. Min. Engrs., "Milling Methods" (1930), p. 43.

[13] Fahrenwald, A. W., Hammar, G. W., Lee, H. E., and Staley, W. W., "Ball-Mill Studies, II. Thermal Determinations of Ball-Mill Efficiency," Am. Inst. Min. Engrs., Technical Publication no. 416 (1931).

[14] Martin, G., Blythe, C. E., and Tongue, H., "Researches on the Theory of Fine Grinding, I," British Portland Cement Research Assoc. pamphlet No. 4, Trans. of Ceramic Soc. (England), vol. 23, p. 61 (1924).

[15] Gaudin, A. M., "An Investigation of Crushing Phenomena," Am. Inst. Min. Engrs., no. 1665B (February, 1926).

TABLE 3 TEST DATA, DRY GRINDINGS

Fixed factors

Feed { Ottawa sand, 24 to 30 mesh

Mill { 60 pebbles, 3250 g, 60 to 65 rpm

Mill size
Ball charge
Mill speed
Fluid medium—air
Mill charge—silica sand, 24 to 30 mesh

Variable factors

Weight of sand
Time of grinding

Diameter range number	Diameter range, microns	Average diameter, microns	Per cent per micron: Dry ground full charge—weight of sand, 1100 g: Sample 13	Sample 12	Sample 11	Sample 8	Sample 7	Dry ground small charge—weight of sand, 400 g: Sample 204-D	Sample 205-D	Sample 202-D	Sample 207-D
Time of grinding			30 min	1 hr 4 min	2 hr	5 hr 9 min	14 hr 7 min	1 hr	2 hr	5 hr	14 hr
Total revolutions			1756	3873	7219	21,166	55,143	....	....	....	....
No. 26	833 to 701	767.0	0.080	0.014	0.0003	...	....	0.0002	....	....	....
No. 25	701 to 589	645.0	0.390	0.100	0.0047	...	....	0.0005	....	....	....
No. 24	589 to 495	542.0	0.115	0.072	0.0080	...	....	0.0006	....	....	....
No. 23	495 to 417	456.0	0.079	0.103	0.0282	...	....	0.0013	....	....	....
No. 22	417 to 351	384.0	0.069	0.119	0.0568	...	....	0.0034	....	....	....
No. 21	351 to 295	323.0	0.075	0.151	0.127	...	....	0.011	0.0008	....	....
No. 20	295 to 246	271.0	0.060	0.124	0.111	...	....	0.028	0.006	....	....
No. 19	246 to 208	227.0	0.071	0.174	0.211	...	....	0.090	0.006	....	....
No. 18	208 to 175	192.0	0.065	0.152	0.196	...	....	0.142	0.006	....	....
No. 17	175 to 147	161.0	0.079	0.229	0.339	0.007	....	0.303	0.003	....	....
No. 16	147 to 124	136.0	0.049	0.132	0.184	0.007	....	0.254	0.005	....	....
No. 15	124 to 104	114.0	0.080	0.238	0.379	0.049	0.0086	0.472	0.042	....	0.0043
No. 14	104 to 88	96.0	0.072	0.219	0.416	0.165	0.0039	0.428	0.107	0.0014	0.0036
No. 13	88 to 74	81.0	0.037	0.108	0.174	0.187	0.0043	0.308	0.145	0.0034	0.0022
No. 12	74 to 55.5	64.75	0.055	0.239	0.292	0.356	0.000	0.390	0.164	0.000	0.000
No. 11	55.5 to 37	46.25	0.050	0.218	0.300	0.860	0.180	0.427	0.670	0.148	0.000
No. 10	37 to 27.75	32.38	0.078	0.198	0.421	1.18	0.863	1.002	1.635	0.876	0.259
No. 9	27.75 to 18.5	23.13	0.084	0.203	0.581	1.77	1.72	0.890	2.294	2.14	1.508
No. 8	18.5 to 13.875	16.19	0.085	0.173	0.534	1.87	3.08	1.213	2.672	3.16	3.25
No. 7	13.875 to 9.25	11.56	0.072	0.120	0.644	2.28	2.41	1.24	1.945	3.46	3.40
No. 6	9.25 to 6.94	8.09	0.053	0.106	0.765	2.24	3.34	1.52	2.78	3.75	4.33
No. 5	6.94 to 4.625	5.78	0.057	0.417	0.597	2.12	3.17	1.03	2.24	3.50	6.49
No. 4	4.625 to 3.469	4.05	0.096	0.274	0.557	3.05	5.41	1.004	2.09	5.71	7.28
No. 3	3.469 to 2.313	2.89	0.104	0.397	0.448	3.23	6.17	0.783	2.19	4.97	5.81
No. 2+	2.313 to 1.734	2.02	0.171	0.649	0.516	4.02	10.20	0.772	2.94	5.33	5.85
No. 2—	1.734 to 1.156	1.44	0.159	0.625	0.540	2.89	11.10	0.700	1.78	4.98	9.47
No. 1+	1.156 to 0.578	0.87	0.344	0.743	0.507	3.77	9.39	0.865	1.52	6.92	4.45
No. 1—	0.578 to 0.000	0.289	0.017	0.202	0.093	0.47	1.68	0.144	0.19	0.62	0.57

TABLE 4 TEST DATA, FLUID VARIED

Fixed factors

Feed { Ottawa sand, 24 to 30 mesh

Mill { 60 pebbles, 3250 g, 60 to 65 rpm

Mill size
Ball charge
Mill speed
Mill charge—silica sand, 24 to 30 mesh
Weight of sand

Variable factors

Nature of fluid:
Air
Water
Linseed oil
Time of grinding

Diameter range number	Diameter range, microns	Average diameter, microns	Per cent per micron: Dry ground small charge—weight of sand, 400 g: Sample 204-D	Sample 205-D	Sample 202-D	Sample 207-D	Ground in water, charge: 400 g sand, 1200 cc water: Sample 205-W	Sample 102-W	Sample 103-W	Sample 101-W	Ground in oil, charge: 400 g sand, 1200 cc linseed oil: Sample 302-O	Sample 301-O	Sample 301-O	Sample 305-O
Time of grinding			1 hr	2 hr	5 hr	14 hr	1 hr	2 hr	5 hr	14 hr	1 hr	5 hr	14 hr	50 hr
No. 26	833 to 701	767	0.0002	....	....	....	0.0002	...	...	...	0.0009	....	....	...
No. 25	701 to 589	645	0.0005	....	....	....	0.0005	...	...	...	0.0160	....	....	...
No. 24	589 to 495	542	0.0006	....	....	....	0.0010	...	...	...	0.0600	....	....	...
No. 23	495 to 417	456	0.0013	....	....	....	0.0016	0.008	...	...	0.172	....	....	...
No. 22	417 to 351	384	0.0034	....	....	....	0.0031	0.002	...	...	0.207	....	....	...
No. 21	351 to 295	323	0.011	0.0008	....	....	0.0095	0.001	...	...	0.254	0.0019	....	...
No. 20	295 to 246	271	0.028	0.006	....	....	0.024	0.006	...	...	0.191	0.0015	....	...
No. 19	246 to 208	227	0.090	0.006	....	....	0.084	0.006	...	...	0.221	0.0038	....	...
No. 18	208 to 175	192	0.142	0.006	....	....	0.154	0.006	0.012	...	0.149	0.0059	....	0.002
No. 17	175 to 147	161	0.303	0.003	....	....	0.389	0.012	0.005	...	0.221	0.012	....	0.002
No. 16	147 to 124	136	0.254	0.005	....	....	0.308	0.014	0.003	0.012	0.098	0.014	0.0022	0.002
No. 15	124 to 104	114	0.472	0.042	....	0.0043	0.612	0.092	0.007	0.009	0.175	0.045	0.0015	0.003
No. 14	104 to 88	96	0.428	0.107	0.0014	0.0036	0.534	0.183	0.008	0.011	0.144	0.069	0.0040	0.003
No. 13	88 to 74	81	0.308	0.145	0.0034	0.0022	0.373	0.230	0.008	0.008	0.089	0.092	0.0046	0.002
No. 12	74 to 55.5	64.75	0.390	0.164	0.000	0.000	0.614	0.235	0.000	0.000	0.073	0.678	0.350	0.000
No. 11	55.5 to 37	46.25	0.427	0.670	0.148	0.000	0.524	0.720	0.488	0.000	0.139	0.987	0.501	0.000
No. 10	37 to 27.75	32.38	1.002	1.635	0.876	0.259	0.512	2.150	0.966	0.264	0.225	1.06	1.05	0.000
No. 9	27.75 to 18.5	23.13	0.890	2.294	2.14	1.508	0.786	1.783	2.58	1.06	0.188	1.54	1.33	0.644
No. 8	18.5 to 13.875	16.19	1.213	2.672	3.16	3.25	0.941	2.15	3.75	3.39	0.178	2.32	2.35	2.46
No. 7	13.875 to 9.25	11.56	1.24	1.945	3.46	3.40	0.620	2.20	3.29	4.64	0.225	2.49	3.04	2.67
No. 6	9.25 to 6.94	8.09	1.52	2.78	3.75	4.33	0.553	2.08	3.82	5.70	0.198	2.63	4.89	5.66
No. 5	6.94 to 4.625	5.78	1.03	2.24	3.50	6.49	0.498	1.52	2.43	6.54	0.135	1.72	3.91	6.12
No. 4	4.625 to 3.469	4.05	1.004	2.09	5.71	7.28	0.526	1.98	3.51	7.48	0.228	1.80	4.17	5.73
No. 3	3.469 to 2.313	2.89	0.783	2.19	4.97	5.81	0.395	1.60	2.15	4.56	0.174	1.59	3.11	7.53
No. 2+	2.313 to 1.734	2.02	0.772	2.94	5.33	5.85	0.377	1.57	2.30	4.96	0.194	2.23	4.71	13.55
No. 2—	1.734 to 1.156	1.44	0.700	1.78	4.98	9.47	0.364	1.07	1.35	3.04	0.182	1.89	3.08	10.35
No. 1+	1.156 to 0.578	0.87	0.865	1.52	6.92	4.45	0.360	1.22	1.35	3.69	0.264	2.84	4.53	20.40
No. 1—	0.578 to 0.000	0.289	0.144	0.19	0.62	0.57	0.064	0.22	0.26	0.53	0.027	0.17	0.38	1.13

elimination of feed material to produce particles ranging between feed size and zero diameter. As time goes on, the particles are further broken, with the ultimate elimination of the coarser sizes. The drift of the weight curve into the fine regions is progressively slower with time, as the number of breakings to produce a unit shifting is increased.

Differences may be noted in the dry grindings in which the weight of charge is varied. In the first case, 1100 g practically filled all the interstices between the balls. In the D series, the 400 g did not fill the interstices, and the charge was subject to freer ball action and greater ball load. Elimination of coarse material was more rapid in the D runs. However, for equal removal of coarse material, greater quantities of finer material were produced in the first series. (See Figs. 2, 3, and 4, and com-

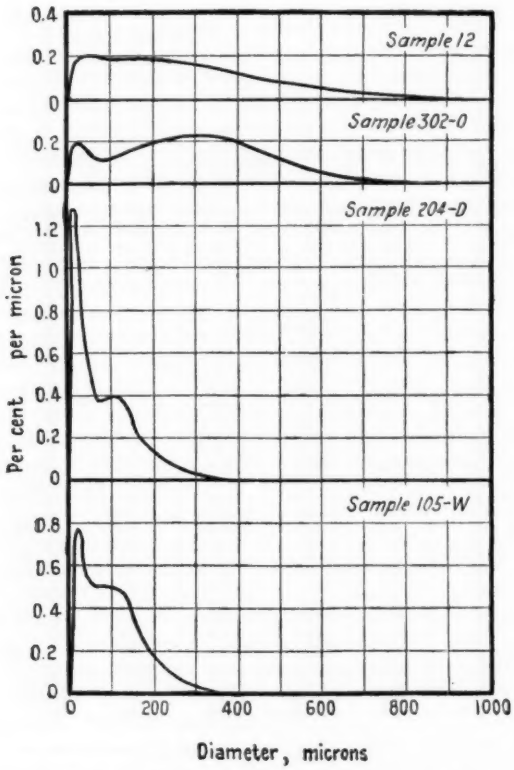


FIG. 1 CURVES FOR SAMPLES GROUND 1 HR

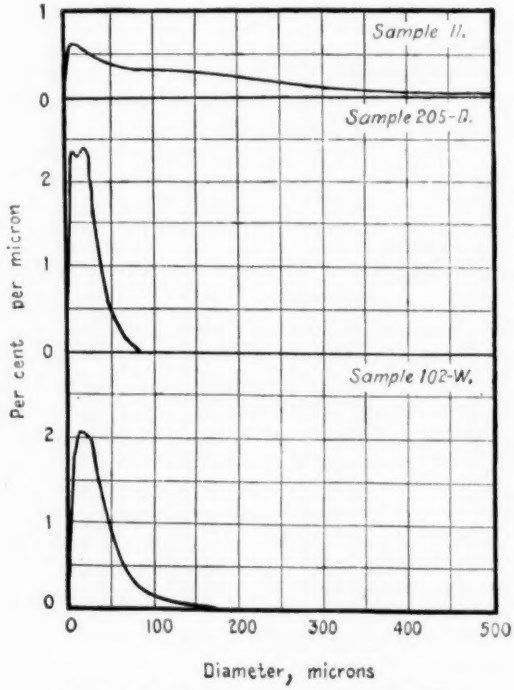


FIG. 2 CURVES FOR SAMPLES GROUND 2 HR

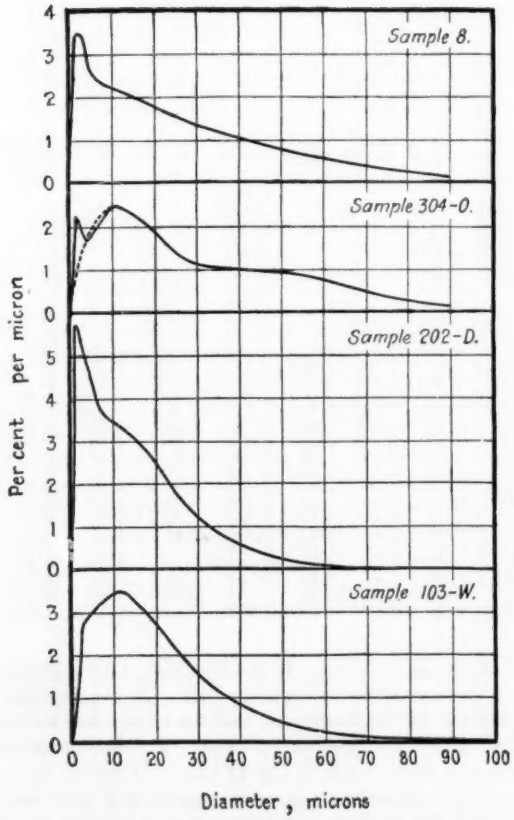


FIG. 3 CURVES FOR SAMPLES GROUND 5 HR

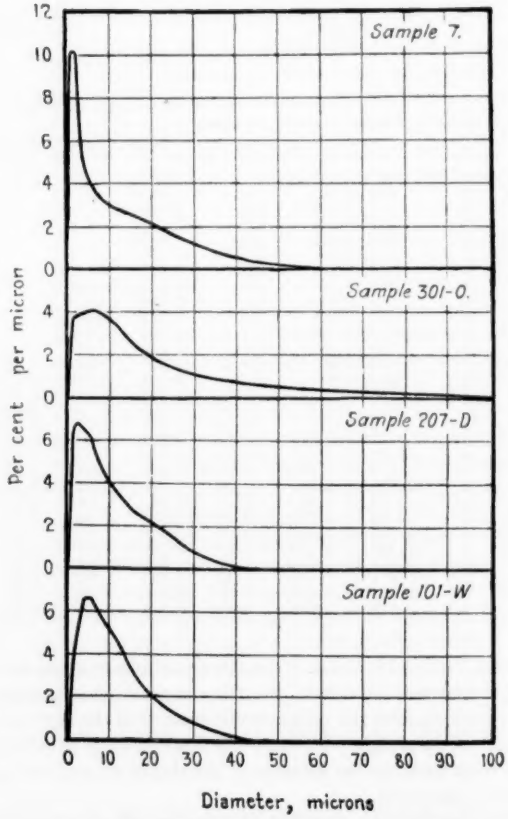


FIG. 4 CURVES FOR SAMPLES GROUND 14 HR

pare 205-D with sample 8 and 202-D with sample 7.) The grinding time to approximately equivalent fineness was on the order of the weight of the charge. This is not safe as a general conclusion.

Similarly, differences may be noted between the D, W, and O runs (400 g sand, dry, in 1200 cc of water and in 1200 cc of linseed oil). Increasing viscosity of fluid slows down elimination of coarse material. That the difference is not more marked between the D and W runs is probably due to the liquid dispersing the solid more widely. The action of elimination of particles seems to be a combination of the tendency to hold the particle in place for action and the slowing of the ball velocity. The O series was dissimilar to the first, but of the same order of time for equivalent grinding.

Grindings in the liquids gave a lesser content of extremely fine material. This is suggestive of a lubricating action which changes the predominance of wear to a larger size. In addition, the mill wear was greater with the dry grindings.

The data submitted in Tables 3 and 4 have been subjected to graphical and statistical analysis to determine the nature of the breaking of homogeneous solids. The same treatment has been used to fit values from other ball mills and also other types of mills.

Some of the samples tested by Weigel[16] and by Klein and Parrish[17] represent the products of direct grinding. Data on other mills submitted by Gaudin[18] and Haultain[19] have also been checked. It was observed from the large mass of evidence that the grinding of a homogeneous solid follows a relation involving at least two functions. The analysis of the relation requires consideration as to whether these functions are similar in all types of grinding operations and what is the basic nature of each of these functions. Beyond that, the problem is one of resolving the curve forms into these two functions, to study their variation with time of grinding and to show the application of the composite function for grinding operations.

A study of the relations as they occur in a large variety of fineness curves indicates the likelihood of purity of one function in the coarse range of sizes. In general it would seem that the other function which is predominantly for fine particles cannot be appreciable above one-third to one-half of the entire diametral range in which the particles occur and that it is actually significant only below one-quarter of the maximum diameter. It is thus possible to analyze the primary function over three-quarters of the diametral interval and to estimate its influence over the remaining interval.

This primary function is the "paraboloid loop" of Gaudin, when the data are plotted according to the sieve scale. In the per cent-per-micron plot it is the peak form in the coarse range. The frequency curve is much more illuminating. When the logarithm frequency is plotted against the corresponding particle diameter, the values fall on a straight line over the coarser three-quarters of the diametral range. Very small values in the extremely coarse sizes may appear to contradict this, but that is due to an overemphasis of insignificant values in the logarithmic plot or to unused feed. Ranging from the point of departure of the curve from the rectilinear down to zero diameter, there is a rapid rise which is considered indicative of the secondary function.

[16] Weigel, W. M., "Size and Character of Grains of Nonmetallic Mineral Fillers," U. S. Bur. of Mines Technical Paper No. 296 (1924).

[17] Klein, C. A., and Parrish, J., "Size of Particles With Special Reference to Pigments," Jour. Oil and Color Chemists' Assoc., vol. 7, pp. 54–80 and 82–106 (1924).

[18] Loc. cit.

[19] Haultain, H. E. T., "A Contribution to the Kick Versus Rittinger Dispute," University of Toronto, School of Engineering, Res. Bull. No. 4, p. 145 (1924).

It is evident that the general statistical relation which fits this primary function is the exponential proposed by Martin, Blythe, and Tongue[20] as the law of grinding. Their experimental work was done on silica sand in an 18-in. by 18-in. mill with steel balls. Sizing tests were made under the upper three-quarters of the diametral range. These data, added to the other, are contributory evidence as to the nature of this as one function.

The residual frequency or weight may be considered to represent the secondary function. Typical curve forms are given to show how the primary function fits the total figure. (Fig. 5.)

In view of the graphical representation of the primary relation, attention was directed toward graphical representation of the secondary action. An adequate basis was found when total increments of the logarithm frequency above the primary line were plotted as a straight line with the logarithm of the corresponding particle diameter as abscissa.

The data submitted earlier in the paper have been tested with these relations. (Fig. 6.) Lest there be confusion due to the orders of magnitude of the frequency logarithms, and lest slight

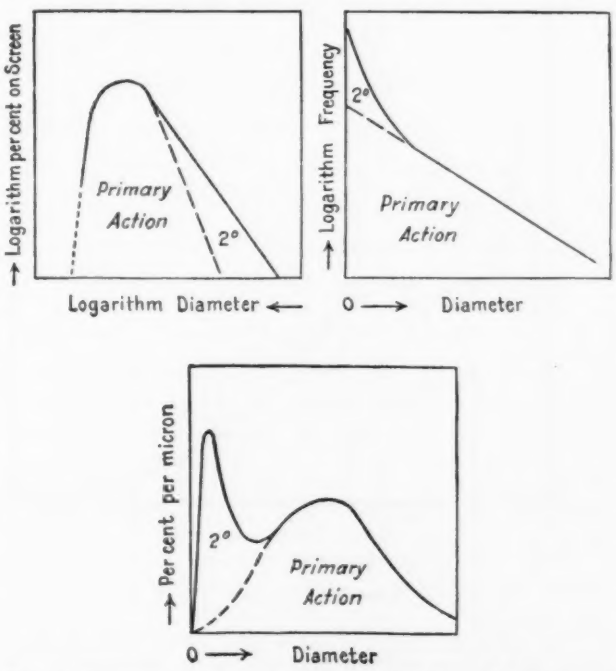


FIG. 5 DIVISION OF TYPICAL GRAPH FORMS INTO PRIMARY AND SECONDARY ACTION

variations of the points from the line be large on the weight basis, the frequency values taken from the straight-line relations have been calculated back to the per cent per micron, and the discrepancies have been noted there. This also has been done for a few curves taken at random from the experiments of Haultain[21] and of Gaudin[22] on the rolls and of Weigel[23] on the tube mill. (Fig. 7.)

Discrepancies occur at either end of the curve form, if at all. In the region of zero to two microns, they are usually small, being attributable to mill wear, dusting loss, and errors in the assumption of the mean diameter. The last factor is very important only when there is a great change in frequency across the range of the class interval. It was impossible to diminish the intervals sufficiently below 2 microns to eliminate this error.

[20] Loc. cit.

[21] Loc. cit.

[22] Loc. cit.

[23] Loc. cit.

The count in the lowest size range may be low, as the limits of visibility were close. These discrepancies have been so reduced in importance by the conditions of sizing that they need not be further considered.

Of greater significance are those few variations occurring at the coarse end of the curve. They have not been noted in the ball-mill tests, but are observed in some of the roll-mill data. They

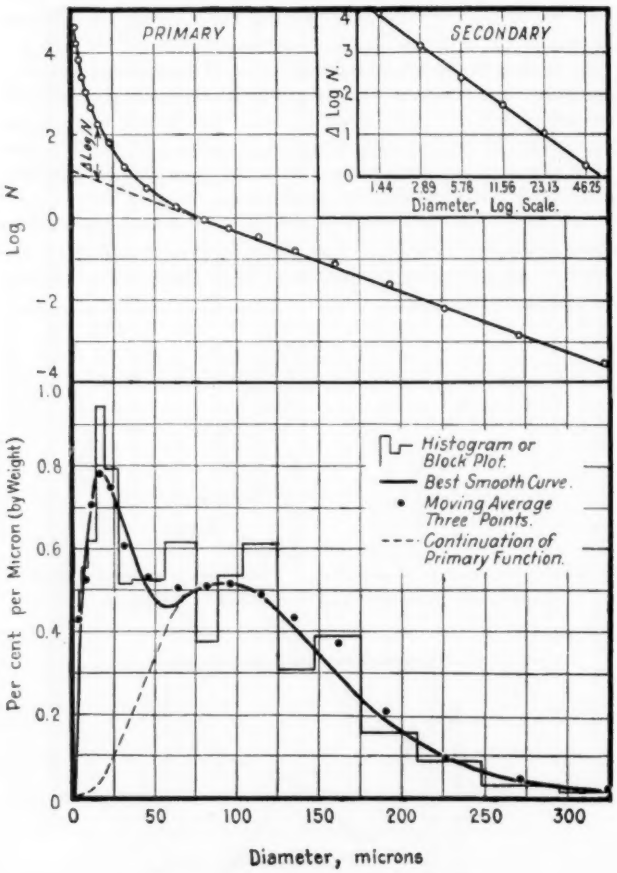


FIG. 6 RESOLUTION SAMPLE 105-W

show a greater elimination of coarse particles than the primary relation will permit. While some portion of this may be attributable to the effect of the roundness factor on the diametral measurement, the controlling factor is the positive sizing action of the rolls. Later analysis of the statistical formula will show how that can be.

There will be considered at this time a simple picture of the action which takes place in crushing to show how the general concept can fit the relations which have been proposed. The film of Haultain on the action of rolls in crushing shows that a particle may be disintegrated to produce a continuous function of size distribution from the maximum, near feed size, down to finer powder. It makes apparent the fact that the primary function may legitimately cover the full range from feed size to impalpable powder. The accumulation of such breaking action gives the general expression of the primary action in crushing. (Fig. 8.)

At the start of grinding there are N_d particles of feed size d. After a certain time, A_d particles of feed size will have been eliminated by breaking.

$$N_d - A_d = F_d$$

In breaking, one particle of size d will generate a statistical frequency in each of the finer ranges, as follows:

N'_{d-1} particles of size $(d-1)$
N'_{d-2} particles of size $(d-2)$
. . . . . .
N'_3 particles of size (3)
N'_2 particles of size (2)
N'_1 particles of size (1)

Each of these frequencies multiplied by A_d will give the total frequency in the respective size ranges, which are the result of the breakdown of feed size. (See Table 5.)

Similarly, further breaking will cause to be eliminated some of the particles of size $(d-1)$ to form finer particles, according to the law for the breakage of this size. The addition of the total of each of these breaking actions will give the resulting frequency for each size of particle in the total product of breaking.

From this relation, it is possible to observe the limitations of the exponential relation found to hold so generally and to analyze further the primary action in other types of grinding operation.

$$F_d : F_{d-1} : F_{d-2} : \ldots : F_3 : F_2 : F_1 = K$$

If the elimination factor A is a relatively small portion of the total, it may be neglected from the total of most of the frequency values. In addition, it is likely to be present in the same ex-

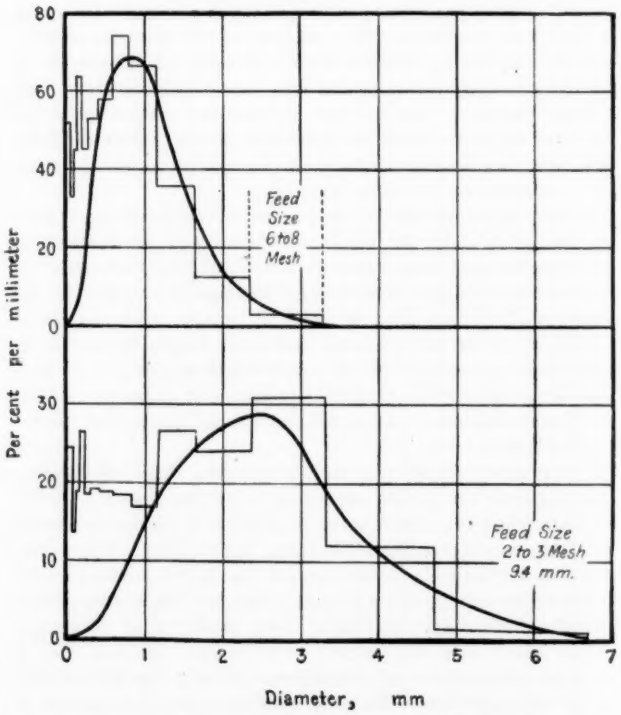


FIG. 7 HAULTAIN ROLL-MILL DATA; FITTED WITH CURVES OF PRIMARY ACTION

ponential relationship, as striking, and presumably breaking should be a function of the frequency of each particle size. It becomes suggestive that a unit-breaking action, such as the first, follows the exponential relation. A sized feed put through rolls, at not too great a reduction in size, would give that condition. There would be, however, a positive sizing action (the factor $-A_d$) of elimination.

The curves from Haultain's work fit well over a long range,

TABLE 5 SUMMATION OF PARTICLE BREAKINGS

Number of breakings	Size ranges: d	$d-1$	$d-2$	***	3	2	1
Feed	N_d						
First	$-A_d$	$+A_dN'_{d-1}$	$+A_dN'_{d-2}$	...	$+A_dN'_3$	$+A_dN'_2$	$+A_dN'_1$
Second	...	$-A_{d-1}$	$+A_{d-1}N''_{d-2}$	...	$+A_{d-1}N''_3$	$+A_{d-1}N''_2$	$+A_{d-1}N''_1$
Third	...	.....	$-A_{d-2}$	...	$+A_{d-2}N'''_3$	$+A_{d-2}N'''_2$	$+A_{d-2}N'''_1$
*****	...	.....	*****	...	*****	.....	.....
$d-2$	...	.....	.....	...	$-A_2$	$+A_2N_2{}^{d-2}$	$+A_2N_1{}^{d-2}$
$d-1$	...	.....	.....	.....	.....	$-A_2$	$+A_2N_1{}^{d-1}$
Total	F_d	F_{d-1}	F_{d-2}	***	F_3	F_2	F_1

while one of Gaudin's, carrying a high reduction ratio, shows the sizing action of the mill. When the rolls are set to less than minimum size of feed, each particle must be subjected to at least one breaking, and $N_d = A_d$. It may also chance that $A_dN'_{d-1} = A_{d-1}$, thus eliminating size $d-1$. The maximum particle size is controlled by this sizing action, and the elimination factor is necessary in such a formula as will express these cases. In ball milling, when the feed is eliminated and the accumulated fines are significant, the elimination factor and the feed factor are unimportant and the curve approaches more nearly its pure form. In that case, the elimination frequencies may well be proportional. If the breaking action is sufficiently positive so that each strike means the fracture of a particle, the frequency of breakage would be a function of the frequency of particle occurrence.

It has been observed that the secondary relation appears appreciable only below one-quarter of the maximum diameter. The lack of continuity with feed precludes its being directly the outcome of breaking. It seems likely that this is an action in which one particle loses a small corner without appreciable change in its own diameter and with the formation of a smaller particle. Until evidence is submitted to justify calling this action wear, this concept must be regarded as that of pulverization associated not alone with attrition, but possibly a secondary or even a different character of particle breaking.

For this action there has been found a straight-line relation between the logarithm of frequency above the primary value and the logarithm of diameter which is capable of analysis to a statistical formula. Log N_1 (representing primary relation) + $\Delta \log N$ (representing secondary relation) = log N (total).

$$\log N_1 + \Delta \log N = \log N$$

$$(N_1) \times \log^{-1} (\Delta \log N) = N$$

If $\Delta \log N$ versus $\log x$ is a straight-line relation,

$$\log x = q - p\,(\Delta \log N)$$

where

$$q = \text{value of } \log x \text{ when } \Delta \log N = 0$$

$$p = \text{slope } d(\log x) \div d(\Delta \log N)$$

$$\Delta \log N = -(1/p)\log x + (q/p)$$

$$\log^{-1}(\Delta \log N) = 10^{(q/p)}\, x^{-(1/p)}$$

Substituting

$$N = a(10)^{-bx}\,(10^{(q/p)}x^{-(1/p)})$$

$$N_1 = a(10)^{-bx}$$

Therefore

$$N_2 = a(10)^{-bx}\,(10^{(q/p)}\,x^{-(1/p)} - 1)$$

The statistical relation for secondary action may be taken as:

$$N_2 = (mx^{-n} - 1)\,a\,(10)^{-bx}$$

$$W_2 = (mx^{-n} - 1)x^3a_1(10)^{-bx}$$

where

$$m = 10^{(q/p)}$$

$$n = (1/p)$$

W_2 being the weight per unit diameter due to the secondary action.

The statistical relation which has just been expressed in mathematical terms has the significant feature that the secondary action is a function of the primary action and an exponential function of the particle diameter. This is to be expected from the concept which has been given. It may be assumed that, as the result of the secondary action, each particle gives the same number of finer particles of a specified diameter, which is to be a definite proportion of the original particle size. In such a case, the frequency curve for the secondary action would be an exponential, differing only in its constants from that of breaking. But though such a piece may statistically be the source of the same number of particles, they will actually be quite different in size. In addition, the frequencies observed at any one time are the accumulation of an earlier period, as well as those just produced. Breakage will gradually eliminate these, but account must be taken of the time effect. This leads to the conclusion that the expected relation is one dependent upon the primary function, with or without time lag, and upon the behavior of the individual particle in secondary action just pictured. The statistical relation is a reasonable expression of that concept.

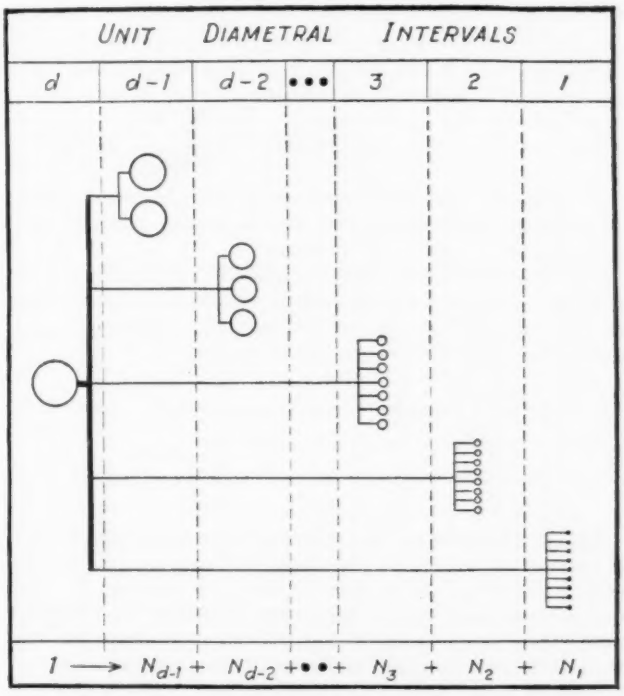


FIG. 8 GENERAL CASE OF BREAKING OF SINGLE PARTICLE

The formula may be limited where the exponential is not established for breaking, and possibly for excessively great wear. Variations will normally be introduced into these portions of the curve when non-homogeneous solid is ground. The graphical method is a simple means of determining the critical constants, and these in their turn are of use in defining the grinding action. The secondary function will be used between the limits of $x = 0$ and $x = \log^{-1}q$, to avoid negative values. The breaking function might better be used between feed size and zero to avoid an accumulation of small quantities between feed and infinity.

In themselves, the formulas derived from the graphical study are of use to secure the several values so frequently derived from fineness data. The materials which are both coarser and finer than 200 mesh are calculated by the weight basis for correlation, but ground products finer than sieve mesh or material sized microscopically over the range may be resolved directly. The constant corresponding to a will be erroneous, but that may be determined by resolution of the formula to the weight basis and making the total weight equal to 100 per cent:

$$100 \text{ per cent} = \sum_0^{\text{feed}} W = \sum_0^{q} 10^{(q/p)}\, x^{[3-(1/p)]}\, a(10)^{-bx} + \sum_q^{\text{feed}} a(10)^{-bx}\, x^3$$

Considering $W = dw/dx$, the form may be integrated, values of x substituted, as well as q, p, and b, and the constant a obtained for unit weight of the material.

Instead of performing a laborious calculation of surface at each diameter to secure the total surface, calculation may be made directly from this formula, by setting up the equation of unit surface and integrating to get the total surface.

$$\int_0^{\text{feed}} ds = \int_0^{q} (q/p)\, x^{[2-(1/p)]}\, a(10)^{-bx}\, dx + \int_q^{\text{feed}} ax^2(10)^{-bx} dx$$

The establishment of the constants also makes simple the calculation of any of the several diameters, based on weight, surface, or frequency, and what may prove to be more significant as well as more exact, the mean diameter of each of the functions independently. Take the weight basis as an example. If the mean diameter falls below $\log^{-1} q$ (or Q), the first portion of the summation will be divided, and that from zero to x equated to the second portion from x to Q, plus the breaking function from Q to feed. If it is greater than Q, the breaking function will be divided. Indications as to which one it is will occur in the signs of the summation. The same type of calculation may be used for calculations of mean diameter based on surface and frequency. (Table 6.)

Through the courtesy of Columbia University, the services of Dr. T. H. Gronwall were utilized in effecting the integration. The method involved expansion of the exponential, and an evaluation of the integral by Simpson's rule.

This analysis of the fineness characteristics of ground materials presents quantitative data on the trend of fineness in a small ball mill and a statistical treatment of the fineness distribution curves obtained in milling. There are shown to be two functions: the exponential of Martin and a secondary relation containing this exponential. The first of these relations represents the formation of relatively coarse pieces through direct breaking of the particle. The second represents the production of excess fines, either during or subsequent to breaking. Further analysis may show more exactly the origin of these laws and variations of the product with any mill type.

TABLE 6 SIGNIFICANT CONSTANTS

Sample	Time ground	Primary action: Slope LN/x	Primary action: Intercept LN_1, $x = 0$	Secondary action: Slope Lx/LN	Secondary action: Intercept Lx, $LN = 0$
First Series, Dry Grindings, 1100-G Charge					
13.....	30 min	0.00317	—1.60	0.360	+2.27
12.....	1 hr 4 min	0.00441	—0.93	0.362	+2.17
11.....	2 hr	0.00825	+0.11	0.368	+2.01
8.....	5 hr 9 min	0.0393	+2.81	0.370	+1.36
7.....	14 hr 7 min	0.0776	+3.97	0.375	+1.11
D Runs, Dry Grindings, 400-G Charge					
204-D.....	1 hr	0.0134	+0.93	0.381	+1.85
205-D.....	2 hr	0.0570	+3.51	0.418	+1.13
202-D.....	5 hr	0.0850	+4.25	0.435	+1.03
207-D.....	14 hr	0.1272	+4.87	0.482	+0.90
W Runs, Wet Grindings, 400-G Charge of Sand, 1200 Cc Water					
105-W.....	1 hr	0.0143	+1.13	0.415	+1.765
102-W.....	2 hr	0.0400	+2.85	0.461	+1.435
103-W.....	5 hr	0.0793	+4.15	0.548	+1.110
101-W.....	14 hr	0.1300	+4.97	0.628	+0.930
O Runs, Oil Grindings, 400-G Charge of Sand, 1200 Cc Linseed Oil					
302-O.....	1 hr	0.00497	—0.62	0.335	+1.98
304-O.....	5 hr	0.0390	+2.80	0.426	+1.45
301-O.....	14 hr	0.0583	+3.42	0.430	+1.27
305-O.....	50 hr	0.139	+4.97	0.449	+0.93

Physical Concept of Crushing and Pulverization

Innumerable terms have been used to define the actions of mills in the reduction of materials. These terms are in many cases the same as those used in a physical testing laboratory for the testing of specimens. It may be well, therefore, to consider grinding as a random physical test upon a peculiarly shaped specimen. Such factors as the nature of the substance, the shape of the piece, and the manner of loading will have an important bearing on the fineness of the resultant product.

The nature of the material under a testing machine will give some indication as to the manner of breaking in a mill. Take a metal like steel and subject it to compression. It will bulge slightly and finally break, probably into two pieces at an angle of approximately 45 deg with the direction of loading. Thus a ductile material is not easily crushed and does not easily produce dust. Take a more brittle substance like glass and subject it to compression after fracture. There will be the base cones or pyramids, some large pieces which will have broken from the sides, and a scattering range of sizes down to dust. The elastic deformation of material in the larger piece apparently produces several fractures, due either to multiple paths of breakage or to elastic recovery after breakage. A non-homogeneous material like concrete will furnish a similar fracture, but there may be less dust due to the fact that fracture tends to avoid the aggregate and passes around it. The analogy in grinding is shown in the work of Gaudin[24] where crushing disintegrates preferentially to the grain size of the material. There is another type of material which has either very easy cleavage or very weak bonding between grains. This finally disintegrates into an excessively large amount of uniformly fine material. Thus does the nature of the substance in the testing machine illustrate the wide differences in fineness distribution and in ease of breakage which may be obtained by compression loading in a mill—factors which have already been partially evaluated as grindability.

The illustration just presented covers only compression loading, and that under the conditions of flat, restricted ends. In grinding, the particles are usually round, and it is only by rare chance that a flat particle face meets a flat grinding face, but still a great amount of breaking is effected by direct pressure on these irregular specimens. The shape of the initial piece has a distinct bearing on the character of the fines which may be produced. A short investigation carried out by Norman Cudworth Cohn under the author's direction will show this difference. Silica gravel was sieved between $^1/_2$ in. and $^3/_4$ in., and the pieces falling within that interval were divided into three classes:

(1) Round or flat particles which in crushing between plane faces would have one point of contact on each face.

(2) Pieces which would have one point of contact on one face and two points of contact on the other face—bridge loading.

(3) Pieces which would have one point of contact with one face and three points of contact with the other.

Each individual stone was subjected to breakage in a small receptacle in a hydraulic press. The receptacle was stopped in its advance as soon as rupture had occurred. When the dust

[24] Loc. cit.

had settled, qualitative observation was made of the breakage, and the broken pieces were stored with the others of their kind. Table 7, of sieve percentages, will show a noticeable decrease in the finer sizes with a bridge support; in other words, the tension break produces less dust.

TABLE 7 SHAPE OF PARTICLE

Mesh No.	Aperture, mm	Per cent of weight 1-point round	2-point bridge	3-point
.. ..	13.33	9.11	20.0	8.87
.. ..	9.423	40.4	36.4	41.14
3.....	6.680	28.0	23.6	26.28
4.....	4.699	7.26	10.76	8.82
6.....	3.327	4.42	2.62	4.74
8.....	2.362	2.22	1.89	2.35
10.....	1.651	1.56	1.61	1.85
14.....	1.168	1.246	0.899	1.409
20.....	0.833	0.855	0.235	0.745
28.....	0.589	1.057	0.124	0.816
35.....	0.417	0.777	0.062	0.629
48.....	0.295	0.639	0.061	0.513
65.....	0.208	0.390	0.086	0.282
100.....	0.147	0.585	0.286	0.488
150.....	0.104	0.328	0.172	0.288
200.....	0.074	0.226	0.069	0.172
—200.....	— 0.074	0.856	1.208	0.607

It has just been shown that the manner of loading may produce tension, and that tension tends to produce less fines. Grinding mills introduce other factors in the manner of loading which are predominantly selective with respect to size. There is a type known as impact or beater mill in which a piece of material is thrown from a revolving paddle at a high velocity and forced to impinge on a fixed wall. In such a mill the finer particles are likely to follow the revolving air stream and not strike the wall directly. The harder coarse particles may strike the wall, but not at sufficient velocity to produce fracture. It is further conceivable that the type of fracture produced is that caused by rapid loading where the stress is highly localized and preceded the fracture with but little deformation of the entire specimen. In a physical testing laboratory the impact machine would bear the closest analogy to this action, but it must be remembered that most impact machines produce a wedging action with corresponding fracture in tension. Impact breakage produces a minimum of dust. Another selective action which is well known is that of the ball mill. A very coarse feed may not be subjected to sufficient loading by the tumbling of the balls or the compression within the mill to produce initial fracture. Smooth rounded pieces occasionally remain for long times with the grinding elements.

Thus it may be seen by analogy with the testing machines that there are wide differences in the manner of particle breakage which are inherent in the nature of the material and in the manner in which the load is applied. Shape of the particle may, within the same mill, produce differences in the manner of loading. The mill action may be selective, tending to load the specimen in such a way as to produce breakage to predominantly coarse sizes or by failure to act on specific sizes of particles.

Effect of Separating Devices

Separating devices are quite commonly used with mills in what is known as closed-circuit grinding. The charge is run rapidly through the mill, and then goes to a separator, which removes the oversize and returns it to the mill, allowing the undersize to pass on as finished product. This separation may be effected by the use of an air classifier or a liquid classifier, depending upon which is the medium for grinding. The purpose of closed-circuit grinding is largely one of economy. It also permits the production of fine material free of coarse grits. This may be important in the burning of portland cement, where large grits may introduce free lime or insoluble residue in the final product, and in the burning of powdered coal, where coarse particles pass rapidly through the flame without burning completely. Thus closed-circuit grinding is commonly used in the production of fillers for rubber or paint to eliminate particles coarser than 300 mesh which would be objectionable in the finished product.

These advantages are not to be gained without some compensating loss with respect to fineness characteristics of the finished products. It would be a mistake to assume that a product obtained by classification had the same surface as one obtained by direct grinding. This may be illustrated by a comparison of samples 11, 8, and 7. Sample 11 has been ground for two hours, and approximately one-quarter of the material is finer than 50 microns. This would be more uniform than that of either sample 8 or sample 7, and there would be much less exceedingly fine material. Sample 8 having been ground 5 hours is much finer, and approximately 15 per cent is over 50 microns. This could be eliminated by a classifier, with a resulting fineness distribution which would show an appreciable amount of fines, but an amount not equal to the fines of sample 7, where the

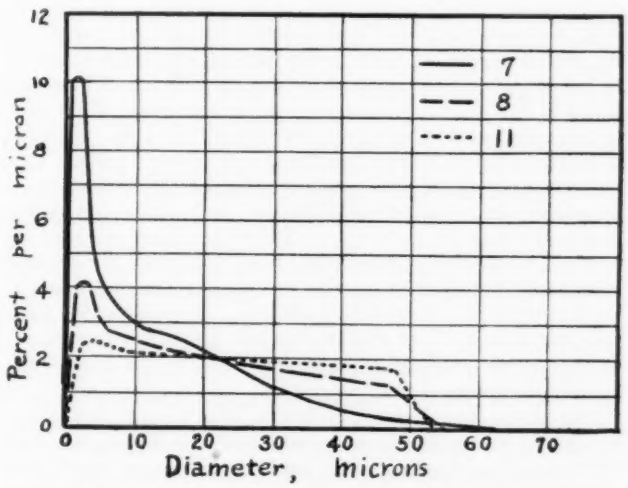


Fig. 9 Action of Classification on Ground Sand

product is ground to 50 microns. The distribution of the fineness of the final product is illustrated graphically in Fig. 9.

Separating devices permit of a marked economy of power and maintenance in the production of a material below a given upper limit of size, but the product is, at the same time, considerably coarser than one brought within the same size limits by direct grinding.

Practical Considerations

Sufficient mention has been made of the different types of crushing and pulverizing mills to show the diversity in operating characteristics. It is also clear that different types of mill have been commercially adapted to similar products, and only rarely is one type of mill used to the exclusion of all others. For example, in the pulverization of coal, hammer mills, roller mills, and ball mills all have their advocates, whereas in the final pulverizing of cement it is almost exclusively a tube-mill operation. At the present time there are not sufficient quantitative data to make easy the selection of mill types, or even of mills within a type, for a definite grinding problem.

The principles which have been laid down here illustrate the importance of grindability. A material which is difficult to grind should be reduced in a mill having extended wearing surface, parts which can be easily replaced, and a relatively slow and positive grinding action. The higher-speed mills with correspondingly less wearing surface find their application with materials offering less difficulty in grinding.

A consideration of energy requirements has shown that there is a large wastage of power over that needed to produce reduction in size. Therein lies an opportunity to improve the present milling procedures as well as an opportunity to compare the actual mill power with basic energy.

The statistical treatment of fineness curves and the analysis of the nature of breaking confirm the earlier work of the author on the difference in fineness-distribution curves produced under different grinding conditions and by different types of mills. Proper choice of the mill, proper loading, and the application of separators offer the opportunity to produce material of specific size characteristics or to produce material below a given size at a minimum of cost. The principles of closed-circuit grinding which have been used for years in mining industries are applicable in the finer range of sizes and offer a still greater economical advantage.

Discussion

R. M. Hardgrove.[25] Too frequently we take the product as it happens to come from the pulverizer, without much effort to fit the products to the needs of the process. For most purposes the ideal product would all lie in a very narrow range of sizes, as both oversize and undersize are usually objectionable. In some processes—as for example, pulverized coal for burning and pulverizing cement clinker—a large surface is desired; but even in cement, too large a percentage of the very small sizes is considered undesirable by some authorities. Efforts are being expended to produce a product approaching a uniform size, eliminating objectionable oversize and saving power and increasing capacity by minimizing the amount of undersize. A control of the fineness distribution in dry grinding can only be obtained by classification, either combined in the mill by air or screens, or external to the mill by screens or air separators.

Pulverizers are extremely inefficient, as is brought out in the paper, and this is largely so because not enough attention has been given to scavenging. If finished product is allowed to remain in the mill, it cushions the grinding action decidedly, and the more thoroughly we remove the fines as soon as they are produced, the higher the efficiency.

Batch grinding is extremely inefficient for that reason, and in open-circuit tube mills the bulk of the work is done in the inlet end where the charge is not cushioned by finished product.

The improvements made in the last few years in capacity, power consumption, and efficiency of pulverizers have all been produced by more recirculation and scavenging and lead us to believe that no pulverizer in use today is producing more than a small fraction of the work that it is really capable of doing, if the material were introduced to the grinding zone without having useless work done on it, and removed as soon as it was pulverized to the required size.

The paper quotes a pulverizer efficiency of 3.0 per cent based on the Gross and Zimmerley constant of 17.56 sq cm per gm of surface per kg cm of work when pulverizing quartz. While this method may not be entirely correct, we can use it in comparing the efficiency of more modern pulverizers.

Mills of the ball-bearing principle using air classification within the mill are daily indicating an efficiency of over 4 per cent, including motor and gearing losses, compared to the 3 per cent for open-circuit tube mills.

Another mill of this ball-bearing principle installed in a cement plant in closed circuit with an air separator is daily showing an efficiency of 5.7 per cent, including motor and machine losses, or nearly double that of the tube mills on open circuit commonly used for grinding cement rock. These results are possible by the use of a grinding principle which applies the force directly to the particles to be pulverized without lifting any unnecessary dead weight, by the low friction losses, due to the true rolling action of the balls, and by the high recirculating load and efficient scavenging. The laboratory method of determining grindability has proved to be extremely useful and reliable, and the more general use of this will enable us all to speak the same language, and eliminate one of the unknowns from the pulverizing problem.

The life of the pulverizer parts, however, seldom has any relation to grindability. Wear is a result of the combined abrasive and erosive action of the material being pulverized. In an impact mill the action is largely erosion, while in a tube mill it is largely abrasion. In mills having a true rolling action it is a combination of both, but is largely erosion. There is a vital need for a reliable index of the abrasive and erosive action of the various materials to be pulverized.

Metallurgical developments have contributed to the pulverizing art by supplying metals at a reasonable cost that have a relatively slow rate of wear, so that it is no longer necessary or good economy to choose a bulky and inefficient pulverizer even for cement rock or clinker.

Horace C. Porter.[26] As a member of Sub-Committee 7 on Pulverizing Characteristics, of Committee D-5 on Coal and Coke, American Society for Testing Materials, the writer would call attention to the work and objects of this sub-committee. As Dr. Work points out, the power requirements and the pulverizing results obtained are very different with different coals. Different types of mill are required for different coals. In burning coal, the surface exposed per unit weight (and also per unit volume, which has to do with density of the cloud) is of paramount importance. This surface factor is dependent upon fineness and shape of the particle.

Dr. Work brings out the effect of closed-circuit grinding on the size-distribution curves for the resulting material. This is very important with coal and is quite generally put into practice. The power requirements are affected thereby, and it is only necessary to produce such a degree of fineness as will permit of complete combustion under the conditions pertaining in the furnace used. There is always the question, however, whether finer pulverization, requiring more power, may not be justified by the better results to be obtained in burning rate and furnace capacity.

Coal is a non-homogeneous material. There are present not only mineral inclusions, most of them harder than the coal substance, but also physically different layers in the coal substance itself. These layers have been found to vary a great deal in hardness and grindability. Friability and grindability are not the same. The former, it has been proposed to determine by a drop or shatter test which by impact produces with different materials a different proportion of fines. Grindability, on the other hand, introduces the different function of abrasion or rubbing of surfaces in addition to the impact function.

It has been thought best in this A.S.T.M. sub-committee to concentrate on a testing device which will introduce both impact and abrasion—namely, the ball mill, with suitable proportion of balls to total space. Any test of this kind, however, is necessarily an arbitrary one and cannot duplicate exactly the conditions of practice.

The moisture in coal, as Dr. Work mentions, is of great importance in affecting its grindability. It may be added that the manner of moisture combination is also extremely important. Some of the moisture in coal is free, and other parts are in combination or physically retained in capillary pores. These

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[26] Consulting Chemical Engineer, Philadelphia, Pa.

forms of moisture have different effects on the brittleness and grindability of the material.

The coking power of a coal, sometimes called cokability, has an influence on its grinding characteristics. A well-known rough indication of cokability is the packing together of fine coal during the grinding process in a mortar or in some types of mill. Temperature, of course, has a large effect on such plastic materials as pitch and wax, possibly also on some kinds of coal.

It may be of interest, in connection with determination of total surface exposed by fine particles, to mention the work of Ralph A. Sherman and C. E. Irion at Battelle Memorial Institute, Columbus, Ohio, who are attempting various physical methods (other than microscopic counting and calculation therefrom) for the determination of surface. Among these are the adsorption of dyes, which depends essentially on the surface exposed, and the adsorption of radioactive material. It has been the writer's opinion that such methods would be complicated by the fact that total surface within the capillary pores might be indicated rather than the exposed surface of the particle as a whole.

The work of the sub-committee has not yet resulted in a standardized method of test, but it is believed that one will be developed as a result of the various studies, such as these of Dr. Work and others in this field.

Anthony Anable and B. J. Larpenteur.[27] Theoretical studies such as these are the greatest advantage, as in the final analysis all improvements in the technique of practical grinding are based upon the application of such theories as Dr. Work has deduced in this paper. In comparing various grinding devices and in comparing one method of grinding with another, Dr. Work has based his comparison on the amount of energy liberated, all of which he measures on the basis of Rittinger's units of total surface produced. While this is entirely satisfactory from the theoretical standpoint, we doubt whether this is of prime importance to the practical plant operator whose chief concern is the producing of the maximum amount of material just through a given mesh per unit of cost.

It is true, as Dr. Work has noted, that closed-circuit grinding produces a material which does not contain as large a percentage of fines as does an open-circuit grinding installation with both handling the same amount of material and both grinding it so that all of it passes a given mesh. This is due to the fact that closed-circuit grinding, through the rapid elimination of finished material from the system, does not tend to overgrind to the same extent.

It is the writers' contention that this very fact, the reduction of overgrinding, is a distinct benefit and one of the chief advantages of closed-circuit grinding. The practical operator in both the metallurgical and industrial fields has generally found, from tests or experience, the exact degree, measured by meshes or microns, to which his particular material must be ground in order that it may be treated efficiently in the next processing step. Any production of material finer than this critical mesh may be of interest to him from the theoretical standpoint, but nevertheless it has very little practical significance, since this finer product is not needed and has consumed power and grinding media which cannot be justified from an economic standpoint.

It is a widely known fact that closed-circuit methods of grinding have become practically universal practice in all metallurgical operations as well as in many operations in other fields where the tonnage to be handled is great and where one of the chief manufacturing costs is that of fine grinding.

Experience has shown that an open-circuit cylindrical mill will consistently produce more and more material of a certain fineness and will liberate more and more energy measured by Rittinger's surface units and Kick's energy units, as the rate of feed to the mill is progressively increased. This relationship was shown by E. W. Davis as a result of studies at the Minnesota School of Mines. In order, however, to take advantage of this progressive increase in output and liberation of energy, it has been found necessary to close the circuit by means of a separating device or classifier so that only material of the desired critical size may be released from the circuit for subsequent processing. The large amount of oversize which obviously is carried through the mill, due to the heavy load, may be returned for further grinding, and thus retained within the circuit until it has reached the point where it can be used economically in the next step. In addition, then, to the elimination of objectionable overgrinding, greater tonnage production results from the increase in the amount of material being passed through the mill at any one time.

When grinding in closed circuit, the mill is fed at several times its "open circuit" rate, the material races through the mill in a fraction of the usual time, and the product entering the classifier may contain as little as 30 to 40 per cent of finished material, with some material almost as coarse as the new feed. No attempt is made to finish the reduction in a single pass through the mill, but rather every effort is made to remove the material from the system just as soon as it reaches the required fineness, thus allowing the balls to work unhampered on the unfinished particles. The fact that capacity increases without corresponding increases in power is attributable to the more rapid elimination of fines, the reduction of uneconomical overgrinding, and the increased amount of coarse material which may be exposed to the cascading balls at one time. In practice the ratio of circulating load to new feed is generally in the neighborhood of 2 or 3 to 1, while in some instances the circulating ratio has been carried to 10 to 1 advantageously. Many engineers feel that the economic limits as regards circulating load ratio have not been reached with present practise. A new Canadian installation will employ an 18-to-1 circulating ratio for grinding a hard gold ore.

The practical and commercial aspects of fine grinding must not be lost sight of, even in the case of such an excellent theoretical paper as this of Dr. Work. The production of a very fine product, the liberation of energy, and the producing of increased surface are of definite interest to those who are working out the theoretical concepts of fine grinding. However, these are generally of only secondary importance to the practical man who is not especially concerned with any material ground finer than his own particular critical mesh, and is inclined to regard the presence of "superfines" as definite evidence that he is wasting his power unnecessarily and that his grinding system is not operating at the maximum economic efficiency, although it may be operating at the maximum theoretical efficiency.

G. F. Metz.[28] While the author has set forth much of theoretical interest to engineers interested in grinding, it is believed that a few comments, as the result of our own laboratory and field experience, would be of interest. We have found that the use of a jar or batch mill to determine the grindability of various materials is a step in the right direction, but that it is wrong to use the capacity information from this size mill in figuring capacities of larger diameter mills, or as Martin has worked out in his paper entitled, "The Science of Grinding." In this paper he takes an 18-in. by 18-in. ball mill using 1-in. steel balls and from the work of this small machine calculates the amount of power required to grind 3 tons per hour of silica sand from approximately minus 20 plus 30 mesh to 99 per cent through 200 mesh. We have found that the larger diameter mills are

[27] Engineers, The Dorr Company, Inc., New York, N. Y.

[28] G. F. Metz, Division Engineer, Hardinge Company, Inc., York, Pa.

much more efficient than the smaller sizes. For instance, a 10-ft conical ball mill will grind 15 to 20 per cent more tons per horsepower through a certain sieve mesh than the 8-ft mill. This we attribute to the tremendous mass action in the larger diameter mill. It is this same mass action which allows us to adjust our ball charge to the particular reduction required rather than to the coarsest particle fed to the mill. Due to the tremendous drop of the charge in a 10-ft mill we can use smaller balls than, for instance, on the same material in an 8-ft mill when attempting to grind to the same fineness.

The batch mill is no indication of actual power required in modern grinding equipment, as no provision is made for the removal of finished material, which in itself cushions the grinding charge and prevents efficient reduction. In continuous mill operation, where the mill is operated in conjunction with a classifier, regardless of whether it is wet or dry grinding, there is a continuous removal of fines from the grinding circuit, leaving the mill charge free to work on coarse and granular material.

Referring to the author's quoting of Gross and Zimmerley, it is the writer's opinion that all energy input in a ball mill is dissipated into heat, except that portion which is stored in a deformed particle as potential energy, which in itself would be released by a slight additional blow in the form of heat. The author states that grinding in liquids gave a lower content of extremely fine material. We have found that it is not true generally in practice, and that it is just as easy to obtain superfine material in wet grinding as in dry. All our experience has been that the wear on the parts is greater in wet grinding mills than in dry grinding. For instance, on limestone, the total steel consumption is not over $^1/_4$ lb per ton in a conical ball mill equipped with air separation, when grinding to around 90 per cent through 200 mesh. On the other hand, wet grinding on this same material in a conical ball mill shows a ball wear of $^1/_4$ lb per ton and a lining wear of $^1/_4$ lb per ton, making a total steel consumption of $^1/_2$ lb per ton. The same is true in ore grinding. The steel consumption in wet work is at least two to four times as great as for the same reduction in dry grinding.

In a paper entitled "Modern Grinding" by J. C. Farrant, before the Chemical Engineering Group at Derby, England, on January 23, 1931, the author classifies mills as follows:

1 Low speed (attrition and impact): Pan mills, buhrstones, Hardinge conical ball and pebble mills, cylinder ball and pebble mills, tube mills, rod mills.

2 Medium speed (compression): Ball ring mills, multiple-ring roll (horizontal ring), multiple-ring roll (vertical ring), single-ring roll, vertical emery mill.

3 High speed (impact and shearing): Hammer mill, cage disintegrator, impact mill, shredders.

Based on actual operating data he has listed various materials as being applicable to this list of grinding machines, and in a great many cases some machines which will pulverize the harder materials will do equally as well on the softer ones as even those machines which are designed primarily for soft materials. In Mr. Farrant's paper he has listed considerable actual operating data on modern installations, and it is believed his paper would be of interest to any one having grinding problems.

Laboratory results in grinding experiments, as far as ball and pebble mill tests are concerned, show lower capacities per horsepower input than commercial operations. This is due to the absence of mass action. The height of ball drop on the same size particle in a commercial grinding mill causes much greater impact and considerably more fines on the initial break. Also for a fineness to a given mesh, a commercial mill will show considerably more superfines than the laboratory mill, which we attribute to the foregoing mass action.

MARTIN FRISCH.[29] The statistical method devised for analyzing the performance of crushers and pulverizers, especially for predicting size distribution and for searching out the meaning of the results of sieve analyses, will prove useful to many interested in pulverizers and pulverizing processes.

Unfortunately, so complex are the processes with which this paper concerns itself that even a method of such as introduced by this paper cannot be applied at once to meet the requirements of every one. The requirements of each industry employing pulverizers differ. Size distribution may be all important in one industry, whereas in another, color or specific weight of the product may be the criteria which determine whether or not the performance of a pulverizer is satisfactory. The problems of those who pulverize coal for combustion are probably simpler than those of the chemical engineer. Size distribution as such is of relatively small importance provided the product has a minimum fineness as measured by some one sieve. This minimum fineness is not the same for all coals. The design of the furnace in which the coal is burned is also an influence. Since the performance of most pulverizers sold for the purpose of preparing coal for the firing of furnaces is defined as so many tons per hour at a particular fineness, through say a 200-mesh screen when the size of feed is such that all will pass a screen of a certain size, and when the feed of a given grindability contains not more than a given maximum per cent of moisture, it becomes obvious that particle-size distribution is not as important to either the seller or the user as are such things as the effect of size of feed, initial moisture content, grindability, and ultimate fineness of product on the capacity.

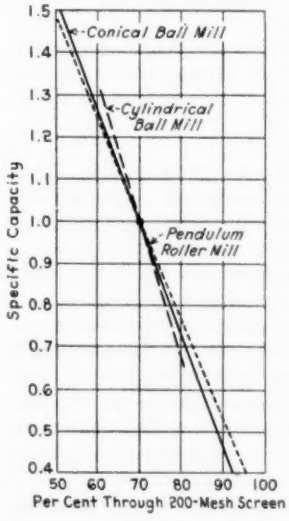


FIG. 10 EFFECT OF FINENESS ON PULVERIZER CAPACITY
(Capacity at 70 per cent through 200 mesh = 1.0.)

Much work has been done to determine the relation between combustion efficiency and fineness. It has been found that greater fineness means more complete combustion and hence higher efficiency. But it has also been found that even with relatively coarse grinding the combustion efficiency is so high that little is to be gained from finer pulverization that could be justified by the increased expense for obtaining the increased fineness. The capacity of each pulverizer drops off very rapidly with each per cent increase in fineness as measured by the standard screen, which happens to be the 200 mesh. Hence initial investment in pulverizing equipment increases quite rapidly. For example, a study has been made of the effect of fineness on the capacity of pendulum roller, cylindrical, and conical ball mills. As shown by Fig. 10, the reduction in capacity with increase in fineness was very marked for each, and of about the same magnitude. Tenney, in his paper, "Pulverization and Boiler Performance," dated February 11, 1931, has shown that the most economical fineness for Illinois coal is approximately 57 per cent. For other coals the most economical fineness may be greater or less, depending on whether the coal is of a coking or free-burning variety.

A pulverizer may be expected to yield from 25 to 50 per cent greater capacity, all other conditions being the same if the size of feed is reduced from all through 1 in. to, say, all through $^1/_4$ in.

[29] Foster Wheeler Corporation, New York, N. Y. Mem. A.S.M.E.

This principle has been tested in the cement-clinker-grinding field as well as in plants pulverizing coal for firing boilers. Whether or not such precrushing is justifiable depends on economic factors generally peculiar to each particular installation.

The effect of moisture on capacity is also very marked, as the author has suggested. Moisture is one of the most serious factors to be contended with in the pulverization of coal. In the early days the use of coal driers was deemed a necessity, because it was found that if the moisture exceeds a certain critical value, depending on the kind of coal and on the type of pulverizer used, the capacity of the pulverizer may be so reduced as to make the pulverizing equipment inadequate to meet the load requirements. In some cases the capacity might disappear entirely. In general, it may be said that for most pulverizers each per cent increase in surface moisture reduces the capacity of the pulverizer from 3 to 7 per cent. The detrimental effects of moisture in the feed have been partially offset by the use of preheated air for carrying the coal out of the mill. The preheated air partially or entirely dries the coal during the pulverizing process and materially increases the capacity with wet coal.

Of all the factors influencing the performance of a pulverizer, grindability is one of the most important. Unfortunately, no standardized method has yet been developed for determining the grindability of coal which all may use as a basis for reporting their findings to the same scale. Due credit must be given to Mr. B. J. Cross, who first developed the laboratory method now most generally used, and to Mr. R. M. Hardgrove, who later developed an improvement on this method. Grindability as defined by Cross and Hardgrove unfortunately does not measure the capacity of a pulverizer directly, because the output of a pulverizer is not directly proportional to the grindability. However, a relation can be established by experience between the output of a pulverizer and any grindability scale. Some doubts have been cast by many on the value of grindability tests because the performances of pulverizers sometimes do not seem to be consistent with the laboratory grindability determinations. These doubts should disappear when the results obtained with commercial pulverizers can be properly interpreted and reduced to a common basis. It is obvious in view of what has already been said as to the effect of initial feed size, moisture, and fineness of the ultimate product that in order to be comparable the results of tests with commercial pulverizers must be corrected to a common basis as to feed size, initial moisture content, and fineness. The company with which the writer is connected has determined the grindability of a large number of coals in its laboratory, and after comparing the results of these tests with actual performance of its pulverizers when grinding those coals we have found that if the results are corrected properly the pulverizer performance is accurately predictable on the basis of the laboratory grindability.

These remarks are offered because, in the writer's opinion, the field of those who would study pulverizing processes mathematically is still unexplored and full of possibilities. Obviously, in the present state of our knowledge a mathematical analysis is of very little value, except to a restricted few. Materials to be pulverized so lack homogeneity and the yardsticks by which we measure whether a given treatment for these non-homogeneous materials yields satisfactory results vary so widely that for a long time to come little may be expected of a purely mathematical study. Statistical analysis of the final product described by the author may help us to understand better what happens in a pulverizer if used in conjunction with experience. But experience must be relied on more for the present.

H. P. Reid.[30] The writer will limit his comments to four general subjects, as follows:

[30] Special Engineer, Universal Atlas Cement Co., Chicago, Ill.

1 In the classification of the general type of machines for reduction of the size of materials, the author has failed to mention the jet impact type of pulverizer. This machine has been used relatively little in this country but it has possibilities and characteristics which may bring it into service for certain desired material reduction. In general it may be said that the jet impact mill may be subdivided into two types—first, that with the use of a target or wear plate or wear body against which the material to be reduced is jetted or blasted, and, second, that with several jets of the material and gaseous or fluid vehicle impinging or impacting each other.

2 In the production of surface exposure, it may be possible with some materials to obtain, by selected milling processes, particles which vary decidedly from spheres or cubes. Where such is possible, the desired chemical reactions may be obtained with apparently coarser particle sizes or more complete chemical reactions may result with a given particle size, as measured by sieves, than would result were the material broken to near-spheres or near-cubes. This may decidedly change the type of mill selected.

3 Fahrenwald, Hammar, Lee, and Staley have, according to the author, carried out experiments which indicate grinding-mill efficiencies from 6 to 25 per cent. With 75 to 94 per cent of the energy input to be dissipated in some useless way, one wonders just how this dissipation takes place. It must be mostly to heat and sound energy. That part which is dissipated as heat must tend to raise the temperature of the material load of the mill. Apparently very little study has been made of the effect of this temperature rise on the grindability of the material. The author enumerates several materials whose properties would be injured by temperature rise, but has not pointed out the possible loss, from even the 6 to 25 per cent grinding efficiencies, which might and in fact evidently do result with many ores and stones. Particularly does this temperature effect appear to be detrimental with extremely fine grinding. There seems to be a greater tendency for the extremely fine particles to cohere, to coat the grinding media, and to build up cakes which cushion the grinding action as the temperatures rise. The exact nature of the heat action and the temperature limits for optimum grinding conditions apparently have not been fully investigated. It may be possible to reduce the amount of the energy dissipated as sound by proper design of mill and of its foundation, etc. But were this accomplished, what would be the net result? Would any more energy be made available for particle reduction with the random action of the grinding media, especially with tube and ball mills? If the energy now dissipated as sound were transferred to heat, the net results would very likely be higher temperatures and increased coating and cushioning actions, which in turn would probably mean still lower mill efficiencies.

4 The author refers to various methods which have been suggested, some of which are in use, for determining the relative grindabilities of materials. He has also pointed out the serious effect of moisture on grindability of coals. Apparently, very little is known of those properties which affect grindability. May it also be that we know relatively little about measuring grindability? Hardgrove has reported some very interesting studies on the grindability of coal. His work has placed a large question mark on the entire subject as he finds coals which have nearly the same general chemical and physical characteristics that differ widely in grindability. What are the factors which govern grindability? Can these factors be economically altered by methods used in drying, treating, storing, etc., of materials which are to be pulverized? Portland cement clinker is produced by heating finely ground raw materials to approximately 2700 F. The methods used in cooling of this clinker to grindable temperatures apparently greatly affect its grindability. Quick quench-

ing of the clinker fractures the lumps by innumerable hair cracks, thus simplifying the crushing. Again the age of the clinker when ground apparently affects its grindability. If this is true with portland cement clinker, may there not also be controllable factors or pretreatment of other materials which could be economically used to reduce the costs of pulverizing?

It thus appears that there is a need for much more research on particle-size reduction of materials in general.

G. H. KAEMMERLING.[21] Probably the most striking fact brought out in the paper is the great discrepancy between the maximum and minimum figures on crushing efficiency. It has been recognized for some time that available pulverizing processes are very inefficient with respect to the utilization of the energy supplied, and even the maximum figure proposed for crushing efficiency leaves a wide margin for improvement. It does not seem likely that ordinary refinements in current pulverizing methods will go very far toward closing the gap between actual and theoretical efficiency. Once given a mark to shoot at, in the form of a generally accepted basis for estimating the theoretical energy requirement for any degree of reduction of a given material, engineers may be expected to investigate more closely the mechanism of the energy losses in present pulverizing methods, with the purpose of developing new and more efficient methods. The determination and satisfactory establishment of such a basis will constitute a very valuable research work.

In the case of a non-homogeneous material, such as coal, there are differences in resistance to crushing among the particles of a given size produced during the pulverizing process, due not only to cleavage planes containing ash, but also to the variation in crushing resistance of the constituents of the coal compared with the constituents of the ash. In the commercial process of pulverization the coarser particles are retained in the mill, or removed and returned to the mill, until sufficiently reduced. This results in building up a charge in the mill of greater average hardness than the feed. It is possible that this effect is not important with ordinary Eastern coals. In the case of coals containing a high percentage of abrasive ash, however, a laboratory investigation of the rate of reduction of the different constituents would produce interesting information regarding the effect of ash content and character on power and maintenance requirements.

[21] Research Director, Erie City Iron Works, Erie, Pa. Assoc. Mem. A.S.M.E.

AUTHOR'S CLOSURE

The discussion represents an effective supplement to the aspects of the problem developed in the paper. One point needs further clarification to avoid apparent contradiction, and that is the comment by Mr. Metz on the relative fineness obtainable when grinding dry or wet. The illustration given in the paper relates to the extremely fine particles up to 3 or 4 microns in size, while the comments offered by Mr. Metz relate to a larger portion of the sub-sieve range. The action described in the paper illustrates the fact that grinding forces cannot be applied to extremely fine particles when the fluid medium is sufficiently viscous to carry the particles with it as the fluid is expelled from beneath the grinding faces. The discussion as a whole has brought out fully the importance of accurate sub-sieve measurement and the very limited amount of data now available.

RP-55-7

High-Pressure Steam-Generator Research

Progress Report of Investigations at Purdue University by the A.S.M.E. Special Research Committee on Critical-Pressure Steam Boilers

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This is a progress report of the investigations with the high-pressure steam generator at Purdue University which have been carried on since January 1, 1932, and are mainly concerned with the measurement of the enthalpy of steam by the use of a condensing calorimeter designed for pressures up to 3500 lb per sq in. The heat leak was determined by assuming that the Keenan Tables were correct within the range at which the calorimeter was calibrated. Thirty-nine tests were carried on at pressures varying from 1900 to 3500 lb per sq in. and temperatures of 709 to 780 F. The results were compared with those of Keenan, Knoblauch, Callendar, and the International Tables. The Purdue University figures are in close agreement with those of Keenan, Knoblauch, and the International Tables, but show considerable variance from Callendar's results.



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IN A PAPER that was presented at the Annual Meeting of The American Society of Mechanical Engineers on December 1, 1931, entitled "Characteristics of a High-Pressure Series Steam Generator,"[4] results were given of the operation of a high-pressure steam generator at a variety of capacities, pressures, and final steam temperatures. The behavior of the unit on a swinging load under automatic control was also demonstrated. Since these preliminary tests were reported, the following studies have been undertaken:

1 The laws governing fluid friction of steam at pressures up to 3500 lb per sq in. and temperatures up to 830 F have been studied. The results to date are insufficient to allow any definite statement; however, it is expected that these studies of fluid friction will be completed within the next year.

2 Major attention since January, 1932, has been devoted to the design, construction, calibration, and use of a condensing calorimeter in order to determine the enthalpy of steam at the conditions under which the steam generator is being operated.

The High-Pressure Condensing Calorimeter

A condensing calorimeter of the steady-flow type was designed to measure enthalpy at pressures up to 3500 lb per sq in. and temperatures up to 800 F. A simplified diagram of this type of calorimeter is shown in Fig. 1. Steam is condensed in a coil of tubing sufficiently heavy to withstand full steam pressure. The rate of condensation is controlled by a regulating valve on the condensate discharge. The cooling water on entering the calorimeter flows through a passageway entirely surrounding the condensing chamber. It then enters the section which contains the coil, receives heat from the steam, and is discharged.

The details of this calorimeter are shown in Fig. 2. Steam is condensed in 60 ft of extra-heavy $^1/_4$-in. iron-pipe-size seamless-steel tubing, coiled on an average center-to-center diameter of $12^3/_8$ in., the total external surface being about $8^1/_2$ sq ft. The coil chamber and water jacket were formed by four coaxial cylinders constructed of 16-gage sheet steel. All joints were made by oxyacetylene welding. For insulation between the condensing chamber and the water jacket, 85 per cent magnesia was used; also for covering the outside of the calorimeter. The inlet and outlet ends of the steam coil, as well as the water-discharge tube, were all brought out through the water jacket in circular openings of sufficient diameter to provide an inch or more of insulation between the jacket and each of the tubes.

Inlet and outlet cooling-water temperatures were measured with mercury-in-glass thermometers. Tycos thermometers graduated in 0.1-deg divisions were used. The two inlet thermometers had a range of 32 F to 120 F, while the outlet thermometer range was 0 C to 100 C. A centigrade thermometer was used

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TABLE 1 RESULTS OF CALIBRATION TESTS

1	Test number	1	2	3	4	5	6	7	8	9	10
2	Date	3-12	3-12	3-12	3-12	3-14	3-14	3-17	3-17	3-17	3-19
3	Duration, min	22.65	22.32	23.67	24.22	24.02	24.15	24.05	24.66	24.90	26.17
4	Total steam condensed, lb	50	50	50	50	50	50	50	50	50	50
5	Total cooling water, lb	1151.5	1177.0	1158.5	1106.5	1132.0	1148.0	1218.5	1105.0	1051.5	1219.0
6	Pounds cooling water per lb steam	23.03	23.54	23.37	22.13	22.64	22.96	24.37	22.10	21.03	24.38
7	Steam pressure, lb per sq in abs	200.7	200.1	201.6	202.3	201.0	202.3	200.9	202.4	203.6	201.7
8	Steam temperature, F	849	851	735	625	726	721	835	683	675	872
9	Condensate temperature, F	74.5	73.0	74.0	73.1	73.3	73.1	75.8	73.8	74.0	75.4
10	Air temperature, F	72.0	72.0	72.0	73.0	71.5	71.3	73.8	72.5	73.0	74.3
11	Inlet-water temperature observed, F	72.56	72.12	73.51	73.13	72.01	71.80	73.91	71.85	72.71	74.47
12	Outlet-water temperature observed, C	56.18	55.28	54.86	55.06	54.98	54.31	54.96	55.10	57.08	55.71
13	Stem temperature, inlet thermometer, F	77.0	78.3	78.0	78.0	77.5	76.0	77.3	76.5	77.0	80.0
14	Stem temperature, outlet thermometer, C	24.6	25.0	24.8	24.4	24.3	24.3	24.6	24.4	24.4	26.1
15	Stem correction, inlet-water thermometer, F	−0.016	−0.023	−0.017	−0.018	−0.020	−0.015	−0.013	−0.017	−0.016	−0.021
16	Stem correction, outlet-water thermometer, C	0.319	0.302	0.294	0.304	0.305	0.294	0.297	0.300	0.330	0.292
17	Inlet-water temperature, corrected, F	72.54	72.10	73.50	73.11	71.99	71.87	73.90	71.83	72.69	74.45
18	Outlet-water temperature, corrected, F	133.70	132.05	131.29	131.66	131.52	130.29	131.46	131.72	135.33	132.78
19	Enthalpy of inlet water (h_1) Btu per lb	40.6	40.1	41.5	41.1	40.0	39.8	41.9	39.9	40.7	42.5
20	Enthalpy of outlet water (h_2) Btu per lb	101.6	99.9	99.1	99.5	99.4	98.1	99.3	99.6	103.2	100.6
21	Enthalpy of condensate (h_c) Btu per lb	42.9	41.4	42.4	41.5	41.7	41.5	44.2	42.2	42.4	43.8
22	Apparent total enthalpy of steam (h_g) Btu per lb	1447.2	1448.8	1389.3	1333.6	1385.9	1381.1	1443.1	1362.1	1356.1	1462.2
23	Corresponding h_g from Keenan Tables, Btu per lb	1451.3	1452.3	1391.4	1333.7	1386.8	1384.1	1443.9	1364.1	1359.8	1463.2
24	Heat leak, Btu per min	9.05	7.84	4.43	0.21	1.87	6.21	1.66	4.05	7.43	1.91

for the outlet-water temperature measurement as no fahrenheit thermometer with a suitable range and with proper graduations was available. The inlet-water thermometer bulbs were in direct contact with the water, the thermometers being held in position by corks in the pipe tees provided for that purpose. The same arrangement was tried at first with the outlet thermometer, but due to rapid temperature fluctuations it was later abandoned and a thermometer well was used. The thermometer well was screwed into a counterbored cork and was thus insulated from the pipe and tee. Details of the thermometer well and also of the baffles for mixing the outlet water are shown in Fig. 2. By using magnifying glasses in reading the thermometers, it was possible to estimate the readings to the nearest 0.05 deg. Stem corrections were made for all tests. The temperature of the surrounding air was measured by thermometers on either side of the calorimeter.

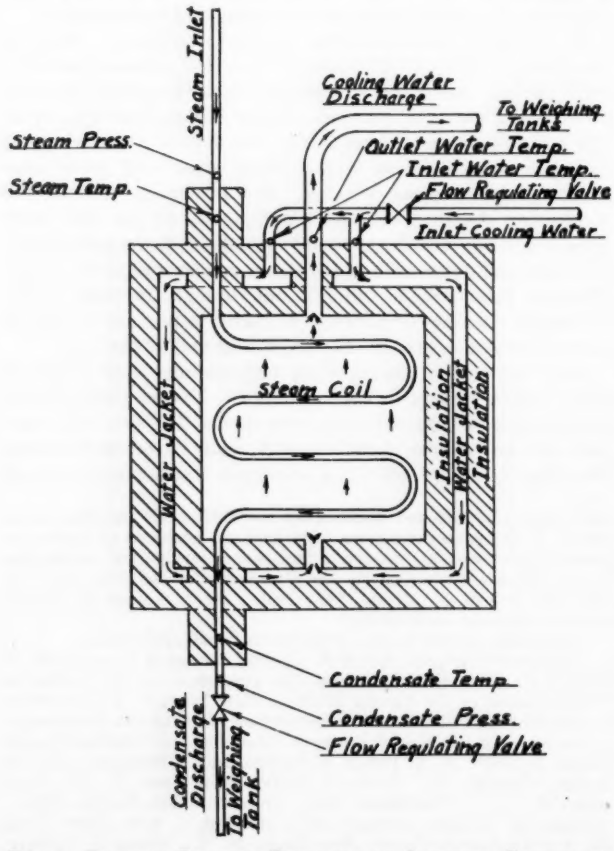


FIG. 1 DIAGRAM SHOWING PRINCIPLE OF JACKETED CONDENSING CALORIMETER

During the calibration tests Crosby bourdon-type test gages were used for steam and condensate pressure measurements. For the high-pressure tests the entering steam pressure was measured by the Bailey dead-weight pressure-measuring device installed as part of the high-pressure steam-generator equipment. Thermocouples, constructed as shown in Fig. 3, were used to obtain the steam temperature at entrance to the calorimeter and also the condensate temperature at discharge. Flaked mica was used for packing, and fused-quartz tubes were employed to insulate the leads from the thermocouple body. The distance from the hot junction to the end of the thermocouple body was approximately 1 in. The steam and condensate thermocouples were constructed of standard iron-and-constantan thermocouple wire, and the millivolt-temperature tables used were those provided by the manufacturers. Both thermocouples were connected into the entering steam line and tested with saturated steam at different pressures. Their readings were found to agree and to check the corresponding saturation temperatures within the limits of accuracy possible with the potentiometer.

Millivolt readings were obtained by Leeds & Northrup potentiometers. For the calibration tests an instrument was used that was capable of being read to the nearest 0.1 mv in the steam-temperature range and to 0.02 mv in the lower or condensate-temperature range. For the experiments with high-pressure steam greater accuracy in steam-temperature measurement was required due to the rapidity with which enthalpy changes with temperature at high pressures. Accordingly, a potentiometer having a sensitivity of 0.02 mv in the upper range was used in most of these tests. Cold junctions were maintained at 32 F by means of crushed ice.

The weight of the cooling water was obtained by use of two 800-lb-capacity tanks mounted on standard Fairbanks platform scales. The scale beams were graduated in $^1/_2$-lb increments, and the scales, after careful calibration and adjustment, were found to be sensitive to variations of about $^1/_4$ lb. The condensate was weighed in a tank of about 200 lb capacity set on a calibrated Fairbanks platform scales of 250 lb capacity. The scales had beam graduations of $^1/_4$ oz and was sensitive to variations of about 0.2 oz over its entire capacity range. A double-pipe heat exchanger, supplied with steam from the laboratory header, was inserted in the cooling-water line to provide for control of the temperature of the cooling water entering the calorimeter.

CALIBRATION OF THE HIGH-PRESSURE CALORIMETER

With proper adjustment of the inlet-water temperature, the heat leak from a jacketed calorimeter can be made negligible

TABLE 1 RESULTS OF CALIBRATION TESTS—(*Continued*)

1	Test number	11	12	13	14	15	16	17	18	19	20
2	Date	3-19	3-21	3-22	3-22	3-23	3-23	3-23	3-23	3-24	3-24
3	Duration, min	22.46	26.75	25.40	20.54	29.60	24.96	22.17	23.15	19.85	19.21
4	Total steam condensed, lb	50	20	30	26	25	25	25	25	25	25
5	Total cooling water, lb	1089.0	495.0	661.5	576.0	585.5	570.5	534.0	540.0	593.0	395.0
6	Pounds cooling water per lb steam	21.78	24.75	22.05	22.15	23.42	22.82	21.36	21.60	23.72	23.80
7	Steam pressure, lb per sq in abs	200.9	205.0	206.8	205.6	206.9	206.2	206.3	206.0	205.8	205.6
8	Steam temperature, F	626	794	824	698	748	730	566	564	884	869
9	Condensate temperature, F	74.7	75.6	73.8	72.9	74.6	74.0	73.1	73.5	75.7	75.8
10	Air temperature, F	72.5	73.7	74.3	71.7	74.6	73.3	72.6	72.1	77.5	75.8
11	Inlet-water temperature observed, F	72.85	75.01	71.76	71.94	73.66	72.89	72.28	72.70	74.61	74.48
12	Outlet-water temperature observed, C	55.35	54.59	56.90	55.24	55.04	55.22	54.97	54.72	56.73	56.35
13	Stem temperature, inlet thermometer, F	77.7	79.7	81.3	76.7	80.0	79.3	78.5	78.0	85.5	82.5
14	Stem temperature, outlet thermometer, C	24.8	25.0	27.6	24.3	25.8	25.6	25.0	24.7	28.5	26.6
15	Stem correction, inlet-water thermometer, F	−0.018	−0.018	−0.035	−0.018	−0.025	−0.024	−0.023	−0.020	−0.043	−0.032
16	Stem correction, outlet-water thermometer, C	0.299	0.287	0.295	0.303	0.285	0.290	0.293	0.291	0.283	0.296
17	Inlet-water temperature, corrected, F	72.83	74.99	71.72	71.92	73.64	72.86	72.26	72.68	74.57	74.45
18	Outlet-water temperature, corrected, F	132.16	130.78	134.95	131.98	131.58	131.92	131.47	131.02	134.62	133.95
19	Enthalpy of inlet water (h_1) Btu per lb	40.8	43.0	39.7	39.9	41.7	40.9	40.3	40.7	42.6	42.5
20	Enthalpy of outlet water (h_2) Btu per lb	100.0	98.6	102.8	99.8	99.4	99.8	99.3	98.9	102.5	101.8
21	Enthalpy of condensate (h_c) Btu per lb	43.1	44.0	42.3	41.4	43.1	42.5	41.6	42.0	44.1	44.2
22	Apparent total enthalpy of steam (h_g) Btu per lb	1332.0	1419.1	1432.6	1368.1	1396.4	1386.4	1303.0	1298.6	1464.6	1456.7
23	Corresponding h_g from Keenan Tables, Btu per lb	1334.3	1422.3	1438.0	1371.8	1398.1	1388.6	1302.5	1301.4	1469.4	1461.6
24	Heat leak, Btu per min	5.12	2.39	6.39	4.68	1.44	2.20	−0.56	3.02	6.05	6.38

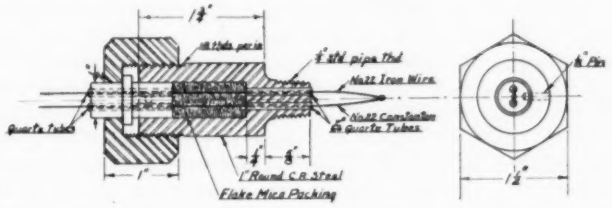


FIG. 3 THERMOCOUPLE FOR STEAM AND CONDENSATE TEMPERATURES

except at three points: (1) steam entrance, (2) condensate discharge, and (3) cooling-water discharge. The greatest loss of heat will occur at the first point, since there the temperature difference is much greater than at either of the other two places. With the inlet cooling water maintained at room temperature and with a constant temperature rise of the cooling water, the heat leak from the calorimeter per unit of time was assumed to be a function of steam temperature only. Changes in condensate temperature with different rates of flow will introduce a slight error, but if the calorimeter is provided with sufficient condensing surface, the condensate will be cooled to within a few degrees of the inlet-water temperature and variations in the heat leak at the condensate outlet will be very small.

On comparing recent steam tables it was found that there is now very close agreement on all the thermal properties of steam at pressures below 350 lb per sq in. It was decided, therefore, to calibrate the calorimeter for the heat leak by subtracting each calculated value of enthalpy, as obtained from test data, directly from the corresponding value in the Keenan Tables. The heat leak is then this difference multiplied by the weight of steam condensed and divided by the duration of the test. In order to avoid the expense of operating the high-pressure steam generator for the calibration tests, it was decided to utilize steam from the laboratory header at about 200 lb per sq in. A special gas-fired superheater was used to permit adjustment of the steam temperatures to any desired point between saturation and about 900 F. The arrangement of the equipment is shown schematically in Fig. 4.

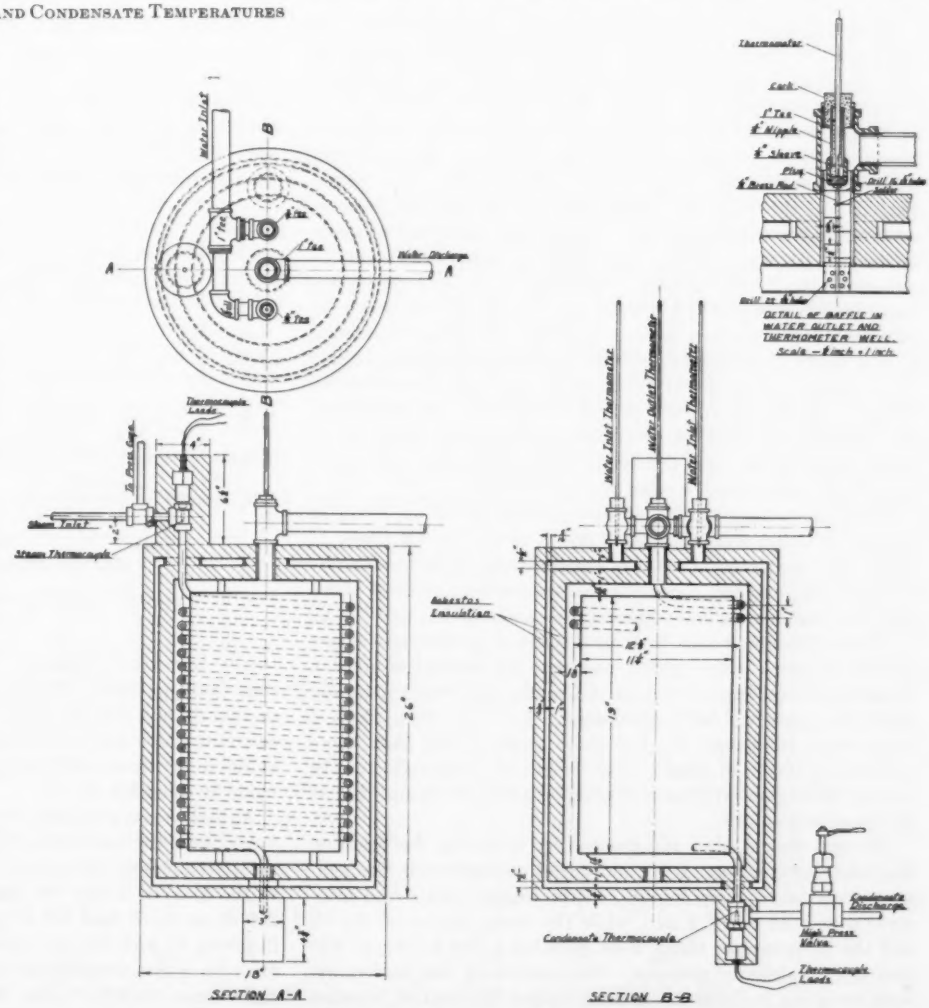


FIG. 2 HIGH-PRESSURE CONDENSING CALORIMETER DETAILS

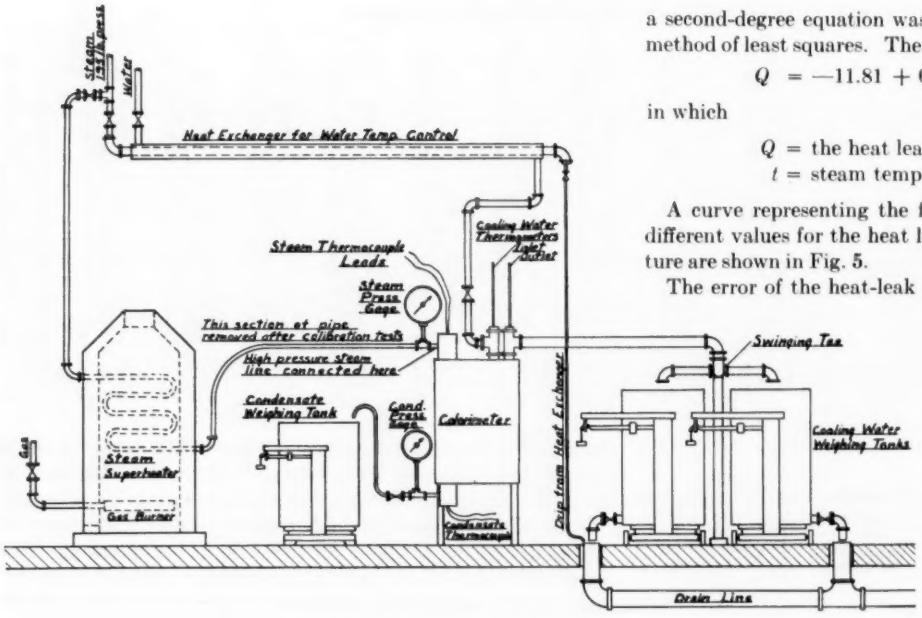


FIG. 4 ARRANGEMENT OF EQUIPMENT

Since the specific volume of steam at low pressures is about 30 times that of the high-pressure steam for which the calorimeter was designed, large pressure drops occurred in the condensing coil during the calibration tests. With steam at 195 lb per sq in. pressure and 700 F, the maximum capacity of the calorimeter was found to be 4 lb per min. At this rate there was a pressure drop of about 25 lb between the inlet and the outlet of the superheater, and the remaining drop down to atmospheric pressure occurred in the calorimeter coil. At the maximum rate of steam flow of 4 lb per min, a steam temperature of about 700 F could be obtained with the superheater. At 2 lb per min the temperature could be raised to 850 F, while at a rate of 1 lb per min temperatures of 900 F and above were obtainable. By proper manipulation of the heat-exchanger steam and condensate valves, changes of less than $^1/_2$ deg could be made in the water temperature. When condensing steam at a rate of 4 lb per min, the condensate temperature at discharge was found to be about 3 deg above room temperature, and the temperature difference was correspondingly less at smaller rates of condensation.

There were altogether 20 calibration tests carried on, covering a range of steam temperatures of 564 F to 884 F. Of these, 11 tests were conducted at rates of condensation of approximately 2 lb per min, and the remaining 9 were made at rates of about half this amount. It had previously been found that a test of 20 to 30 min duration would, in general, yield as good results as one extending over a longer period, and that the amount of error in weighing the cooling water and the condensate was apparently much less than had been anticipated. Tests in which 25 lb of steam were condensed did not show greater error than those condensing 100 lb of steam. The weight of condensate for the tests at the higher rate was accordingly fixed at 50 lb and that at the lower rate at 25 lb.

All tests were started and stopped by balancing the beam on the condensate scales. Inlet and outlet cooling-water temperatures were taken every minute, thermocouples and steam pressures were read every 2 min, while the temperatures of the air and the thermometer stems were recorded every 6 min, as was also the condensate pressure. The results of the calibration tests are given in Table 1. In order to find the relation between rate of the heat leak from the calorimeter and steam temperature, a second-degree equation was derived from the test data by the method of least squares. The equation so obtained is:

$$Q = -11.81 + 0.03227t - 0.0000144t^2$$

in which

Q = the heat leak in Btu per min
t = steam temperature, deg F

A curve representing the foregoing equation and also the 20 different values for the heat leak plotted against steam temperature are shown in Fig. 5.

The error of the heat-leak equation was calculated and found to be 2.32 Btu per min. Since the heat leakage at 900 F is only 5.5 Btu per min, as calculated from the equation, it is apparent that the combined effect of the errors of observation is almost as great as the amount of heat lost from the calorimeter. The error contained in the heat-leak equation, however, does not have a very great effect upon the final result for total enthalpy. The highest rate of heat leakage indicated by the calibration tests (Table 1) was given by test No. 1 as 9.05 Btu per min. If this were entirely neglected, the difference between the enthalpy obtained from the Keenan Tables and the enthalpy as calculated would be 4.1 Btu per lb, an error of 0.28 per cent. From Fig. 5 the heat leak for test No. 1 is found to be 5.2 Btu per min. Multiplying this by the total time and dividing by the weight of steam condensed gives 2.4 Btu per lb to be added

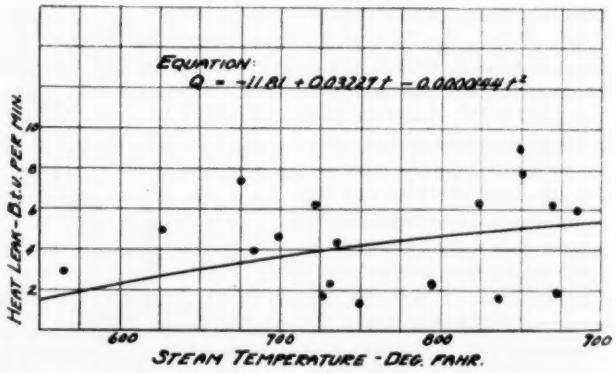


FIG. 5 VARIATION OF CALORIMETER HEAT LEAKAGE WITH STEAM TEMPERATURE

to the calculated enthalpy. The final value for the enthalpy then becomes 1449.6 Btu per lb, producing an error of 1.7 Btu, or slightly less than 0.12 per cent. This indicates the degree of accuracy which may be obtained with this calorimeter with steam at 200 lb per sq in., after the correction for the heat leak has been made by using Fig. 5.

At high steam pressures the final effect of all errors of measurement, with the exception of that of steam temperature, should be approximately the same as at the lower pressure. According to the Keenan Tables the specific heat of superheated steam at 200 lb per sq in. and 725 F is approximately 0.52, while at 3300 lb per sq in. and the same temperature it is about 4.2. A given error in steam-temperature measurement will thus have over eight times the effect upon the calculated value of enthalpy at the higher pressure than at the lower.

Preliminary Enthalpy Determinations at High Pressures

A test procedure with the condensing calorimeter quite similar to that developed for the calibration tests (Table 1) was used for the preliminary enthalpy measurements at high pressures. The same method was employed in starting and stopping the tests and the same observations were taken. The desired steam pressure and temperature were obtained by proper adjustment of the controls of the high-pressure steam generator. The operator of the steam generator held the pressure and temperature as steady as possible during each test and also recorded the steam pressure as indicated by the Bailey dead-weight pressure gage. The steam-thermocouple millivolts were read at 2-min intervals for all tests except the last three, when they were recorded each minute during the test. There were 20 preliminary enthalpy measurements made at high pressures. The rate of steam condensation for the first six tests was adjusted to approximately 3 lb per min, and for the remaining tests a rate of about 4 lb per min was used. The total steam condensed in most of the tests was either 70 or 75 lb, the duration ranging from about 17 min to 26 min, with an average of about 20 min.

The rate of the heat leak from the calorimeter was obtained from Fig. 5. The enthalpy of the condensate was found by adding to the enthalpy of saturated liquid, corresponding to the condensate temperature, the increase in enthalpy due to compression to the condensate pressure. This gain in enthalpy was obtained by a large-scale graphic interpolation of the Keenan Compressed Liquid Water Table published in 1931. During all of the high-pressure tests, steam from the laboratory main was condensed to supply the small amount of make-up feedwater required for the steam generator, using trisodium phosphate for conditioning this water. The water in the hot well was maintained at or near the boiling point, and the hot well was vented to the atmosphere in order to eliminate the non-condensible vapors as completely as possible. The results of the preliminary enthalpy measurements are summarized in Table 2. In Fig. 6 these data are plotted on pressure-enthalpy coordinates. Eight superheated-steam isotherms, plotted from the Keenan Tables (1930), are also included for comparison.

The results of tests 7 to 15 inclusive are not reported as they were inconsistent in themselves and were at wide variance with published steam-table data. The difficulty was traced to a defective steam-temperature thermocouple. The fused-quartz insulators, shown in Fig. 5, had been dissolved under the influence of the high-pressure steam and had corroded the thermocouple wires. The thermocouple was rebuilt without these insulators, and the results of tests 16 to 20 were then obtained. The accuracy of the data in test 6 is also open to question due to the possible failure of the original thermocouple at this point. It is seen that while the calorimetric measurements in tests 16 to 20 are all below those of Keenan, the difference is in no case greater than three-tenths of 1 per cent. At 3510 lb per sq in., where the divergence might be expected to be greatest, the two values agree

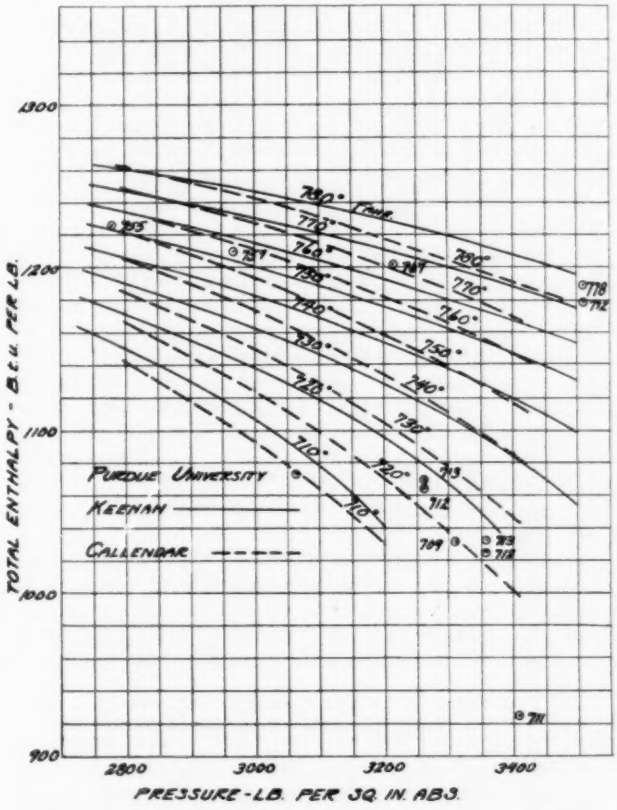


Fig. 6 Enthalpy Values From Preliminary Tests

TABLE 2 PRELIMINARY HIGH-PRESSURE TESTS

1	Test number	1	2	3	4	5	6	16	17	18	19	20
2	Date	4-21	4-21	4-21	4-22	4-22	4-22	5-11	5-11	5-11	5-11	5-11
3	Duration, min	24.1	23.1	22.85	24.18	24.98	26.33	17.42	18.19	17.91	16.57	16.58
4	Total steam condensed, lb	72	70	70	71	70	70	70	70	71	70	70
5	Total cooling water, lb	1382.8	1315.5	1308.5	1219.8	1278.0	1100.5	1277.0	1331.8	1277.5	1183.8	1187.3
6	Pounds cooling water per lb steam	19.21	18.79	18.69	17.18	18.26	15.72	18.24	19.03	17.99	16.91	16.96
7	Steam pressure, lb per sq in. abs	3311	3260	3261	3360	3360	3410	2778	2966	3215	3509	3510
8	Steam temperature, F	709.3	711.6	713.1	712.6	713.2	711.3	754.5	756.6	767.1	778.2	772.0
9	Condensate temperature, F	86.6	87.8	88.1	87.6	87.1	90.3	81.2	80.7	81.0	81.0	80.5
10	Air temperature, F	84.9	88.1	87.4	86.0	89.9	92.5	78.8	78.7	78.2	77.8	78.3
11	Inlet-water temperature observed, F	84.19	86.20	86.45	86.15	85.72	89.12	78.09	77.99	78.00	77.21	77.29
12	Outlet-water temperature observed, C	56.80	59.46	59.88	60.92	59.11	61.76	60.91	58.96	60.59	61.93	61.57
13	Stem temperature, inlet thermometer, F	88.3	93.8	94.3	88.5	92.5	97.3	83.3	83.0	81.3	80.3	81.0
14	Stem temperature, outlet thermometer, C	30.4	33.5	33.6	31.0	33.2	35.7	27.8	27.2	26.5	26.3	26.7
15	Stem correction, inlet-water thermometer, F	—0.020	—0.038	—0.040	—0.011	—0.033	—0.042	—0.022	—0.021	—0.014	—0.013	—0.016
16	Stem correction, outlet-water thermometer, C	0.265	0.272	0.277	0.320	0.270	0.283	0.354	0.331	0.363	0.387	0.377
17	Inlet-water temperature, corrected, F	84.17	86.16	86.41	86.14	85.68	89.08	78.06	77.97	77.98	77.19	77.28
18	Outlet-water temperature, corrected, F	134.71	139.52	140.27	142.24	138.88	143.67	142.28	138.72	141.72	144.17	143.50
19	Enthalpy of inlet water (h_i) Btu per lb	52.2	54.2	54.4	54.1	53.7	57.1	46.1	46.0	46.0	45.2	45.3
20	Enthalpy of outlet water (h_o) Btu per lb	102.6	107.4	108.1	110.1	106.7	111.5	110.1	106.6	109.6	112.0	111.3
21	Enthalpy of condensate (h_c) Btu per lb	63.4	64.5	64.8	64.4	64.0	67.3	56.7	56.7	57.6	58.5	58.0
22	Heat leakage, Btu per min	3.80	3.80	3.80	3.80	3.80	3.80	4.40	4.40	4.50	4.60	4.60
23	Heat-leak correction Btu per lb	1.27	1.25	1.24	1.29	1.36	1.43	1.10	1.14	1.13	1.09	1.09
24	Total enthalpy of steam, Btu per lb	1032.4	1065.4	1070.0	1026.6	1033.7	924.5	1226.4	1210.6	1202.7	1189.4	1179.4
25	Keenan, interpolated, Btu per lb	.....	.....	.....	.....	.....	.....	1229.9	1213.6	1204.2	1192.2	1180.1
26	Difference, Btu per lb	.....	.....	.....	.....	.....	.....	3.5	3.0	1.5	2.8	0.7
27	Per cent difference	.....	.....	.....	.....	.....	.....	0.286	0.249	0.125	0.235	0.059

TABLE 3 ENTHALPY OF STEAM AT HIGH PRESSURES

1	Test number	1	2	3	4	5	6	7	8	9	10	11	13	14	15
2	Date	6–6	6–6	6–6	6–6	6–6	6–6	6–6	6–6	6–6	6–7	6–7	6–7	6–7	6–7
3	Duration, min	19.16	17.90	16.77	20.78	19.95	18.75	18.40	18.21	17.89	18.62	18.79	18.66	19.62	20.24
4	Total steam condensed, lb	70	70	70	70	70	70	70	70	70	70	70	70	70	70
5	Total cooling water, lb	1344.5	1220.5	1151.5	1366.0	1183.5	1193.5	1122.0	1146.0	1121.0	1278.0	1378.5	1336.0	1221.0	1205.5
6	Pounds cooling water per lb steam	19.21	17.44	16.45	19.51	16.91	17.05	16.04	16.37	16.01	18.39	19.69	19.09	17.44	17.22
7	Steam pressure, lb per sq in. abs	1900	2200	2500	2800	3000	3200	3300	3400	3100	2000	2200	2800	3000	3100
8	Steam temperature, F	728.6	729.9	731.8	720.8	734.7	737.3	734.1	736.0	726.0	751.2	748.9	750.7	749.2	748.0
9	Condensate temperature, F	94.5	94.0	92.4	92.9	91.2	95.0	95.2	93.5	94.2	95.0	99.1	99.5	98.5	99.1
10	Air temperature, F	90.0	90.8	89.7	90.3	90.9	92.4	92.1	91.8	94.9	92.6	95.1	96.6	95.8	96.9
11	Inlet-water temperature observed, F	91.02	90.09	88.26	90.62	88.32	91.57	91.73	91.04	91.16	90.51	94.36	95.69	95.83	95.96
12	Outlet-water temperature observed, C	67.59	69.68	70.14	63.87	67.41	67.89	69.31	67.82	69.54	69.22	68.30	68.51	70.82	70.94
13	Stem temperature, inlet thermometer, F	93.0	93.3	92.3	93.3	92.8	94.3	94.8	94.0	98.8	94.8	97.8	99.3	98.3	100.3
14	Stem temperature, outlet thermometer, C	33.1	33.5	32.4	33.2	33.3	34.6	34.4	34.3	36.4	34.3	36.0	36.6	36.3	37.2
15	Stem correction, inlet-water thermometer, F	—0.01	—0.02	—0.02	—0.03	—0.04	—0.03	—0.03	—0.03	—0.07	—0.04	—0.03	—0.04	—0.03	—0.04
16	Stem correction, outlet-water thermometer, C	0.40	0.44	0.46	0.34	0.40	0.39	0.42	0.40	0.40	0.42	0.39	0.38	0.42	0.42
17	Inlet-water temperature, corrected, F	91.01	90.07	88.24	90.59	88.28	91.54	91.70	91.01	91.08	90.47	94.33	95.65	95.80	95.91
18	Outlet-water temperature, corrected, F	154.39	158.21	159.03	147.58	154.05	154.91	157.52	154.78	157.90	157.35	155.64	156.01	160.23	160.44
19	Enthalpy of inlet water (h_1) Btu per lb	59.0	58.1	56.2	58.6	56.3	59.5	59.7	59.0	59.1	58.5	62.3	63.6	63.9	63.9
20	Enthalpy of outlet water (h_2) Btu per lb	122.2	126.1	126.9	115.4	121.9	122.8	125.4	122.6	125.8	125.2	123.5	132.9	128.1	128.3
21	Enthalpy of condensate (h_c) Btu per lb	67.6	67.9	67.1	68.3	67.2	71.4	71.9	70.5	70.4	68.3	72.9	74.8	74.3	75.2
22	Heat leakage, Btu per min	4.10	4.10	4.10	4.00	4.20	4.20	4.20	4.20	4.00	4.30	4.30	4.30	4.30	4.30
23	Heat-leak correction, Btu per lb	1.12	1.05	0.98	1.19	1.20	1.12	1.10	1.09	1.03	1.14	1.15	1.15	1.21	1.24
24	Total enthalpy of steam, Btu per lb	1283.4	1254.3	1230.1	1178.8	1177.8	1150.6	1126.8	1113.4	1139.1	1296.2	1278.9	1225.9	1197.5	1185.9
25	Keenan, interpolated, Btu per lb	1279.5	1254.7	1227.4	1172.8	1172.5	1146.6	1120.7	1105.4	1135.9	1292.7	1274.6	1222.3	1197.9	1183.5
26	Difference, Btu per lb	3.9	—0.4	2.7	6.0	5.3	4.0	6.1	8.0	3.2	3.7	4.3	3.6	—0.4	2.4
27	Per cent difference	0.30	0.03	0.22	0.51	0.45	0.35	0.54	0.72	0.28	0.29	0.34	0.29	0.03	0.20

TABLE 3 ENTHALPY OF STEAM AT HIGH PRESSURES—(*Continued*)

1	Test number	16	17	18	19	20	21	22	23	24	25	26	27	28	30
2	Date	6–7	6–7	6–7	6–7	6–10	6–10	7–16	7–17	7–16	7–16	7–16	7–16	7–20	7–20
3	Duration, min	19.07	17.54	18.07	16.99	18.08	17.57	19.02	19.28	20.19	20.52	17.75	17.28	20.62	17.31
4	Total steam condensed, lb	70	70	71	65	70	70	70	70	70	70	70	70	70	70
5	Total cooling water, lb	1194.5	1184.0	1160.0	1054.0	1264.0	1278.5	1084.0	1048.5	1102.5	1187.0	977.0	1078.5	1241.5	1088.5
6	Pounds cooling water per lb steam	17.06	16.91	16.34	16.22	18.00	18.26	15.49	14.98	15.76	16.96	13.96	15.41	17.74	15.55
7	Steam pressure, lb per sq in. abs	3200	3300	3400	3395	2160	2170	2800	3000	3100	3200	3300	3400	1900	2500
8	Steam temperature, F	750.0	755.9	746.3	755.8	737.6	772.5	774.4	774.4	779.0	779.0	766.2	762.0	739.0	778.0
9	Condensate temperature, F	98.2	97.9	97.8	99.4	92.7	94.7	101.5	106.0	105.7	103.2	105.0	107.7	102.3	99.8
10	Air temperature, F	95.4	95.2	95.2	98.5	91.1	91.8	101.8	101.5	101.6	101.6	101.0	100.5	105.2	105.7
11	Inlet-water temperature observed, F	94.91	94.24	93.61	94.31	89.92	90.30	91.47	96.64	96.66	93.46	95.21	98.83	96.16	91.44
12	Outlet-water temperature observed, C	70.56	70.48	70.12	71.45	68.74	69.62	74.89	78.41	76.03	71.40	79.03	76.20	73.24	75.61
13	Stem temperature, inlet thermometer, F	99.0	98.3	99.0	102.3	94.5	95.8	106.0	105.5	105.7	106.0	104.0	104.7	106.3	109.5
14	Stem temperature, outlet thermometer, C	36.1	36.1	36.7	38.2	33.8	34.7	40.0	39.9	40.0	40.0	39.4	39.0	40.4	42.6
15	Stem correction, inlet-water thermometer, F	—0.04	—0.04	—0.05	—0.08	—0.04	—0.05	—0.13	—0.08	—0.09	—0.12	—0.08	—0.06	—0.10	—0.16
16	Stem correction, outlet-water thermometer, C	0.42	0.42	0.41	0.41	0.42	0.42	0.45	0.56	0.47	0.39	0.54	0.49	0.47	0.43
17	Inlet-water temperature, corrected, F	94.87	94.20	93.56	94.24	89.88	90.25	91.34	96.56	96.57	93.35	95.13	98.77	96.06	91.28
18	Outlet-water temperature, corrected, F	159.76	159.62	158.95	161.36	156.48	158.07	167.62	174.14	169.71	161.22	174.22	170.04	164.58	168.86
19	Enthalpy of inlet water (h_1) Btu per lb	62.8	62.2	61.5	62.2	57.9	58.2	59.3	64.5	64.5	61.3	63.1	66.7	64.0	59.3
20	Enthalpy of outlet water (h_2) Btu per lb	127.6	127.5	126.8	129.2	124.3	125.9	135.5	142.0	137.6	129.1	143.1	137.9	132.4	136.7
21	Enthalpy of condensate (h_c) Btu per lb	74.5	74.4	74.7	76.2	66.5	68.5	76.6	81.4	81.6	79.4	81.3	84.3	75.2	74.2
22	Heat leakage, Btu per min	4.30	4.40	4.30	4.40	4.20	4.60	4.60	4.60	4.60	4.60	4.50	4.40	4.20	4.60
23	Heat-leak correction, Btu per lb	1.17	1.10	1.09	1.15	1.08	1.15	1.25	1.27	1.32	1.35	1.12	1.09	1.23	1.14
24	Total enthalpy of steam, Btu per lb	1181.0	1180.1	1141.9	1163.8	1267.5	1305.7	1256.8	1243.2	1234.2	1230.0	1198.9	1182.3	1289.7	1279.6
25	Keenan, interpolated, Btu per lb	1174.5	1173.4	1136.6	1159.7	1266.2	1300.2	1254.1	1236.6	1234.2	1225.0	1193.2	1173.0	1289.1	1282.2
26	Difference, Btu per lb	6.5	6.7	5.3	4.1	1.3	5.5	2.7	6.6	0	5.0	5.7	9.3	0.6	2.4
27	Per cent difference	0.55	0.57	0.46	0.35	0.10	0.42	0.21	0.53	0.00	0.41	0.48	0.79	0.05	0.19

TABLE 3 ENTHALPY OF STEAM AT HIGH PRESSURES—(*Continued*)

1	Test number	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45
2	Date	11–1	11–1	11–1	11–1	11–8	11–8	11–8	11–8	11–8	11–8	11–8	11–8	11–15	11–15	11–15
3	Duration, min	20.10	21.33	18.66	17.75	16.66	18.25	17.00	18.25	19.66	20.25	20.25	20.25	17.25	17.50	22.00
4	Total steam condensed, lb	68.25	67.91	68.25	69.25	68.00	67.56	67.63	68.18	67.88	67.71	68.13	68.40	68.63	68.47	67.47
5	Total cooling water, lb	1249.5	1334.5	1161.5	1149.0	988.5	1022.0	980.0	1062.0	1146.5	1181.0	1163.5	1156.0	1025.0	1036.8	1324.5
6	Pounds cooling water per lb steam	18.30	19.65	17.01	16.59	14.53	15.12	14.49	15.57	16.89	17.44	17.07	16.90	14.93	15.14	19.63
7	Steam pressure, lb per sq in. abs	3400	3300	3200	3000	3400	3300	3100	3000	2800	2800	3000	3100	3200	3100	2800
8	Steam temperature, F	736.0	735.0	735.0	738.0	732.0	735.0	733.0	730.0	734.0	720.0	723.0	726.0	719.0	709.0	708.0
9	Condensate temperature, F	91.6	92.1	92.7	92.2	82.0	82.8	83.7	82.4	80.1	79.6	80.5	81.1	86.1	85.0	80.5
10	Air temperature, F	80.4	81.0	81.4	78.0	81.5	82.6	83.6	83.6	83.2	83.2	82.0	82.4	77.0	77.9	76.0
11	Inlet-water temperature observed, F	81.53	82.67	83.12	81.51	76.31	77.91	78.70	78.07	76.96	76.58	77.20	77.51	73.81	72.99	71.23
12	Outlet-water temperature observed, C	58.82	57.56	63.26	64.11	63.73	64.20	67.20	64.86	62.12	59.96	60.28	60.36	61.01	59.64	52.56
13	Stem temperature, inlet thermometer, F	84.0	84.3	86.2	82.0	90.0	90.0	86.0	85.8	84.0	85.6	86.0	86.0	81.6	81.0	80.0
14	Stem temperature, outlet thermometer, C	85.4	85.8	88.2	84.4	94.5	94.0	88.0	87.8	86.0	87.6	88.0	88.0	82.8	83.0	81.0
15	Stem correction, inlet-water thermometer, F	—0.02	—0.01	—0.02	—0.01	—0.13	—0.12	—0.07	—0.07	—0.07	—0.09	—0.08	—0.08	—0.07	—0.08	—0.08
16	Stem correction, outlet-water thermometer, C	0.39	0.37	0.46	0.50	0.33	0.34	0.43	0.40	0.36	0.31	0.32	0.32	0.36	0.34	0.24
17	Inlet-water temperature, corrected, F	81.51	82.66	83.10	81.50	76.18	77.79	78.63	78.00	76.89	76.49	77.12	77.43	73.74	72.91	71.15
18	Outlet-water temperature, corrected, F	138.56	136.25	146.69	148.31	147.30	148.17	153.73	149.46	144.46	140.48	141.08	141.22	142.47	139.96	127.04
19	Enthalpy of inlet water (h_i) Btu per lb	49.5	50.7	51.1	49.5	44.2	45.8	46.6	46.0	44.9	44.5	45.1	45.4	41.8	40.9	39.2
20	Enthalpy of outlet water (h_o) Btu per lb	106.4	104.1	114.5	116.2	115.1	116.0	121.6	117.3	112.3	108.3	108.9	109.6	110.3	107.8	94.9
21	Enthalpy of condensate (h_c) Btu per lb	67.9	68.1	67.7	68.1	59.0	59.6	60.0	58.3	55.5	54.9	56.4	57.3	62.4	61.3	55.9
22	Heat leakage, Btu per min.	4.12	4.12	4.12	4.12	3.94	3.94	3.94	3.88	3.94	3.82	3.88	3.88	3.88	3.76	3.76
23	Heat-leak correction, Btu per lb	1.21	1.29	1.12	1.05	0.96	1.06	0.99	1.04	1.14	1.13	1.15	1.14	0.97	0.96	1.22
24	Total enthalpy of steam, Btu per lb	1110.5	1119.6	1147.9	1174.7	1090.9	1122.3	1147.0	1169.4	1194.9	1169.1	1146.4	1133.6	1086.8	1074.6	1151.0
25	Keenan, interpolated, Btu per lb	1105.0	1122.4	1140.6	1177.7	1090.9	1122.4	1151.7	1161.7	1196.4	1171.3	1145.3	1134.9	1090.1	1074.0	1144.7
26	Difference, Btu per lb	4.5	—2.8	7.3	—3.0	0.0	—0.1	—4.7	7.7	—1.5	—2.2	1.1	—1.3	—3.3	0.6	6.3
27	Per cent difference	0.41	0.25	0.64	0.26	0.00	0.01	0.41	0.66	0.13	0.19	0.10	0.11	0.30	0.06	0.60

within about one-tenth of 1 per cent. Tests 1 to 6 are outside of the range of the Keenan Tables.

Determination of Enthalpy at Pressures From 1900 to 3500 Lb per Sq In.

Supplementing the preliminary results which are recorded in Table 2 and Fig. 6, 30 tests were conducted along isotherms of 730, 750, and 775 F and at pressures varying from 1900 to 3400 lb per sq in. The results are reported in Table 3 and Fig. 7.

In general, the testing procedure included during any one day a series of tests at a constant steam temperature and with different pressures. Consequently, after establishing steady temperature conditions, the time required between tests was determined by the ability of the operator to change and to stabilize the operating conditions on the steam generator. The flow rate through the calorimeter was held as constant as possible and the total steam condensed per test was about 70 lb. With this operating procedure, the heat-leak correction was reasonably constant for all tests conducted during the particular day in question.

The differences in the total enthalpy of steam between the observed results and those of Keenan (1930), as interpolated from the Steam Tables, varied from —0.4 Btu per lb to 9.3 Btu per lb, corresponding to a maximum of 0.79 per cent. Only seven of the 28 tests recorded in Table 3 indicated a divergence from the Keenan values of more than one-half of 1 per cent, the average for all tests being about one-third of 1 per cent. Two of the determinations were slightly below the corresponding Keenan values, one was in exact agreement, while the remaining 25 gave values somewhat higher. The results of tests 12 and 39 were not recorded in Table 3, as they were conducted under unsteady steam-generator operating conditions.

Conclusions

The determinations of the enthalpy of steam reported in this paper were undertaken at the suggestion of an organization which was largely instrumental in making possible the installation of the high-pressure steam generator at Purdue University. The main object of these determinations was the comparison of the data secured with the values of Callendar, Keenan, Knoblauch, and the International Tables.

The results of the Purdue University tests are apparently in close agreement with those of Keenan. Due to the fact that the Purdue University tests do not go below 700 F or above 800 F, it is not practical to make comparisons with the values of Knoblauch, Callendar, or the International Tables other than throughout the range of the 752 F isotherm from 2100 lb per sq in. to 3556 lb per sq in. abs pressure. The results of the Purdue University tests are slightly lower than those of Knoblauch, but higher than the values in the International Tables. The values of Callendar are higher than the Purdue University results from 2100 lb per sq in. to about 2800 lb per sq in. abs, but lower from 2800 lb per sq in. to the end of the test results. Fig. 7 indicates that the results of the Purdue University tests are in fairly good agreement with the values reported by Keenan and Knoblauch, but seem to vary in the higher pressure ranges from the Callendar 750 F isotherm. It is recognized that the condensing calorimeter is not a primary device for the direct measurement of the enthalpy of steam, since it requires calibration for heat leak, and reliable data on the enthalpy of saturated and subcooled water at temperatures between 70 and 140 F are necessary. Data now available should eliminate errors from the causes named. The Purdue University tests may be criticized by reason of the fact that pure steam was not used. The authors believe that any appreciable error from this cause should be reasonably constant if feedwater conditions are properly controlled.

It is desirable that further refinements be made in the measurement of steam and cooling-water temperatures. This is particularly true when working with steam near its critical pressure and temperature because of the high specific heat. The generation of high-pressure steam in a unit of semi-commercial size, with close control of temperature and pressure, and its condensation in a calorimeter involve apparatus that must be handled with great care if the accuracy of the physics laboratory is to be approached. It is hoped that the close agreement of the results

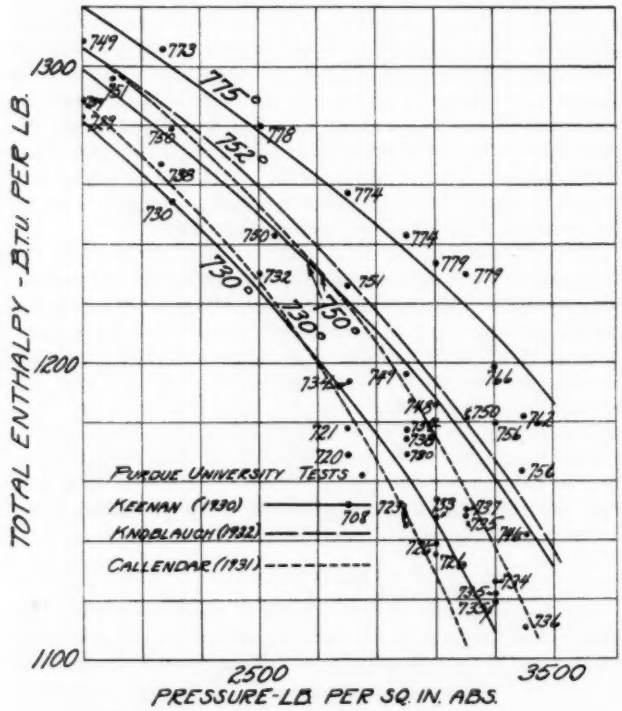


FIG. 7 ENTHALPY OF HIGH-PRESSURE STEAM

presented with the published values of the Keenan and Knoblauch tables over the pressure and temperature range investigated may prove of interest and value. Difficulties with some of the auxiliaries for the high-pressure generator prevented a more complete investigation with the condensing calorimeter. Accordingly, the authors invite suggestions as to the advisability of extending these tests to include a more comprehensive survey of the enthalpy of steam at pressures between 1500 and 3500 lb per sq in.

Discussion

ROBERT C. H. HECK.[5] In general method of heat measurement—that is, in using a water calorimeter to absorb and measure the heat of a sample of steam—these experiments go back to the method of Regnault. Modern steam research, especially in the latter half of its 25-year period, has preferred the alternative scheme of supplying heat in the primary form of closely measured electrical energy. Chiefly, the electrical-supply method offers possibility of more completely isolating or insulating the process.

However, both mechanical and thermal difficulties increase, as the apparatus with electrical supply must contain very high pressure. For this reason, a return to the older method is justified by the rugged simplicity of the steam coil in the calorimeter here described. Present use of the water-absorption method is not novel with the Purdue investigation, since it has been employed in recent years by Callendar in England and by Havlicek in Czechoslovakia. With an equally high grade of skill in physical measurement, there is no inherent reason why this method shall not give correct and reasonably precise results.

As noted by the authors, it is highly important that both pressure and temperature of the entering steam be measured precisely and simultaneously. Consider the region near the critical point in a plot of enthalpy h in pressure p for superheated steam. Not only are the isotherms so widely spaced that specific heat c_p is very high, but they also have such slopes that h varies quite rapidly with pressure. Thus at 3200 lb abs and 750 F, a difference of 1 Btu in h will be caused by about 0.5 deg of temperature or by 7.5 lb per sq in. of pressure. Close to the critical point the pressure influence becomes very strong.

A question arises as to the velocity and kinetic energy of the steam at the place where the entrance observations are taken. In the full-pressure experiments, velocity is so slow that this effect is insignificant, but in the calibration tests it may rise to an appreciable size. Thus at 195 lb abs and 750 deg, and at 2 lb per min of steam flow, the writer figures a velocity in the tube of about 266 ft per sec, equivalent to 1.4 Btu of kinetic energy. This will of course be modified by enlargement of passage and turbulence of current around the thermocouple.

To the writer's mind, the plot of heat-leak tests in Fig. 5 shows a degree of erraticity that will have to be rather strongly suppressed, by more complete insulation, if dependably accurate results are to be obtained.

The rather high range of cooling-water temperature, about 60 F, is favorable to accuracy of heat measurement. However, an error of 0.1 deg in this difference, or of 1 part in 600, is equivalent to more than 2 Btu in the enthalpy per pound of steam. From the scattering of results a probable error is inferred of as much as 0.2 deg in the temperature difference. Even to express the enthalpy of low-temperature water to but one decimal place is to fall below a precisely determinative value of this factor in heat absorption—but perhaps that is a mere tabular condensation from the justified two-place expression in the Keenan table.

To sum up, the investigation here described is a piece of high-grade work in the engineering laboratory, but it does not at all measure up to the accuracy of the work of the steam research program. To the more precise experimentation it makes a broadly confirmatory but not a closely determinative contribution. If the investigation is to be continued and is to give really valuable results, it must be graded up in the accuracy of the physical measurements made.

N. ARTSAY.[6] The paper shows the correctness of measuring instruments installed with the boiler and the dependability of their readings as checked by the condensing calorimeter. The authors should be congratulated on the close results they obtained with the relatively crude apparatus at their disposal. This preliminary work brings the whole installation in readiness for a series of experimental runs which might tackle the host of difficult problems so far poorly investigated or not touched at all.

The authors ask for suggestions on the advisability of extending further the enthalpy determinations. The data given in the paper show that a satisfactory reliance can be placed on the measuring instruments of the boiler, and a further checking of them seems to be superfluous for all practical purposes. So the following suggestions for the future work with the boiler are in order:

1 A definite program to cover several years of semi-commercial

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[6] Mechanical Engineer, New York, N. Y. Mem. A.S.M.E.

experimental work with the boiler should be prepared, widely spread, and discussed to attract interest and support of the manufacturers of high-pressure steam equipment and operating companies. Without their experience and support this undertaking cannot be of lasting practical value. It should be borne in mind that the Purdue installation offers the greatest chance of doing valuable work on a practical semi-commercial scale, and in planning the program any work properly belonging to a laboratory should be avoided.

2 Assuming that in a not so distant future the drumless boilers with forced circulation will make their logical appearance in the industrial and central-station field, the Purdue boiler and its set of instruments and apparatus should be made a center of experimental development work intended to solve the many practical difficulties inherent to the apparatus using extreme pressures, temperatures of steam, and actual feedwater. Scale formation, carry-over effects, instability of tube materials, valve troubles, resistance to flow—all this could be investigated. These results could be valuable not only for the coil boilers, of course.

3 The extremely important paper by De Baufre deals with the strength of boiler tubes; this problem needs experimental verification. Short lengths of tubes should be welded in the boiler coil in the spot of highest heat absorption, the tubes be selected of various diameters and thicknesses, the hours of service and operating conditions be recorded, and the shape of tubes be checked for ascertaining their relative behavior in a long service.

4 The failure of boiler tubes versus stresses and heat transfer has never been properly ascertained. A steel-protected refractory-lined chamber may be built and arranged to be fired with an independent gas burner. Tubes of various thicknesses and materials then could be tested to destruction and the question of the optimum thickness and material settled.

5 Efforts should be strengthened to develop a sensitive differential-pressure gage to investigate pressure drops through various pipes and fittings. The boiler coil, for this purpose, should be tapped and valved in several places to get different proportions of water and steam in the mixture or steam at various temperatures. The writer can offer some ideas for such an instrument.

6 Pieces of tubes of various steel should be tested for creep in the region of superheated steam to check the laboratory tension-creep rates versus actual behavior of the metal in the tube form and under compound stress.

7 A bypass tube should be arranged from a spot near the boiling point leading to the valve of a test set built to investigate the scale formation. A removable test stretch of a tube fixed in the set should be subjected to a high rate of heat transfer by firing a separate gas burner. Concentrated solutions of different salts could be then continuously injected in the bypassed flow and the scale formation investigated under varying pressures, temperatures, concentrations, mass velocities, and rates of heat transfer. The test tube must be easily removable for inner inspection. Observation of metal temperature would give actual data on heat conductivity of scale.

8 Manufacturers of high-pressure equipment and operating concerns should be polled on their individual problems and needs in experimental work. There is still a lot to do; for instance, in pipe and tube welds, flanges, flange bolts, gasket material, and so forth.

A wealth of valuable practical information could be gained through the tests of the Purdue boiler, but the success will depend much on the planning and cooperation of all concerned. The start has been made in the right direction, but much strenuous work is yet ahead.

W. F. Ryan.[7] The second paragraph of this paper states that a study is being made of the laws governing the fluid friction of steam at high pressure and at high temperature. This work will fill a very serious gap in the reliable data required by designers of steam plants and steam equipment. Further work on steam tables research can only make minute corrections to the data now available within the range of pressures and temperatures in practical use. Determination of the viscosity of high-pressure, high-temperature steam, however, will supply needed data in a field where there is practically no information whatever avilable.

The viscosity of superheated steam is the one remaining unknown factor which affects friction in pipe lines, turbine nozzles, turbine blades, etc., and a rational calculation of pressure drop or heat transfer, where high-pressure, high-temperature steam is involved, is now quite impossible. The empirical equations which have been developed by various manufacturers unfortunately lack universal application. The data which the authors intend to produce in this field are therefore of the greatest importance.

Perhaps the chief value of this paper is that it shows the accuracy with which measurements can be made with this apparatus, as Professor Heck has pointed out. This paper cannot be regarded as a check on the work of the Steam Tables Research Committee, but certainly the Keenan Tables can be regarded as a check on the accuracy and consistency of the work done by these authors with the existing apparatus. Therefore, when information is offered in regard to the laws affecting fluid friction of steam, the data will be received with the greatest confidence by all engineers.

Authors' Closure

The authors wish to express their appreciation of the many valuable suggestions that have been made relative to a program of experimental work which should be carried out on the high-pressure steam generator at Purdue University.

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Progress in Railroad Mechanical Engineering

IN COMMON with the experience of recent years, the railways are being called upon to consider and introduce many innovations which are profoundly changing their past methods of operation. While the field cannot be entirely covered in this survey, a number of the more widely known recently acquired practices are referred to under the various headings which follow.

Motive Power

The few new locomotives that have been placed in service are capable of hauling full-tonnage freight trains at unusually high sustained speed or they may be employed to handle extra-heavy passenger trains and thereby maintain the regular schedule. These units are so designed as to be best fitted for operation on long non-stop runs, the tenders having a capacity of at least 18,000 gal of water and 20 tons of coal. Four-wheel trailing trucks are used, together with booster, to accommodate large firebox arrangements and assist in starting heavy trains.

Multi-Pressure Locomotives. The first two multi-pressure-boiler locomotives placed in service on the American Continent are the Canadian Pacific 2-10-4-type wheel arrangement and the New York Central 4-8-4-type wheel arrangement, both of the three-cylinder-compound formation. The steam generation and utilization principles employed in the two locomotives have been employed in three locomotives in Europe, with what are described as giving very satisfactory results in fuel saving and overall efficiency, but the largest European unit of the type is only 42 per cent of the weight and develops only 36 per cent of the power of the Canadian Pacific multi-pressure one. These locomotives use steam at a pressure of 850 lb per sq in. in the high-pressure cylinder, whereas the best of our conventional form of locomotives use steam at 275 lb, so there is an increase in steam pressure of over three times. There are two superheaters, one for a steam pressure of 850 lb and one for 250 lb; there are two boiler-feed pumps, one for feeding to the low-pressure boiler and one from the low-pressure into the high-pressure boiler. There is the ordinary exhaust-steam feedwater heater for heating the water that is drawn from the tender before it goes into the first boiler.

Water Glasses. There have been some interesting departures from usual practice in water glasses used on the closed circuit and on the high-pressure boiler, carrying 1700 and 850 lb, respectively, on the Canadian Pacific and New York Central multi-pressure locomotives. Glasses in one shape or another have been used almost exclusively on locomotives and boilers for other purposes, but with increasing steam pressures, the tubular glass has gradually been replaced by thick "bull's eye" or heavy flat glasses, referred to in railroad practice as "Klinger." Glasses for these still higher pressures were ordered, but the manufacturer furnished what was to the railroads an innovation in that the glass was replaced by a thin sheet of mica. The manufacturer advised that experiments with high steam pressures had indicated that the thin mica sheets were more durable than the glass. While the flat glasses were deemed safer than the tubular glass they would break when the water level was no longer visible, although the glass held and did not shatter. Mica "glasses" have been in service now for over a year with very good results.

Poppet Valves. A departure of more than ordinary interest has been the application of Lentz and Caprotti poppet valves, fitted with both oscillating and rotary valve gear, to several hundred locomotives abroad and a number of test applications in this country. There appears to be a realization that the losses due to mechanical friction are much less in the case of poppet valves than in that of the piston type. In like manner, less energy is absorbed in actuating the valves, so that not only a theoretical but a practical gain of some degree is recorded. To determine the extent of this gain would necessitate the employment of a locomotive testing plant; otherwise the feature as to the degree of energy conservation must continue to remain a matter of judgment. Aside from any disposition to credit a better distribution of steam, an economy in maintenance cost realized thereby would be a further reason for the consideration of this construction, in view of its greater initial cost as compared with conventional designs. Long valve travel has been widely adopted for use on modern high-pressure locomotives in order to allow a relatively short cut-off to be employed.

Roller Bearings on Locomotives. A number of railways, chief among which are the New York Central and the Delaware, Lackawanna & Western, are successfully operating locomotives fitted completely with roller bearings on all axles. Prevailing opinion indicates that anti-friction bearings will make greater progress in the next few years on motive power than they will on cars, but it will be by reason of their contribution to reduced maintenance and freedom from attention rather than from any possibility of locomotive redesign which they may offer.

The valve motion and rods present a peculiarly well-adapted field for roller-bearing applications, since the absence of wear in them guarantees that the valve setting will remain exactly as initially intended when the locomotive was turned out of the shop. So far as the rods are concerned, it is found that anti-friction bearings materially reduce the need for bearing attention and there is no tendency toward that pounding which develops in the ordinary plain bearing locomotive and with the costly effects with which railway mechanical officers are thoroughly familiar.

The reduction of locomotive maintenance expense through the careful selection and adaptation of new ferrous-metal products has progressed with considerable success. Oftentimes with a change in material a modification design is necessary in order to improve a given part and minimize the risk for defects, due to improper or improperly controlled manufacturing practices. Increases in strength and ductility are obtained through the use of alloy elements of steel which are largely applicable in railway service. During the year a number of Diesel-electric locomotives were built with the cab, underframe, and trucks fabricated from structural-steel shapes and plates and arc welded. Bolts were employed only for those parts requiring occasional removal or renewal. It was found advisable to weld the body bolster directly to the side sills, the joint surfaces being reinforced with heavy gusset plates and brace plates, all welds being continuous.

The use of large high-tensile-strength steel castings has become general for the understructure of the locomotive and tender. The elimination of highly stressed bolts and rivets in locomotive parts has been a potent factor in this accomplishment. These cast-steel structures are really assembled in the mold in liquid

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form, and the metal can be placed in the right proportion exactly where it is needed for strength, thus securing weight economy and uniformity of fiber stress. This method of assembly permits the incorporation of the numerous necessary brackets, which in many cases are utilized to provide not only much greater strength for the detail part itself, but often to add to the value of the entire unit. Until eight years ago the cast-steel frame acted solely as a bedplate for the separate tank which it supported. With the increased demand for coal and water capacities, the length of tender increased to an intolerable limit, to say nothing of the weaving and buckling of such tanks causing leakage and cracking of the sheets. This led to the introduction of the water-bottom tender, which in addition to permitting more coal and water storage, provided a strong tender unit that combined the tank and frame with the solid corrosive-resistant bottom face of the frame, which in turn forms the lower member of a deep box-type girder. This construction permitted a lower center of gravity and eliminated not only the bottom tank sheet maintenance, but because of its greater rigidity added the necessary strength to the tank structure as a whole to permit of economical maintenance. The steam locomotive bed was first made with separate cylinders, which was followed with the cylinders being made integral with the locomotive frames.

Rolling Stock

The Pennsylvania Railroad placed in service a new type of motor car for the Long Island Railroad. It is a double decker and will seat 44 more persons than the suburban passenger coach now in use, with its capacity of 76, the new type carrying 120. This car has two tiers of seats, one level being above the other. The cars have a center aisle, as in cars now used, and on each side of this aisle seats are placed, facing forward, each seat being roomy enough for two persons. The floor board of this lower tier of seats is 14 in. below the aisle, so that the seats are almost flush with the aisle floor. Passengers will descend one step to the lower-tier seats. The floorboard of the upper-tier seats is above the central aisle. Passengers mount two steps to the higher seats. These pairs of seats are not directly above the lower ones, but the upper and lower pairs are staggered, one lower-tier pair of seats being succeeded by an upper-tier pair as the passenger walks down the car. The sunken seats in the lower tier permit the upper-tier seats to overhang, with still enough space for headroom. Windows are so placed that each seat, upper and lower, has its window.

Further attention has been given to passenger-car design to meet special conditions. Combination cafe-parlor cars have been used for some years, but now there are being built various other combinations, such as cafe-coach, coach-parlor, and sleeper-club cars, where one combined will take the place of two separate cars when traffic is light.

In the fast-freight line, baggage cars have been built fitted with tanks for the shipment of live fish.

Air Conditioning. In recent years there has been considerable experimenting with air-conditioning equipment for railway passenger cars, and there is a rising demand on the part of the traveling public for this travel comfort. Although the art is undoubtedly in a formative stage, tested equipment for this purpose is available. Close attention is being given, not only to improvement in quality of service and to better adaptation of types of accommodation to customer demand, but also to popularization of travel facilities.

Rail Cars

During the year oil-electric rail coaches were built in Germany to operate at 93 mph, which is probably the first effort of a railway company to meet the competition of air services, there being a regular service of the latter type between Berlin and Hamburg. The equipment is made up in two parts joined by gangway bellows. Each of the outer ends is supported on a four-wheel truck having a wheelbase 11 ft 6 in. long, and the inner ends are carried on a common truck with the same wheelbase. Since the designed speed is 93 mph, air resistance is of primary importance, and model tests were carried out in the wind tunnel of the Zeppelin works to determine the most suitable form for the body construction. The outer ends are streamlined, and in addition the overall height of the cars has been kept as low as possible, and any projecting fittings have been avoided, the doors and window frames being flush with the outer surfaces. To avoid eddy currents under the cars, the whole of the underside is incased in metal sheeting. The cars are driven by two 12-cylinder, 410-bhp, 1400-rpm engines directly coupled to a generator. The current generated is supplied to two electric motors geared to the driving axles. The seating capacity in the unit is 102 passengers and there is a compartment for baggage, a small section in the center of the cars being reserved as a buffet. The question of braking on a service operated at such high speed was recognized as being most important, since with constant brake resistance the stopping distance varies as the square of the velocity. Owing to the special design of the brakes employed, however, it is anticipated that it will be possible to bring the cars to rest from full speed in a distance of approximately 3600 ft. The cars are equipped with two braking systems, one oil and the other air operated, the car wheels being provided with special brake drums and the brakes being of the outside type, lined with an asbestos composition. In order to avoid long and heavy levers, a number of small brake cylinders were installed, designed to operate in the usual way. The total length of the combined articulated unit is 138 ft, with a light weight of 77 tons.

The introduction of the pneumatic rail car, with the practicability of its adaptation to railway service, is of special interest alike to the railways and the traveling public. This new unit of equipment involves a modified view of railway service and a radical departure from the heavy equipment which has been generally employed. The pneumatic rail car, as its name implies, is a light-weight unit, approaching automotive practice to a greater extent than conventional railway usage, with its light-steel construction, pneumatic tires, and internal-combustion engine. It was developed in Europe by engineers working in collaboration with railway men, who sought relief from the growing traffic loss attending many of the operations of the uneconomic steam units. The pneumatic tire, with the conventional type of rail flange, provides riding comfort not possible in any other type of rail equipment now in use, and the adhesion of this tire on the steel rail, which is three times greater than that obtaining for the conventional type of steel wheel, insures maximum acceleration and braking qualities. Weight, which is a limiting factor in the construction of this equipment, has been met by the utilization of stainless steel and a patented process of shot welding. The average dead weight, empty, of 337 lb per passenger capacity, makes possible the utilization of a relatively small internal-combustion engine, which is nevertheless sufficiently powerful and flexible even when acting through a straight mechanical drive to serve the purposes desired.

Car Trucks. With increasing train speeds, very serious attention has been paid and exhaustive tests have been run to determine the effect on passengers and lading from the periodicity of spring oscillation. Remarkable discoveries were made as regards the disturbance and damage to lading in the case of present standard-coil freight-truck designs of conventional pattern. It was quickly evident that coil springs can be em-

ployed to advantage to provide sensitive response to small changes in load, and elliptic springs for the purpose of dampening spring oscillations and amplitudes of oscillation, one desirable combination appearing to be that of coil and elliptic springs arranged in series. In such cases the elliptic spring is permitted the greater part of the admissible springing and the coil springs the smaller part. The small changes of load, which the elliptic spring transmits without oscillation, due to friction between the leaves, can then be easily taken up by the coil springs.

During the year test reports were issued regarding 100-car freight-train operation, with and without roller bearings, on a comparative basis, such equipment having been tested under the auspices of the Timken Roller Bearing Company. Ten 70-ton hopper cars, with bodies made from aluminum, were placed in service, with a saving in weight of 21,200 lb, as compared to the use of steel as ordinarily employed.

Tank cars handling dry granular commodities with practically the same convenience and efficiency that standard tank cars now handle liquids were perfected after nearly two years of research and road tests by the General American Tank Car Corporation and are designed for bulk movement of such dry materials as cement, lime, silica sands, sulphur, fertilizers, soda ash, pottery clays, and fuller's earth.

A car is being designed by the Standard Steel Car Company for the quick freezing of perishable foods. This car is not unlike the all-steel baggage car, inasmuch as it has a central door opening, with a partition beyond the center door which forms a small compartment for refrigerating machinery operated by electric motors. The large compartment is equipped with two 8-stage frosters, the combined capacity being equal to 40,000 tons of frosted foods every 24 hours.

This car is unique in the sense that it is planned to send it to various sections of the country during the growing season and have it collaborate with packing plants, so that the food is first prepared by the packing plants and is then turned over to this car, where it is packaged and quick-frozen. These packages are than put into cartons, sealed, and placed in refrigerator cars for shipment to warehouses and final distribution. In other words, it is a manufacturing plant on wheels, from which emanates considerable tonnage during a year's service.

The same company is also designing a new type of refrigerator car to handle frosted foods that must be transported at low temperatures. This design is unique in that there is practically one car body built within another, with heavy insulation on the sides, floor, and roofing, the inner shell being absolutely insulated from the outer shell, thus eliminating as far as possible the transmission of cold or heat from without and within.

Determined to adopt every possible means of coordinating railway and truck service and of meeting highway competition, a number of railways are offering an adaptation of the ferry-truck plan, which consists of the movement of entire motor trucks and trailers, loaded with merchandise, by flat cars.

Containers have also proved a most valuable acquisition to the railways in meeting highway competition.

Horsepower and Tractive Effort of Steam Locomotives (Locomotive Ratios)

By A. I. LIPETZ,[1] SCHENECTADY, N. Y.

Cole's Ratios, by the use of which the tractive-force curves of locomotives can be calculated and plotted, knowing certain of the locomotive dimensions, have been successfully used for the past 15 to 18 years. The Type E superheater, the feedwater heater, long-travel valve motions, and certain changes and refinements in proportion and design have tended to increase the power of steam locomotives built since 1920. The power calculated by Cole's method does not therefore accurately represent the actual power of modern steam locomotives. In his paper the author reviews the work of Cole and others, checking it against available test-plant and road-test data, and finds that some of Cole's constants still hold good in a great many cases. Others he finds it necessary to modify not only as to the figures, but also as to the method itself.

He develops constants for a new, easily applied method, based on boiler evaporation and a factor depending on the number of revolutions of the driving wheels, for use in connection with modern locomotives, and suggests the continued use of Cole's Ratios for locomotives in the design of which the modern factors are not included.



BY the term "locomotive ratios" are commonly understood formulas and tables which permit the calculation of tractive efforts of steam locomotives from their dimensions. F. J. Cole, late chief consulting engineer of the American Locomotive Company, devised a method for plotting tractive-effort curves of locomotives,[2] which has been used for the last 15 to 18 years with great success. His figures were based on locomotive test data available in his time, and even now they still represent fairly accurately the power and tractive effort of locomotives of the designs known up to the end of the World War.

Since 1920 a number of improvements has been made tending to increase the power of the steam locomotive, the most outstanding of which are type "E" superheaters, feedwater heaters, and long-travel valve motions. Changes and refinements introduced into the designing and proportioning of locomotive parts have also increased the power of the locomotive. Consequently, when Cole's method is applied to modern locomotives, built since 1920, there is a discrepancy between the calculated and the actual power of the locomotive. It was therefore thought necessary to revise Mr. Cole's figures and recommend new ones instead which would apply to modern locomotives.

The author was fortunate enough to have access to Mr. Cole's basic working sheets. He could therefore go over all the information on which Cole's method and figures were based, and critically review them. He was struck by the correctness of some of his constants, which still hold good in a great many cases, but in others he found it necessary to modify not only the figures, but the method itself. Especially have Cole's evaporation figures been found to be very accurate, and applicable to modern locomotives. However, slight corrections to care for changes in design are sometimes required. As regards the underlying Cole idea of figuring the tractive effort on the basis of cylinder sizes and factors depending upon piston speed, it was thought better to base the calculation on boiler evaporation and a factor depending upon the number of revolutions of the wheels.

It should be borne in mind, however, that no one method or formula can express the tractive effort of all locomotives. So many variables affect the performance of a steam locomotive that attempts to devise a universal formula have never been successful. As an example, it suffices to say that in designing steam cylinders the proper dimensioning and shaping of steam ports and bridges, and the relative location of exhaust- and live-steam passages, are of great consequence as regards the question of losses caused by wire drawing and condensation. No formula can express the ability of the designer to provide an economical cylinder. The most that can be done by a formula is to express the average results of a well-proportioned locomotive of a design that is characteristic of a certain period of locomotive development, and to give an idea of the power of a steam locomotive under ordinary working conditions with a degree of accuracy sufficient for practical purposes.

With this end in view, the author attempts in what follows to provide means of quickly and fairly accurately evaluating the horsepower and tractive effort of a modern two-cylinder, simple-expansion locomotive equipped with a superheater of sufficient size, with or without a feedwater heater, with a proper valve motion, and with other parts characteristic of modern locomotive design.

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[1] Consulting Engineer, American Locomotive Company. Nonresident professor, Purdue University. Mem. A.S.M.E. Mr. Lipetz was educated at the Warsaw (Poland) Polytechnic Institute, from which he received the degree of Engineer Technologist (mechanical engineer) of the first grade in 1902. In 1903 he entered railway service in Russia as an apprentice on the Moscow-Kiev-Voronesh Railway, later serving as fireman, locomotive driver, inspector, and assistant master mechanic. From 1906 to 1909 he was assistant professor of thermodynamics and railway mechanical engineering at the Kiev Polytechnic Institute, passing examinations preliminary to degree of Doctor of Engineering. For three years he held administrative positions on the Tashkent Railway, and for the three years following was chief of the locomotive department, Ministry of Railways, Russia. From 1915 to 1920 he served the Russian Railways in the United States, first as representative of the Russian Ministry of Railways and then as assistant chief and, later, chief of the Russian Mission of Ways of Communications in the United States. Since 1920 he has been connected with the American Locomotive Company, first as European representative at Paris, and since 1925 as consulting engineer at Schenectady, N. Y. Since 1927 he has also been nonresident professor of locomotive engineering at Purdue University. He was granted a number of early patents on Diesel locomotives and was the designer of the Russian Decapod locomotives of the war period. He is the author of many papers on steam and Diesel locomotives, and was the reporter for America on locomotives of new types at the International Railway Congress held at Madrid, Spain, in 1930.

[2] "Locomotive Ratios," by F. J. Cole, American Locomotive Co. Bulletin No. 1017, 1914.

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For locomotives of an earlier period, Cole's formulas are sufficiently accurate, and although the method about to be explained could also be extended to cover all locomotives, no attempt has been made to supersede the Cole method for locomotives of prewar design.

Tractive Effort

The basis of all tonnage and speed-time-distance calculations is the tractive effort of a steam locomotive. Of the different tractive efforts that are sometimes being used, the basic one is the "indicated tractive effort," which is the direct result of the work of steam in the cylinders. For a two-cylinder, simple-expansion locomotive it is usually expressed by the formula

$$T_i = p_i \frac{d^2 s}{D} = \alpha p_b \frac{d^2 s}{D} \quad \ldots\ldots\ldots\ldots\ldots [1]$$

in which

T_i = indicated tractive effort, lb
p_i = mean indicated pressure in the cylinder, lb per sq in.
d = diameter of cylinder, in.
s = piston stroke, in.
D = diameter of driving wheels, in.
p_b = boiler pressure, lb per sq in.
$\alpha = p_i/p_b$ = mean indicated pressure ratio.

The "rail tractive effort," which is the imaginary force applied to the rim of the wheels, and the drawbar pull on the tender drawbar, can be found from the indicated tractive effort, if the engine friction and total locomotive resistance are known. The object of this paper is to establish a method and figures for the calculation of indicated tractive effort as a function of speed.

As can be seen from [1], the tractive effort depends upon the boiler pressure, size of cylinders, and diameter of the driving wheels. At starting and low speeds, when the capacity of the boiler is able to supply the cylinders with steam at maximum cut-off, the enumerated factors are the only ones that need be considered. In this case the mean effective pressure is from 85 to 93 per cent of the boiler pressure, and the tractive effort is easily calculated from Formula [1]. By making $\alpha = 0.85$, the formula gives what is known as the "rated tractive effort" of a steam locomotive, namely,

$$T = 0.85 p_b \frac{d^2 s}{D} \quad \ldots\ldots\ldots\ldots\ldots [2]$$

Ordinarily it is assumed that this tractive effort does not depend upon the speed, which is more or less correct for low speeds. However, as the speed increases, the tractive effort drops for the reason that losses due to steam friction in the steam ports and passages (wire drawing) decrease the mean effective pressure. The tractive-effort curve at the maximum cut-off (85 per cent), or as it is sometimes called, the "cylinder tractive effort," varies with the speed, as shown by the curve $RAA'M$, Fig. 1.[3]

If the maximum cut-off should be maintained for some time, a speed (V_1) will be reached beyond which the boiler evaporation is not sufficient at that cut-off for sustained performance. The boiler pressure will begin to drop. Just before this speed is reached, the cut-off should be reduced, say, to 81 per cent, when the cylinder tractive effort will be represented by a new curve, BB', which will also have a sloping characteristic similar to that of the maximum cylinder tractive effort. At a certain speed (V_2) the boiler evaporation limit will again be reached and the cut-off will have to be further reduced, and so on. Accordingly, there will be a whole series of curves which will represent tractive efforts at different cut-offs. Points A, B, C, D, E, etc. if connected, will form a border line for all possible tractive efforts below the evaporation limit. In other words, it will represent the maximum tractive effort of the locomotive, limited by the boiler evaporation, or the "boiler tractive effort."

Curve RA and curve $ABCDE$ together form what is called the tractive-effort curve of the locomotive. One portion, RA, is part of the maximum cylinder tractive effort, while the remainder, $ABCDE$, is the boiler tractive effort. Two locomotives with the same cylinder sizes, boiler pressures, and wheels, but with different boilers, would have cylinder tractive efforts that were practically alike, but the boiler tractive efforts would differ, as shown by curve $A'B'C'D'E'$. It is therefore evident that a knowledge of cylinder sizes and other parts of the machinery is not sufficient for establishing the tractive effort of a locomotive.

As the horizontal portion of the maximum cylinder tractive effort is the rated tractive effort, which does not vary with speed, it can be stated that the tractive-effort curve of a locomotive is limited on the one hand by the rated tractive effort, which is a horizontal line, and by the boiler tractive effort, which is a curve of hyperbolic shape.

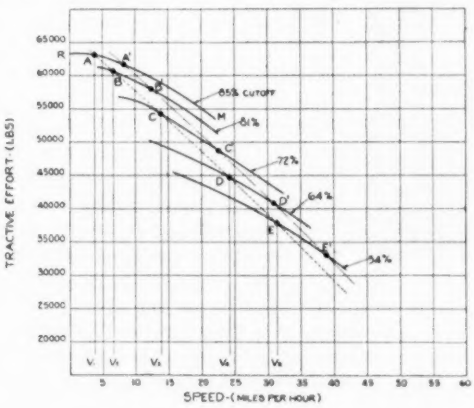


Fig. 1

A method of plotting the tractive-effort curve on the basis of cylinder sizes only, using the rated tractive effort and certain factors, irrespective of the size of boiler, is in principle incorrect. Nevertheless, in practice, it can give good results for the reason that in a locomotive of a certain period and of conventional proportions, the sizes of the locomotive boiler and cylinders are in a certain ratio to each other. However, as with improvement in locomotive design the boiler becomes larger in proportion to the engine, in order to meet the increased speed requirements of traffic, the formulas based on cylinder sizes and Cole's factors begin to give too low values.

Mr. Cole himself introduced the idea of boiler horsepower. He determined it to be equal to the maximum evaporation in pounds per hour divided by 20.8 for superheated steam, this figure representing the consumption of steam in pounds per horsepower-hour, including auxiliaries.[4] This made it possible to determine what is called the boiler percentage, which means the ratio of maximum cylinder horsepower (at 1000 fpm piston speed) to the boiler horsepower, the maximum cylinder horsepower being found from the Cole formula, and the boiler horsepower as stated above. Cole recommended designing locomotives with boiler percentages as close to 100 per cent as possible. In this case his method of plotting the tractive-effort curve would be correct, strictly speaking, at least for one point, namely, 1000 fpm piston speed. However, for locomotives with larger or smaller boilers, the boiler

[3] Compare with Lomonossoff, "Locomotivversuche in Russland," Berlin, 1926, p. 84; also "On the Calculation of Drawbar Pull of Steam Locomotives," by R. Nakamura, part iii, Tokyo, 1931, p. 4.

[4] "Locomotive Ratios," p. 9; also "Locomotive Handbook," American Locomotive Co., 1917, p. 64–65.

percentage had no effect on plotting the curves. Therefore some locomotive builders and railroads have already found it necessary to make corrections in the Cole factors in relation to the boiler percentages. This in fact is a roundabout way of figuring the correct tractive effort for a piston speed of 1000 fpm, and an approximate value for other speeds.

Direct Methods of Establishing Tractive-Effort Values

There are at least two possible direct methods for establishing the tractive-effort values for different speeds—one, which might be called the analytical method, and the other, an empirical method.

The first, the analytical method, is logical and seemingly very simple; however, in order to give good results, it requires a number of corrections, which can be determined only from actual experience. It would seem very simple, if the evaporation of the boiler and principal dimensions of the locomotive engine are known, to calculate the amount of steam per stroke for each speed and the tractive effort for that speed resulting from the work of the figured amount of steam. Knowing the diameter of the driving wheels D in inches, we can find the number of revolutions per minute n from the formula

$$n = \frac{336.134\ V}{D} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

V being the locomotive speed in miles per hour.

Denoting the clearance volume of a cylinder in relation to the swept volume of the piston by l; the cut-off, or admission, in relation to the stroke by a; the compression in relation to the stroke by c; the swept volume of a piston, in cubic feet, by U; the specific volume of steam, in cubic feet per pound, at the moment of cut-off by v; the specific volume of steam, in cubic feet per pound, at the beginning of compression by w; the total evaporation of a locomotive boiler in pounds per hour by E; and the amount of steam spent on auxiliaries in relation to the total evaporation by x, the enumerated quantities for a two-cylinder simple locomotive are tied together by the following formula

$$E(1-x) = U \times 4n \times 60\left[\frac{a+l}{v} - \frac{c+l}{w}\right] \ldots\ldots [4]$$

from which the admission a can be found by the method of approximation.[5]

If we know the boiler pressure, the admissions thus found for certain numbers of revolutions corresponding to certain speeds, permit the plotting of indicator cards for these speeds and thus establishing the mean effective pressures p_i of steam in a locomotive cylinder, also for various speeds, which in turn will make it possible, by using Formula [1], to figure the tractive effort of the locomotive as a function of its speed.

This method was proposed and used in the early years of locomotive development, and for a long time it was considered to be the only possible method in view of its comprehensibility. It is rational in principle, and therefore had a great many adherents. The difficulty with the method, however, lies in the fact that two important and very complex factors are neglected: first, the cooling of steam in the cylinder, and second, the distortion of indicator cards due to speed, both of which cause considerable losses.

The cooling of steam in the cylinder, or what is known as the heat exchange between the steam and the cylinder walls (and heads), cannot be expressed by any theoretical formula. A large number of studies has been made, some of them employing very complex forms of mathematical analysis,[6] in order to establish the cooling effect of the cylinder, but none of them has given practically satisfactory results, as too many variables are involved. With highly superheated steam the conditions are slightly better and some formulas can be established, but even then, resort must be had to empirical figures from tests, as will be shown later.

The second factor, distortion of indicator cards, depends upon the valve-motion characteristics, steam-port dimensions, and steam passages, as all of them entail drops in pressure, depending upon the speed. These laws are also very intricate and do not easily lend themselves to mathematical expression; and when formulas are used, they must also be corrected by test data, which of course are only approximate.

The best example of such a theoretical study in recent years is that made by the late Professor Strahl of the German State Railway in his book "Der Einfluss der Steuerung auf Leistung, Dampf- und Kohlenberbrauch der Heissdampflokomotiven."[7] He bases his calculations on superheated steam at 300 C (572 F), and assumes for such steam a contraction of volume equivalent to a cooling of 60–80 C (108–144 F). For higher superheats he further assumes that each 5 deg C (9 deg F) increase in superheat decreases the steam consumption by 1 per cent. These constants are recommended as approximate general results from tests.

For the mean effective pressure p_i he gives the formula

$$p_i = bp_s - p_d\,(b - 0.6a) - p_e$$

where p_s is the steam-chest pressure, a is the admission, and b, p_d, and p_e are expressed by long and very intricate formulas embodying a number of experimental constants.[8]

Professor Strahl shows that for the German standard locomotives used in 1917, his method gives fairly good results. The author tried to apply it to an American 4-6-4 locomotive for which test data were available. It took an experienced calculator two days to determine three points—three tractive efforts for three speeds—and these were not in agreement with the test results.

It is possible, however, that if tables were worked out for the constants and common principal dimensions of American locomotives, the calculations could be simplified, and that certain modifications would make the formulas more suitable for American locomotive proportions.

The main difficulty, however, is that a number of assumptions has to be made for the derivation of these formulas, and in order to make them conform to actual practice, the constants must be taken from test data, which, of course, are good only for locomotives of approximately the same principal dimensions; in other words, these complicated formulas do not eliminate the need of using empirical data, and therefore the question arises whether it would not be better to base a method on empirical data only, taking into consideration the principal dimensions of the locomotive, and thus save the amount of work involved in these intricate calculations.[9]

The second method, which is an outcome of just this consideration, was suggested also years ago by locomotive investigators. Frank, von Borries, Richter, and others tried to establish empirical tables and curves for the horsepower of locomotives of different types as a function of the locomotive speed expressed in number of revolutions per minute. Table 1 gives the figures

[5] This is necessary in view of the fact that c is a function of a.

[6] "Steam Power," by W. E. Dalby, New York and London, 1915, p. 243.

[7] Published by the Hanomag-Nachrichten Verlag G.m.b.H., Hannover-Linden, 1924.

[8] See pp. 9, 11, 12, 25, and 27 of Professor Strahl's work just cited.

[9] In his doctor's thesis, Achterberg suggested calculating the losses due to heat exchange in cylinders instead of using Strahl's empirical constants, but had to introduce other constants based on test results. See "Kurzes Lehrbuch des Dampflokomotivbaues," Dr.-Ing. F. Meineke, Berlin, 1921, pp. 66, 68–69.

TABLE 1

	Revolutions per minute					
	90	120	150	180	210	240
Horsepower per square meter of gas heating surface (inside)	4.7	5.2	5.5	5.7	5.7	5.5
Horsepower per square foot of evaporative heating surface (outside)	0.39	0.43	0.46	0.48	0.48	0.47

which von Borries recommended for plotting horsepower curves for two-cylinder simple-expansion locomotives operating with saturated steam.[10] They are referred to one square meter of gas fire-side (inside) heating surface and given in relation to the rpm of the driving wheels.

The last line of the table represents the same horsepowers referred to outside heating surface in square feet, in order to make them comparable with data on American locomotives. They are from 25 to 35 per cent lower than corresponding figures of our modern locomotives with superheated steam, but the general character of power increase with speed is the same.

As soon as the law of the horsepower curve was established, the tractive-effort curve could be easily figured by the known formula

$$P = \frac{T \times V}{375} \quad \ldots\ldots [5]$$

where P = indicated, rail, or drawbar horsepower
T = corresponding (indicated, rail, or drawbar) tractive effort, lb, and
V = locomotive speed in miles per hour.

Similarly, in 1901, the author while a college student, found that the parabolic shape of the horsepower curves suited very well the then existing types of locomotives. He published an article[11] in which he suggested the use of formula

$$\frac{P_i}{H} = \frac{A\sqrt{n}}{1 - B\sqrt{n}}$$

in which A and B are functions of the principal dimensions of the locomotive, mainly heating surface and grate area.

These formulas are also out of date, but it might be possible to establish new constants even for the most modern locomotives, as all horsepower curves seem to follow the same general law.

The advantage of the empirical method, expressed either by tables or formulas, lies in the idea of basing the calculations on horsepower rather than on tractive effort, as the former depends upon a smaller number of variables than the latter. If the locomotive is built in accordance with certain standards of perfection and refinement, the horsepower depends mainly upon the amount of steam generated by the boiler. If, in accordance with previous notations, the amount of steam available for the engine is

$$E \times (1 - x)$$

and if the properly designed locomotive engine requires S_h lb of steam per ihp-hr, the indicated power of the locomotive is evidently

$$P_i = \frac{E \times (1 - x)}{S_h} \quad \ldots\ldots [6]$$

The sizes of cylinders do not come directly into consideration. If they vary within a limited range, as is the case in locomotives, and if the proportions are correct, they should not influence S_h. As is known from the theory of steam engines, the important factors are the cut-off and the number of revolutions per minute (rpm).[12] The first factor determines the expansion ratio and the thermodynamic efficiency of the steam cycle, while the second affects the condensation and steam-friction losses.

Some investigators think that steam consumption depends on piston speed, and so Mr. Cole found in his investigation of locomotive data, but the majority of steam-engine authorities refer steam losses to the rotary speed of the crank and not to piston speed. Steam-consumption figures based on recent locomotive test data also show more uniformity when they are referred to crank speed rather than to piston speed.

However the case may be, the advantages of referring the steam consumption to piston speed, or to the rpm, cannot be great, as the piston strokes do not differ much in modern locomotives, being between 28 in. and 32 in. On the other hand, as the cut-off depends on the rpm and not on the piston speed (see [4]), it seems to be more practicable to consider S_h as depending on the rpm.

It cannot depend on linear speed of the locomotive as this would be equivalent in stationary practice to the peripheral speed of the flywheel. Nothing like that has ever been found, and therefore the diameter of the driving wheels can have no influence on the indicated power of a locomotive. It is reflected, though, in the tractive-effort values.

The tractive effort of a locomotive is one of the two components of the locomotive horsepower. The cylinders and driving wheels permit resolving the horsepower into tractive effort and speed in accordance with Formula [5], namely, $P = T \times V/375$.

Thus the tractive-effort calculations can be simplified by determining first the locomotive horsepower as a function of either the rpm or the piston speed, depending upon which of the two influences on the efficiency of the locomotive is reflected in S_h, and then later resolving the horsepower into tractive effort and locomotive speed.

The foregoing, of course, refers to the boiler tractive-effort curve, which has a hyperbolic shape (Fig. 1). As to the horizontal portion of the tractive-effort curve, this is determined entirely by the rated tractive effort (Formula [2]). There is no connection between these two except only as far as the sizes of the cylinders and boiler in a well-proportioned locomotive are dependent upon each other.

Boiler Tractive Effort

Thus the tractive-effort curve depends in its major part upon the amount of steam generated in the boiler and the consumption of steam per horsepower-hour. The natural thing, therefore, is to establish the laws of the two variables. The analytical method can hardly lead us to the result sought for, and we have to rely entirely upon test figures.

Unfortunately, tests are not being conducted in such a way as to enable us to establish the two above-mentioned variables as completely as is desired. Even the most elaborate tests, which are supposed to give all data in which an investigator should be interested, attempt to find solutions for various problems not always with a view to establishing the tractive-effort curve. However, by analyzing available data, certain conclusions can be drawn, as will be seen below.

Strictly speaking, horsepower and tractive effort are not definite conceptions unless we specify certain conditions at which power and tractive effort are being considered. There can be the *maximum* tractive effort which the engine is capable of producing irrespective of the amount of coal burned and the cost of such a forced operation. There can be the *most economical* tractive effort, and this from the point of view of fuel only, when short cut-offs and long expansions are used, or from the point of view of

[10] "Das Eisenbahnmaschinenwesen der Gegenwart," by Blum, von Borries, and Barkhausen, I—Lokomotiven, Wiesbaden, 1912, p. 123.

[11] Bulletins of the Warsaw Polytechnic Institute, 1901–1902.

[12] "The Steam Engine," by Sir J. Alfred Ewing, Cambridge, 1926, p. 208; also "Steam Power," by W. E. Dalby, p. 237.

total operation expenses. The latter is the more important, and it is safe to assume that under ordinary conditions the mode of operation of locomotives, established by long experience, is in the end the most economical. This can be called "performance tractive effort." Only in exceptional cases, when the locomotive is pulling a fast passenger train at top speeds and is forced to the limit of its power, is the maximum tractive effort, or what is called the "capacity tractive effort," developed.

The method of plotting the maximum and average performance curves is clear from Fig. 16. Both horsepower and tractive-effort test figures have been used and mutually checked by Formula [5].

Some European investigators, in order to introduce more definiteness into the terms, consider horsepower and tractive effort at a certain evaporation rate either normal or maximum. Professor Nordmann, who is in charge of the Locomotive Test Department of the Reichsbahn, has established a standard evaporation of a locomotive boiler equal to 57 kg of steam per square meter of inside (fire side) heating surface. This is equivalent to 11.7 lb per sq ft of inside (fire side) heating surface, or approximately 10.6 lb per sq ft of outside (evaporative) heating surface, as it is customary to figure in this country. For the purposes of the tests conducted by Professor Nordmann, such a method is perfectly justified, because the main object of his tests is to compare the power and efficiency of German standard-type locomotives. This standard evaporation figure makes it possible to reduce, for purposes of comparison, all locomotives to one basis.

However, such a method, which is also followed by other European railroads, only with different standard evaporation figures, would not enable us to plot the locomotive tractive effort as we understand it, because a locomotive working under actual operating conditions is not regulated in such a way as to generate a certain constant amount of steam per square foot of heating surface under varying conditions of work.

The amount of steam which is generated by a locomotive depends, primarily, upon the amount of coal burned in the firebox, and, secondly, upon the efficiency of the boiler under that condition. There is a limit to both the amount of coal that can be burned on a certain area of grate and to the amount of heat that can be transferred through the heating surface of a boiler into the water. In locomotives with small grate areas the first will be the ruling factor. In locomotives with excessive grate areas, the second may be the controlling factor. In well-designed locomotives the proportions are such that the two factors will more or less balance.

On the other hand, the amount of coal burned on the grate of a locomotive depends upon the draft in the smokebox, which in turn depends upon the amount of steam put through the exhaust nozzle.

Thus the evaporation depends upon the rate of firing, which in turn depends upon evaporation. This establishes a balanced condition in a locomotive boiler at certain rates of firing, permitting a lesser or greater forcing of the boiler. A locomotive can be operated very hard, at the limit of its capacity, burning as much as 200 lb or more of coal per square foot of grate area, but the efficiency of the boiler will then be very low and the total amount of steam may not be much more than at a lower firing rate. Experienced firemen and engineers know what the normal working conditions of a locomotive should be, and ordinarily they fire so as to develop a certain, reasonably economical, amount of steam. Road tests give a fairly good indication of these conditions. When an engineer takes over the road a full-load train, it may be expected that he operates the locomotive so as to develop a reasonable horsepower and tractive effort under the existing operating conditions. Certain horsepowers and tractive efforts are then developed which were referred to above as performance curves. As it will be shown later, they correspond to what we usually consider normal evaporation of the locomotive boiler.

However, a locomotive may be able to develop more than its performance curves would indicate. The limit of its capacity is normally higher. Therefore, on some roads, in addition to performance tests, special capacity tests are being made in order to establish the maximum output of the locomotive. These tests should be so conducted that both the maximum boiler pressure and the water level remain constant during the test. The fulfilment of these two conditions is absolutely necessary in order to obtain reliable figures for sustained capacity operation; when they are not adhered to, exaggerated results will follow.

The underlying idea of the European method of testing locomotives at constant evaporation of the boiler is that the evaporation does not depend upon the performance of the locomotive engine. The advocates of this theory claim that a locomotive boiler can generate any desirable amount of steam, up to its limit, independently of the speed of the locomotive. They assume that because more steam is generated and passed through the exhaust nozzle, the draft is accordingly increased, permitting in turn the generation of the increased amount of steam. It is also claimed that this amount is independent of the manner in which the steam is exhausted, whether intermittently or continuously; in other words, by a small number of heavy exhausts, or a large number of small puffs, creating in the latter case a practically uniform draft. Professor Goss was a believer in this theory,[13] and there are other authorities (for instance, L. H. Fry[14]) who share this view.

On the other hand, an opposite view is held by a large number of engineers. Von Borries, for instance, was of the opinion that uniformity of draft results in better boiler efficiency and higher evaporation rates, which, he thought, was proved by tests with cross-compound locomotives having only half the number of exhausts.[15] Further, in the author's opinion, the vibration of the locomotive on a railway track, which increases with the speed, must influence steam generation very favorably. The amount of air which is forced into the ashpan in larger quantities as the speed goes up should also affect the evaporation. In other words, as a result of all these causes, the boiler evaporation increases with the speed.

There are no direct tests which would indicate the correctness of either of these theories, because, to the best of the author's knowledge, no tests have ever been conducted with this purpose in view. Ordinary road tests do not give the information, because the speed during the test varies all the time. The amount of water which is measured at the beginning and at the end of the test cannot show the relation between total evaporation and variable speed.

It would seem that stationary tests should give the figures of maximum evaporation and speed, because each test is conducted at constant cut-off and constant speed, and the most severe combination of these two variables should be close to the limit of boiler capacity. With this in view, curves were plotted for five Pennsylvania locomotives (K4s, E6s, K2sa, L1s, and I1s), as shown in Fig. 2, for different numbers of revolutions of the driving wheels per minute. The data were taken from bulletins published by the Pennsylvania Railroad on locomotive tests at the Altoona testing plant. For better comparison, evaporation is given not in absolute figures but in percentages of evaporation

[13] "Locomotive Performance," by W. F. M. Goss, New York, 1907, p. 138.

[14] Trans. A.S.M.E., vol. 47 (1925), p. 1271.

[15] "Compound Lokomotiven," by von Borries, *Organ für die Fortschritte des Eisenbahnwesens*, 1883, pp. 190–192; also "Das Eisenbahnmaschinenwesen der Gegenwart," by Blum, von Borries, and Barkhausen, I—Lokomotiven, Wiesbaden, 1912, pp. 118–119.

for respective boilers in accordance with Cole constants (A. L. Co. Hand-Book, 1917, p. 59).

It will be seen that while the curves differ very widely, their shapes all have the same parabolic characteristic of increase with speed up to a certain limit, and subsequent drop for higher speeds. The peak points are from 12 to 28 per cent above the Cole figure. This must have been due to forcing the boilers to high evaporation rates. The peak point on curve 1 (locomotive K4s), for instance, corresponds to a rate of firing of 171 lb of coal per sq ft of grate area per hr. On curve 5 (locomotive I1s) the rate of firing is still higher, due to the small grate area of this locomotive. In Fig. 2 the rates of firing for peak points are inscribed in rectangles. The figures in circles represent the ratios of heating

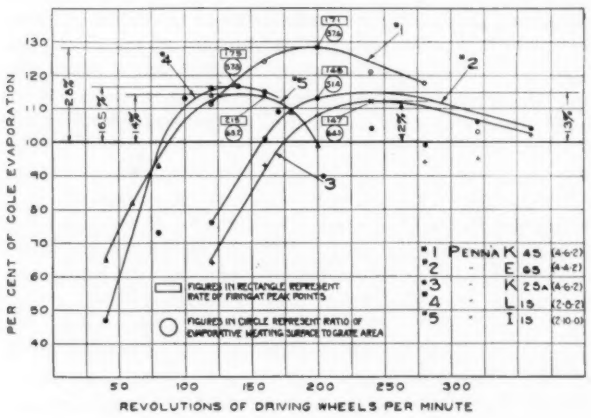


FIG. 2

surface to grate area for each of the five locomotives. In other cases they have also been very high, indicating that Cole's evaporation corresponds very closely to a rate of firing of 120 lb per sq ft of grate area per hr, as was assumed by him.

The shape of the curves evidently depends upon the ratios between the principal dimensions of the boilers. It can be assumed that the ratio between total heating surface and firebox heating surface, as well as the ratio between total heating surface and grate area, influenced the shape of the curves and the location of the peak points. The drafting arrangements and the size and type of the exhaust nozzle must have also been of great importance in this respect.

It is not possible to find a formula for the curves of evaporation as functions of these variables. They do not run consistently enough, and there are other variables that cannot be put in a formula, as, for instance, the quality of coal. More consistent results were obtained from plotting curves of road tests, as shown in Fig. 3. Three locomotives are considered here (New York Central J-1a and L-2, and Boston & Albany A-1), representing three different classes of service: high-speed passenger, high-speed freight, and moderate-speed freight.

As stated above, there are no test data which would permit plotting these curves directly. An indirect method was therefore evolved for this purpose. From road tests with locomotives the maximum-performance indicated horsepowers and corresponding speeds and cut-offs were chosen. From the Altoona tests, locomotives of similar engine size were selected, and the steam consumption per horsepower was found. It was corrected for difference in boiler pressures and superheats, and further, a certain percentage for auxiliaries was added from road-test data. Thus the probable specific consumption of road locomotives per indicated horsepower at various speeds was established. The product of horsepower and the established specific steam consumption furnished a means of finding the probable steam evaporation of the road locomotive for the maximum-performance power.

It will be seen that in this case the variation of the maximum evaporation of different locomotives is much less; it is between 2.7 and 6.7 per cent greater than the Cole figure. The variation in the location of the peak points in relation to speed is also less, and they all happen to be between 150 and 170 rpm. Nevertheless, even in this case it was not possible to tie up the shape of the curves with the principal dimensions[16] of the corresponding locomotives by a mathematical formula.

Other locomotives were also studied, and on the basis of the accumulated information it was found possible to draw a probable curve of evaporation, as shown in Fig. 4. It has a peak point 7 per cent above that of the Cole figure, and at about 200 rpm.

All locomotives shown in Fig. 3 were equipped with feedwater heaters, and it is very striking to find that the average maximum evaporation is about 7 per cent above the Cole figure. This is easily explained by the increase in efficiency due to feedwater heating. In other words, we may assume that the Cole evaporation figures still hold good for boilers on modern locomotives without feedwater heaters, but when locomotives are equipped

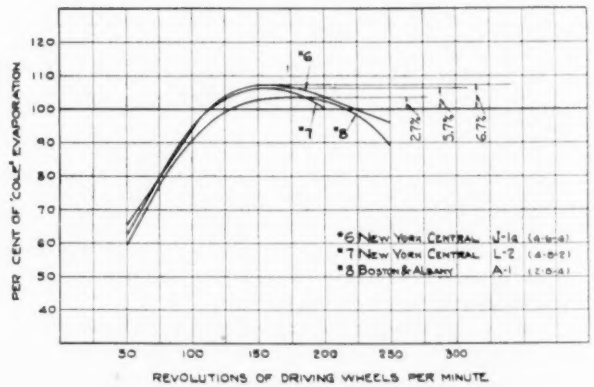


FIG. 3

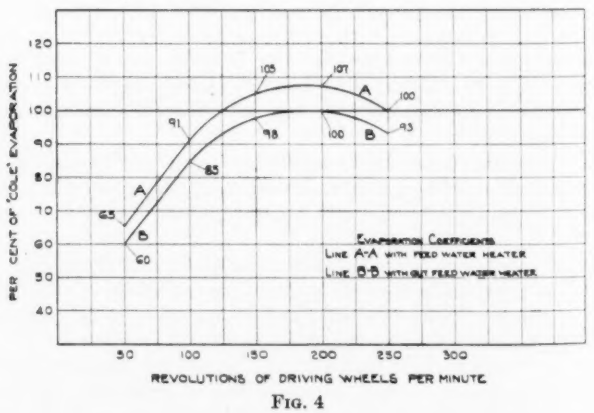


FIG. 4

with feedwater heaters, the boilers generate 7 per cent more steam.

It will be seen that the curve was plotted rather conservatively; at low speeds it follows the lowest portions of curves from road tests, while at high speeds it is slightly above the average for road tests—this for the reason that other locomotive data with feed-

[16] The principal dimensions of all locomotives represented in Figs. 2 and 3 are given in Appendix No. 1.

water heaters pointed in this direction. The plotting of this curve can be done by using the following formula:

$$E = \beta E_c \quad \ldots\ldots [7]$$

where E = boiler evaporation at a certain speed
E_c = Cole evaporation figure
β = evaporation coefficient in relation to the Cole evaporation figure.

The evaporation coefficients are shown in Fig. 4 and repeated in Table 2 in relation to the crank speed.

TABLE 2

Speed, rev per min	Evaporation coefficient — Locomotives without feed-water heater	Evaporation coefficient — Locomotives with feed-water heater
50	0.60	0.65
100	0.85	0.91
150	0.98	1.05
200	1.00	1.07
250	0.93	1.00

It may appear that the average curve was plotted rather arbitrarily, and that another curve could be chosen for the average evaporation with equal justification. It will be seen later that the ultimate recommendations for plotting horsepower and tractive-effort curves are based on test results and do not depend directly on the evaporation coefficients. They are independent of the evaporation curve plotted in Fig. 4, which is given here simply as an illustration of the principle of the method.

Recommended Method

When the boiler evaporation is determined, the indicated boiler horsepower can be found from the previously given Formula [6]:

$$P_i = \frac{E(1 - x)}{S_A} \quad \ldots\ldots [6]$$

where E = evaporation, lb per hr
x = ratio of steam used for auxiliaries
S_A = steam consumption per horsepower-hour, lb (steam rate)

provided S_A is known.

It was thought more advisable to study $S_E/(1 - x)$, which represents S_a, the steam consumption per horsepower-hour, including auxiliaries, rather than S_A and x separately. Various locomotives were analyzed and it was found that the average steam consumption per horsepower-hour, including auxiliaries, can be well represented by one of the two curves shown in Figs. 5 and 6, depending upon whether we refer them to the rpm or to the piston speeds.

It can be seen that the two curves differ very little; in other words, the piston speed has actually very little effect on steam consumption within the limits of strokes in modern locomotives, namely, 28 to 32 in. It is therefore practically of little importance for the computation of boiler horsepower whether the steam rate is taken as a function of crank speed or piston speed, at least for strokes between 28 in. and 32 in.

The method of figuring boiler horsepower can thus be reduced to the following:

On the basis of general boiler dimensions the Cole evaporation E_c is figured[17] and the various evaporations E for different crank speeds n (rpm) are calculated in accordance with Formula [7] and Table 2. The figures thus obtained for E are divided over the corresponding steam rates S_a of Fig. 5 and the quotients obtained are horsepowers P_i for the various speeds.

If it is desired to use Fig. 6, which gives steam rates with respect to piston speeds, then it is necessary that the evaporation figures estimated in accordance with Formula [7] and Table 2 be referred to piston speeds, which can be calculated for various crank speeds n by the formula

$$S_p = \frac{s \times n}{6} \quad \ldots\ldots [8]$$

where S_p = piston speed, ft per min
s = stroke, in.

In either case, whether the horsepower has been figured in relation to crank speed or piston speed, in order to find the tractive effort it is necessary to refer horsepower to locomotive speed, which can be done by using the formula

$$V = \frac{D \times n}{336.134} \quad \ldots\ldots [9]$$

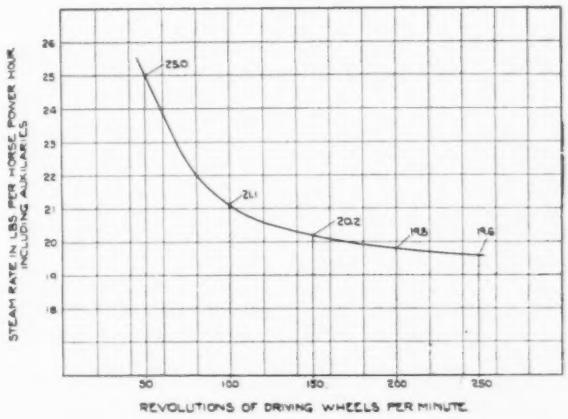


Fig. 5

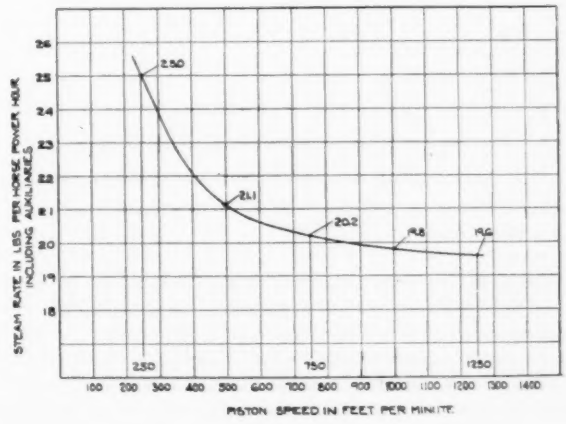


Fig. 6

The tractive effort is easily calculated by using Formula [5]. Tables for the conversion of crank speed into piston speed and locomotive speed for different strokes and diameters of wheels are very helpful in making these calculations.[18]

Thus the boiler tractive effort is determined and can be plotted on a chart. The rated tractive effort is then figured in accordance with Formula [1], plotted as a straight constant-effort line on the same chart, and extended until it intersects the boiler tractive effort. The point of intersection determines the speed at maximum tractive effort.

Regarding steam-rate curves (Figs. 5 and 6), the following is to be said:

The test data were not sufficient for plotting through the whole

[17] See Appendix No. 2.

[18] See Appendix No. 3.

range of speed. Only a portion of the curves, representing the most frequent average speeds of road tests, could be plotted. In order to cover the whole range of speed, an inverted method has been used; assuming that the coefficients of evaporation (β) as above given were correct, the total evaporations of several existing locomotives were calculated as functions of speed and divided over the corresponding horsepowers. The results for various curves are shown on Figs. 7 and 8 in reference to rpm and piston speed. Figs. 5 and 6 represent averages of the curves shown on Figs. 7 and 8.

It will be seen that the steam rates S_a give very consistent results at low speeds up to about 150–180 rpm, while above 200

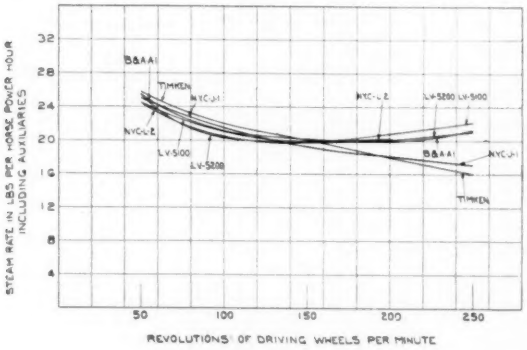


FIG. 7

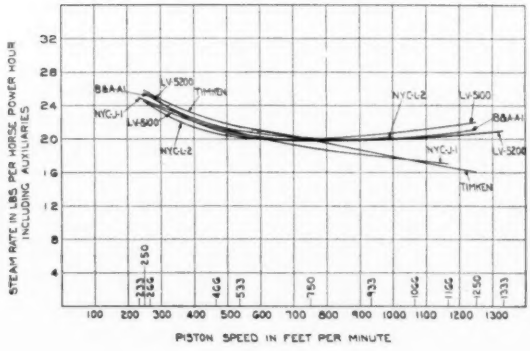


FIG. 8

rpm the steam rates vary rather widely. This cannot be due to the coefficient of evaporation β, because this is practically constant between 150 and 250 rpm. It is due to the usual discrepancy between locomotive-test results at high speeds, because the influence of such factors as quality of coal, size of nozzle, drafting arrangement, steam passages in the cylinders, and valve-motion design become more pronounced at higher speeds.

The average curves check fairly well with steam-consumption figures obtained from tests for the ranges of speed for which data are available. By correcting them back through the coefficient of evaporation and test results for horsepower, the influence of β is eliminated if the horsepower of an existing locomotive is calculated as suggested above, on the basis of β and of steam rate S_a corrected on the same basis. This will be made still clearer later, when the simplified method is described.

An example may better illustrate the method. Suppose that we wish to find the tractive-effort curve for a 4-6-4 locomotive with 25 × 28-in. cylinders, 225 lb per sq in. working pressure, 79-in. driving wheels, type "E" superheater, feedwater heater and a boiler of given dimensions (New York Central J-1a).

We determine first the Cole evaporation E_c in accordance with the A. L. Co. Handbook and table given in Appendix No. 2, or calculate the evaporation by some other means—for instance, assuming an average evaporation of 12.2 lb per sq ft of total evaporation heating surface. If we follow the A. L. Co. figures, we find that

$$E_c = 54{,}662 \text{ lb of steam per hour}^{19}$$

Suppose that we prefer to figure on the basis of steam rate in relation to piston speed rather than rpm. In this case draw up Table 3 on the basis of the curve, Fig. 6. Then plot the boiler tractive-effort curve DE (Fig. 9). Calculate the cylinder (starting) tractive effort in accordance with Formula [1], and plot a horizontal line AD corresponding to the cylinder tractive effort up to its intersection with the boiler tractive effort. The line ADE represents the indicated tractive effort of the locomotive calculated on the basis of piston speed.

If instead of Fig. 6, Fig. 5 were followed, i.e., with the steam rate referred to crank speed in rpm instead of piston speed, the procedure would be simpler, because no conversion to piston speed would be necessary, as can be seen from Table 4.

The results are plotted on Fig. 9 as curve BC, and ABC represents the indicated tractive effort of the locomotive calculated on

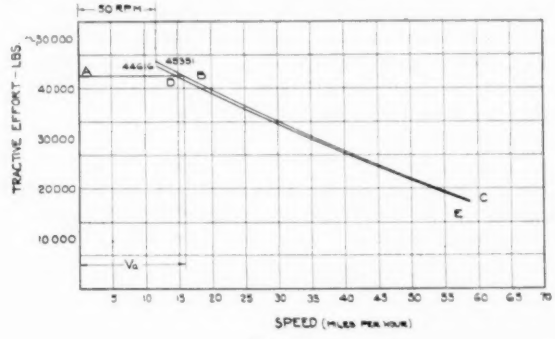


FIG. 9

the basis of crank speed. It can be seen that the difference between curves ABC and ADE is very slight.

SIMPLIFIED METHOD

It can be seen that if the above-described method is carried out on the basis of crank speed in rpm (Table 4), the calculation can be simplified by combining various constants into a single one, and obtaining the indicated-horsepower and tractive-effort figures directly. We already have the following relations:

$$P_i = \frac{E(1-x)}{S_h} = \frac{E}{S_a} \quad \ldots\ldots\ldots\ldots\ldots [6a]$$

$$E = \beta E_c \quad \ldots\ldots\ldots\ldots\ldots [7]$$

TABLE 3

Cole evaporation (E_c)	54,662 lb per hr				
Revolutions per minute (n)	50	100	150	200	250
Coefficient of evaporation (β)	0.65	0.91	1.05	1.07	1.00
Total evaporation (E)	35,530	49,742	57,395	58,488	54,662
Piston speed, ft per min	233	466	700	933	1166
Steam rate, lb per ihp-hr	25.4	21.3	20.3	19.9	19.7
Indicated horsepower (P_i)	1398	2335	2827	2939	2774
Speed, mph (V)	11.75	23.50	35.25	47.00	58.75
Indicated tractive effort, lb (T)	44,616	37,260	30,074	23,449	17,706

TABLE 4

Cole evaporation (E_c)	54,662 lb per hr				
Revolutions per minute (n)	50	100	150	200	250
Coefficient of evaporation (β)	0.65	0.91	1.05	1.07	1.00
Total evaporation (E)	35,530	49,742	57,395	58,488	54,662
Steam rate, lb per hp-hr (S_h)	25.0	21.1	20.2	19.8	19.6
Indicated horsepower (P_i)	1421	2357	2841	2954	2788
Speed, mph (V)	11.75	23.50	35.25	47.00	58.75
Indicated tractive effort, lb (T_i)	45,351	37,611	30,223	23,568	17,795

[19] Other evaporation figures are given in Appendix No. 1.

$$T_i = \frac{375P_i}{V} \quad [5_i]$$

$$V = \frac{Dn}{336.134} \quad [9]$$

Consequently

$$P_i = \frac{\beta}{S_a} E_c \quad [10]$$

and

$$T_i = \frac{375\beta \times 336.134}{S_a \times n} \frac{E_c}{D} = \frac{126{,}050 \times \beta}{S_a \times n} \frac{E_c}{D} \quad [11]$$

or denoting the factors preceding E_c in [10] by M_p, and preceding E_c/D in [11] by M_t, we may write

$$P_i = M_p \times E_c \quad [12]$$

$$T_i = M_t \times \frac{E_c}{D} \quad [13]$$

The values M_p and M_t may be called the "horsepower modulus" and "tractive-effort modulus," respectively, and they may be calculated from the above definitions, namely,

$$M_p = \frac{\beta}{S_a} \quad [14]$$

$$M_t = \frac{126{,}050 \times \beta}{S_a \times n} \quad [15]$$

Coefficient β and the steam rate S_a are functions of the crank speed n; therefore M_p and M_t are also functions of n and may be calculated on the basis of the previously stated values for these variables (Table 2 and Fig. 5). They are given in Table 5.

TABLE 5

Revolutions per minute (n)	50	100	150	200	250
Modulus $M_p \times 1000$[20]	26.00	43.13	51.98	54.04	51.02
Modulus M_t[19]	65.55	54.36	43.68	34.06	25.72

Thus the calculation of horsepower and tractive effort reduces to the operations indicated in Table 6, the results being practically identical with those of Table 4.

TABLE 6

Cole evaporation (E_c)	54,662 lb per hr				
Revolutions per minute (n)	50	100	150	200	250
$M_p \times 1000$	26.00	43.13	51.98	54.04	51.02
$P_i = M_p \times E_c$	1421	2357	2841	2954	2788
M_t	65.55	54.36	43.68	34.06	25.72
$T_i = M_t \times E_c/D$	45,354	37,611	30,222	23,566	17,795

If the tractive effort is the only curve which is desired and the horsepower values are not immediately wanted, the calculation can be still simplified by figuring only T_i and using the modulus M_t.

As an illustration of the simplified method, the previously mentioned locomotives, for which reliable road-test results are available (New York Central L-2, J-1a, Boston & Albany A-1, Timken roller-bearing locomotive, and Lehigh Valley locomotives Nos. 5100 and 5200[21]), have been checked and the results plotted on the charts of Figs. 10–15 in relation to crank speed and locomotive speed in miles per hour. The performance curves previously referred to are drawn on all these charts, and the corresponding simplified-method curves are marked "1932." The Cole curves are also given for comparison, and on two locomotives the capacity-test curves are also shown. It will be seen that the agreement is sufficient for practical purposes, and that the simplified method gives the maximum performance figures very closely. In the case of the New York Central J-1a and Timken locomotives, the discrepancy shown may have been due to the good quality of coal used in the tests.

The 1932 curves are drawn in solid lines up to 200 rpm. The agreement is very good, because the test results, as has already been pointed out, are very consistent up to that speed. For higher speeds there are not sufficient available data on which to base a precise method. For this purpose more than merely the principal dimensions would have to be taken into consideration, namely, some of the ratios already mentioned, such as total heating surface to firebox surface, heating surface to grate area, combustion-chamber surface to firebox surface, type, location and size of exhaust nozzle, valve-motion characteristics, areas of steam passages in cylinders, and quality of fuel expressed in a way suitable for a formula. For average conditions the curve will most probably have, at least for performance tests, a drooping characteristic, as shown for curves 1932, but it is not impossible that it might approach a horizontal line, shown by dots and dashes. More experimental data at high speeds will be necessary in order to impart more definiteness to that portion of the 1932 curve.

It has been pointed out earlier that the ultimate figures for horsepower and tractive effort do not depend upon the evaporation coefficient. This can be more clearly seen from Formulas [14] and [15], in which the ratio of S_a to β is what matters. The steam rates S_a as plotted in Fig. 7 are, as has been explained in the proper place, figured on the basis of the evaporation coefficients of Table 2, and therefore their ratio does not depend upon the numerical value of either of the two. In other words, if instead of variable β, a constant coefficient of evaporation of 100 per cent, as advocated by some investigators, had been taken, the values of S_a shown in Fig. 7 would be changed in the ratio of 1.00 to β. The moduli M_t and M_p would, in this case, remain unchanged because instead of variable β, 1.0 would be inserted in Formulas [14] and [15].

There is no doubt in the author's mind that for locomotive horsepower, the boiler evaporation is the controlling factor, at least for the proportions found in existing locomotives. However, if we imagine a boiler of very large proportions, one able to supply more steam than we usually get in the most modern locomotives, longer cut-offs would become possible, and at high speeds the mean effective pressures might drop below those that are obtained at present for corresponding cut-offs, on account of the steam resistances in passages (wire drawing). It is doubtful whether this would happen on locomotives of conventional proportions, but as a safeguard it may be advisable to set a limit for the possible increase in maximum horsepower compared with Cole's cylinder horsepower, which for superheated steam is expressed by 0.0229 times the area of one cylinder in square inches times the boiler pressure in pounds per square inch.[22]

In the author's opinion, that limit could be made not less than 24 per cent for engines with feedwater heaters, but as a conservative figure, 20 per cent should be recommended until further information can be gathered.

Of locomotives for which the charts of Figs. 10–15 show horsepower and tractive-effort curves, only one, the Lehigh Valley No. 5200, had a boiler sufficiently large to show an increase over the Cole formula of more than 20 per cent. Curve 1932 of Fig. 15 was therefore drawn in such a way as to have a peak point only 20 per cent above the Cole figure.

[20] These moduli apply to locomotives with feedwater heaters; for locomotives without feedwater heaters they should be reduced by 7/107 = 6.54 per cent.

[21] The principal dimensions of these locomotives are given in Appendix No. 1.

[22] A. L. Co. Handbook, 1917, p. 54.

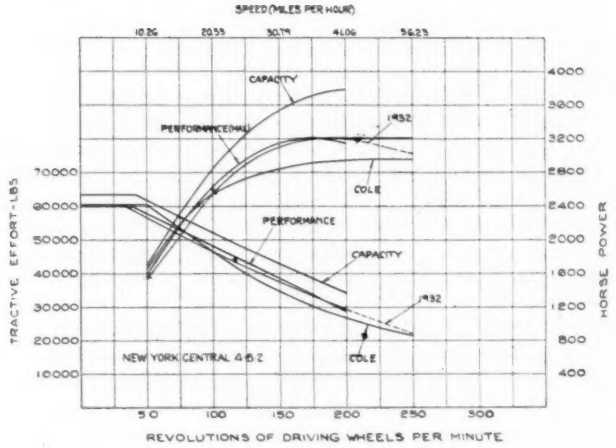


FIG. 10

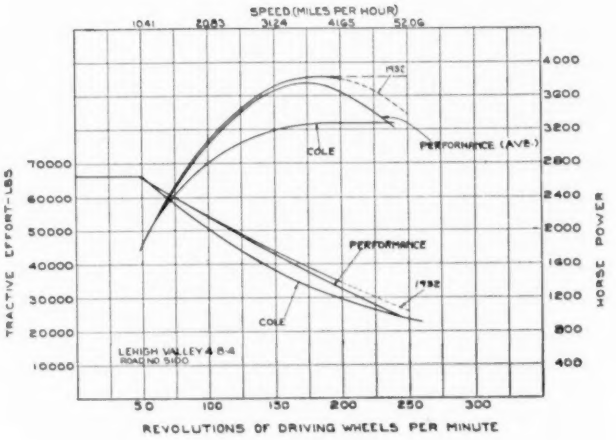


FIG. 14

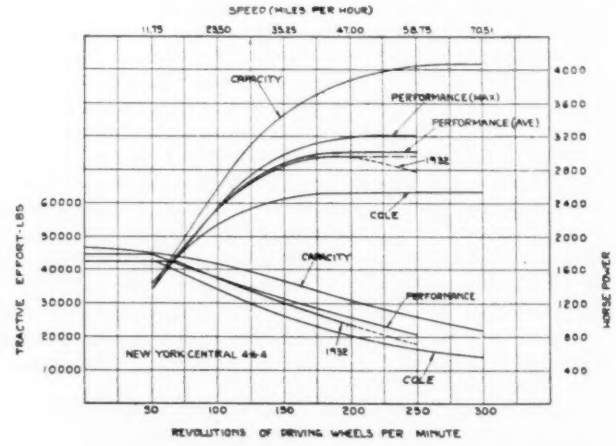


FIG. 11

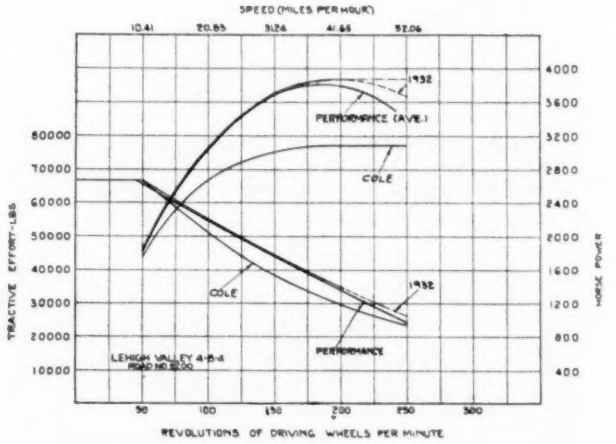


FIG. 15

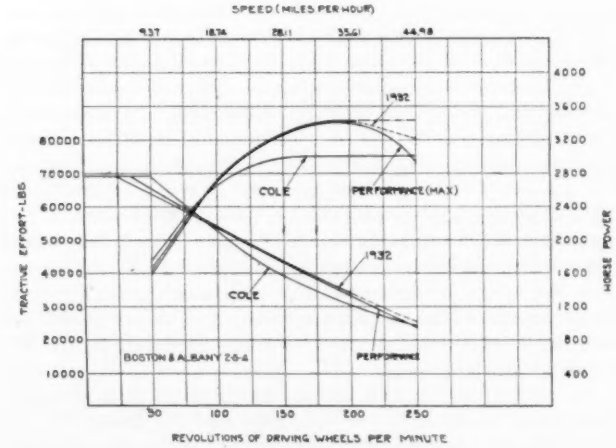


FIG 12

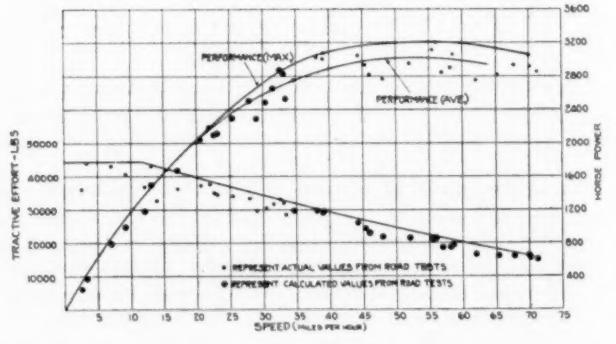


FIG. 16

Conclusion

As was pointed out at the beginning, the object of this paper is to suggest a simple method for figuring horsepower and tractive effort for modern locomotives. To this class belong locomotives with type "E" superheaters, feedwater heaters, and valve motions with about $8^1/_2$ in. of valve travel. This presupposes that the superheater heating surface assures sufficient superheat, which in locomotives with the type "E" superheater is about 250 F. If locomotives have higher superheats, the power may increase, but it was not thought advisable to give constants for various superheats for the reason that before a locomotive is tested, its superheat is not known, and therefore these constants would not be helpful in calculating the horsepower of a locomotive beforehand. It would be logical to base such constants on the relation of the superheating surface to the evaporative heating surface, or to the amount of generated steam, but a comparison of these ratios with test results of various locomotives had not been conclusive.

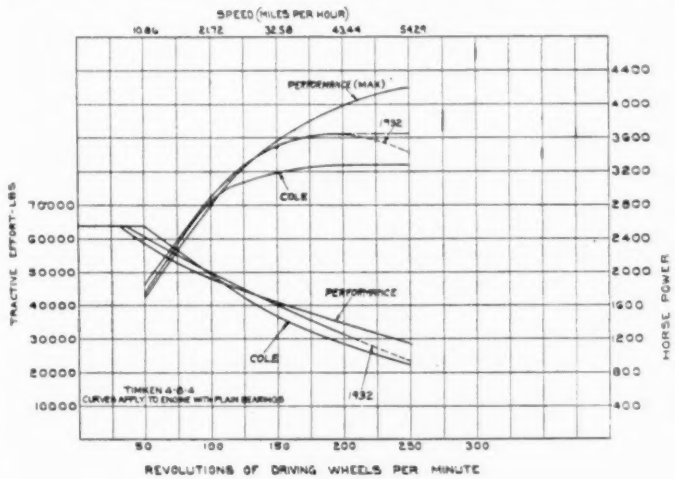


Fig. 13

For locomotives of older design, with type "A" superheaters, new constants could be worked out similar to those given in the paper, but it is suggested that for these latter locomotives the Cole formula should be used. While it would be possible to work out new constants on the basis of boiler horsepower instead of cylinder power, it is thought that until the new method has proved to be practical, the Cole formula should be used for locomotives with type "A" superheaters. This should not imply, however, that type "A" superheaters can never develop the horsepowers recommended by the new method. Locomotives are known that have given very high performance figures with type "A" superheaters on good coal.

A question may arise whether the suggested method can be used in designing locomotives. It is evident from the paper that this can be done very simply. If the desired maximum tractive effort is given, the cylinder dimensions are thus established. If, in addition, the desired speed is given, the horsepower is then established and from Formula [10] the evaporation can be figured. For instance, if a locomotive is desired with a certain tractive effort represented by the line AB, Fig. 9, necessary for pulling a given load on a certain grade, and if the speed V_a is desired on that grade, the horsepower P_i at point B is determined by Formula [5]. Formula [10] enables us then to find E_e because coefficient β and the steam rate S_a at speed V_a are known from Figs. 4 and 7. The boiler can be designed so as to generate E_e pounds of steam per hour.

If the speed for a certain train be given for a tractive effort other than maximum, for instance, on a level, the tractive efforts at this speed, and consequently the horsepower, can be determined as above. Formula [10] will also enable us to calculate the evaporation E_e and to design a boiler accordingly.

For practical purposes the moduli values given in Table 7 can be used.

Acknowledgments

Thanks are due to Mr. J. B. Ennis, vice-president of the American Locomotive Co., New York, N. Y., for encouragement and valuable advice given to the author during the preparation of the paper, and to Mr. C. W. Todd, assistant engineer, American Locomotive Co., Schenectady, N. Y., for his assistance in calculating tables and plotting curves.

Appendix No. 2

Evaporation figures for tubes and flues in pounds of steam per hour per square foot of outside heating surface are given in Table 28 of the A. L. Co. Handbook, p. 59. This table covers 2-in. and $2^1/_4$-in. open tubes and $5^3/_8$-in. and $5^1/_2$-in. flues (with superheater units). Tables 8 and 9 give similar values for $3^1/_2$-in. open tubes and $3^1/_2$-in. flues (with superheater units),

TABLE 7

Revolutions per minute (n)	50	100	150	200	250
Locomotives with feedwater heaters:					
Modulus $M_p \times 1000$	26.0	43.1	52.0	54.0	51.0
Modulus M_t	65.6	54.4	43.7	34.1	25.7
Locomotives without feedwater heaters:					
Modulus $M_p \times 1000$	24.3	40.3	48.6	50.5	47.7
Modulus M_t	61.3	50.8	40.8	31.8	24.0

Appendix No. 1

PRINCIPAL DIMENSIONS OF LOCOMOTIVES

Railroad	New York Central	New York Central	Lehigh Valley	Timken	Boston & Albany	Lehigh Valley	Pennsylvania	Pennsylvania	Pennsylvania	Pennsylvania	Pennsylvania
Class, or serial number	J-1a	L-2	5100	1111	A-1	5200	K-4s	E-6s	K-2sa	L-1s	1-1s
Wheel arrangement	4-6-4	4-8-2	4-8-4	4-8-4	2-8-4	4-8-4	4-6-2	4-4-2	4-6-2	2-8-2	2-10-0
Cylinder diameter, inches	25	27	27	27	28	26	27	$23^1/_2$	24	27	$30^1/_2$
Cylinder stroke, inches	28	30	30	30	30	32	28	26	26	30	32
Wheel diameter, inches	79	69	70	73	63	70	80	80	80	62	62
Boiler working pressure, lb per sq in	225	225	250	250	240	255	205	205	205	205	250
Weights in working order, lb:											
On drivers	182,000	244,000	270,000	264,000	248,200	268,000	202,880	133,100	179,900	235,800	352,500
Total locomotive	343,200	364,000	408,000	417,500	385,000	422,000	309,140	240,000	293,200	315,600	386,100
Heating surfaces, sq ft:											
Firebox	281[1]	354[1]	490[1]	474[1]	337[1]	508[1]	306.8[1]	232.7[1]	208[1]	301.5[1]	287[1]
Tubes and flues	4203[1]	4095[1]	4932[1]	4637[1]	4773[1]	4933[1]	3728.4[1]	2643.9[1]	3451[1]	3728[1]	4487[1]
Total evaporative surfaces	4484[1]	4449[1]	5422[1]	5111[1]	5110[1]	5441[1]	4035.2[1]	2876.6[1]	3659[1]	4029.5[1]	4774[1]
Superheating surfaces, sq ft	1951[1]	1938[1]	2256[1]	2157[1]	2111[1]	2243[1]	1171[2]	810.6[1]	989.32[2]	1171.63[2]	2410[2]
Superheater type	E	E	E	E	E	E	A	A	A	A	E
Grate area, sq ft	81.5	75.2	88.3	88.3	100	88.3	70	55.1	55.4	70	70
Cole evaporation, lb per hr	54,662	59,514	70,530	67,370	62,958	71,694	.....	.....	.....	.....	.....

[1] Wetted side. [2] Fire side.

based on recent tests (New York Central R.R. Gardenville Enginehouse Evaporation Tests, 1923). These tubes and flues are being used in type "E" superheaters.

TABLE 8 EVAPORATION FROM 3½-IN. TUBES (WITHOUT SUPERHEATER UNITS) IN POUNDS OF STEAM PER HOUR PER SQUARE FOOT OF OUTSIDE SURFACE

Length in feet	Spacing in inches						
	5/8	11/16	3/4	13/16	7/8	15/16	1
10	14.72	14.89	15.06	15.23	15.40	15.57	15.74
11	14.38	14.54	14.70	14.86	15.02	15.18	15.34
12	14.06	14.21	14.36	14.51	14.66	14.81	14.96
13	13.75	13.90	14.04	14.18	14.32	14.46	14.60
14	13.45	13.60	13.74	13.87	14.00	14.13	14.26
15	13.16	13.31	13.45	13.58	13.70	13.82	13.94
16	12.88	13.03	13.17	13.30	13.42	13.53	13.64
17	12.61	12.76	12.90	13.03	13.15	13.26	13.36
18	12.35	12.50	12.64	12.77	12.89	13.00	13.10
19	12.10	12.25	12.39	12.52	12.64	12.75	12.85
20	11.86	12.01	12.15	12.28	12.40	12.51	12.61
21	11.63	11.78	11.92	12.05	12.17	12.28	12.38
22	11.41	11.56	11.70	11.83	11.95	12.06	12.16
23	11.20	11.35	11.49	11.62	11.74	11.85	11.95
24	11.00	11.15	11.29	11.42	11.54	11.65	11.75
25	10.81	10.96	11.10	11.23	11.35	11.46	11.56

TABLE 9 EVAPORATION FROM 3½-IN. FLUES (WITH SUPERHEATER UNITS) IN POUNDS OF STEAM PER HOUR PER SQUARE FOOT OF OUTSIDE SURFACE

Length in feet	Spacing in inches						
	5/8	11/16	3/4	13/16	7/8	15/16	1
10	12.77	12.93	13.09	13.25	13.41	13.57	13.74
11	12.35	12.50	12.65	12.80	12.95	13.10	13.25
12	11.95	12.09	12.23	12.37	12.51	12.65	12.79
13	11.57	11.70	11.83	11.96	12.09	12.22	12.35
14	11.20	11.33	11.45	11.57	11.69	11.81	11.93
15	10.84	10.97	11.09	11.20	11.31	11.42	11.53
16	10.49	10.62	10.74	10.85	10.95	11.05	11.15
17	10.15	10.28	10.40	10.51	10.61	10.70	10.79
18	9.82	9.95	10.07	10.18	10.28	10.37	10.45
19	9.50	9.63	9.75	9.86	9.96	10.05	10.13
20	9.19	9.32	9.44	9.55	9.65	9.74	9.82
21	8.89	9.02	9.14	9.25	9.35	9.44	9.52
22	8.60	8.73	8.85	8.96	9.06	9.15	9.23
23	8.32	8.45	8.57	8.68	8.78	8.87	8.95
24	8.05	8.18	8.30	8.41	8.51	8.60	8.68
25	7.79	7.92	8.04	8.15	8.25	8.34	8.42

Appendix No. 3

TABLE 10 MILES PER HOUR FOR A GIVEN DIAMETER OF WHEEL AND NUMBER OF REVOLUTIONS PER MINUTE

(Sample to be extended for other numbers of revolutions and wheel diameters)

Number of revolutions per minute	Wheel diameter in inches				
	63	69	70	73	79
50	9.37	10.26	10.41	10.86	11.75
100	18.74	20.53	20.83	21.72	23.50
150	28.11	30.79	31.24	32.58	35.25
200	37.49	41.06	41.65	43.44	47.00
250	46.86	51.32	52.06	54.29	58.75

Discussion

G. F. STARBUCK.[23] The paper fills an important need in making possible the closer estimate of the indicated tractive effort of modern locomotives than could be made by the Cole method, although that has filled a very useful place for many years. The recounting of the research by which the author reached his method is of value in showing the impossibility of accurately expressing the tractive effort of all locomotives by any one method or formula. Besides differences in steam pressure, superheat, heating surface, grate area, firebox volume (with combustion chamber), draft appliances, and the efficiency of the feedwater heater, we have to consider the condition of the locomotive, quality of the coal, and the ability of the crew to get the most out of the engine. Heating surface will be variously divided into firebox, siphon, combustion chamber, and tube and flue and superheating surface, the tubes and flues being of varying lengths. So, when plotting the results of actual tests, we must expect considerable scattering of the plotted points, with a few of them near the computed line.

Since in formula [4] U is the swept volume of the piston, it seems that the cut-off does depend on the stroke. Moreover, it seems to be the practice to let the diameter of the cylinder, regardless of the stroke, determine the dimensions of the ports, distributing valves, and dry pipe. So it seems from these considerations more logical to consider the steam per indicated horsepower-hour as depending on piston speed.

It is believed that a discussion of the effects of limited and restricted cut-off would be of value. Since the diameter of the cylinder is made larger for short maximum cut-offs than for 85 per cent or longer, it seems that the ratio of the steam rate to the angular velocity would be different. Apparently at least some of the tests forming the basis of the factors of the method were made with a limited cut-off locomotive.

It is often desired to plot tractive-effort curves for locomotives having boosters, and it has been the custom to add the tractive effort of the booster to that of the main engine. That appears to be wrong, for, with the booster working, the cut-off must be shortened at less speed: that is, up to the speed at which the boiler can furnish steam for both the booster and the engine at full cut-off, the tractive effort is the sum of the two efforts; from that speed up to the speed for cutting out the booster, the tractive effort of the main engine is reduced on account of the necessary earlier cut-off or drop in steam-chest pressure.

It may be questioned whether valve travels in the region of 8½ in. are properly to be considered a necessary feature of modern locomotives. Long travel adds greatly to the cost of valve-gear maintenance and increases port openings by but a small amount. It appears better to secure free passage through the ports by the use of a valve of sufficient diameter, made as light as possible by well-known features.

H. B. OATLEY.[24] The paper is of more than ordinary value, as it is a carefully prepared study on a subject that is of prime importance in the design of steam locomotives.

The Cole ratios, which have been considered as a standard for more than a generation, were and still are of great value, and have had widespread use by both railroads and locomotive builders. The marked change in value of factors entering into the design of locomotives during this time has made it necessary that extensions to and modifications of the Cole ratios be made so that there could be comparisons between different types of locomotive and the effect of improvements be evaluated.

The paper offers methods and formulas which, while somewhat more complicated than have been used hitherto, are more easily applied and far less complex than methods used abroad. Any set of equations must of necessity be considered approximate within recognized limits. The interrelation between combustion, heat absorption, and conversion to external work is so intimate that the design of the various parts forming this "linked triangle of fire, water, and air," as well phrased by Mr. Lawford H. Fry, introduces a large number of variables that cannot be accurately determined. These conditions have been considered by the author.

While it is proposed to use rpm instead of piston speed, it appears to be of relatively small moment which is used, as in either case a conversion has to be made to "miles per hour of the locomotive"—which is of course the measure that is of interest in locomotive design.

GEORGE W. ARMSTRONG.[25] The author has fittingly brought out the interdependence of tractive power, a function of cylinders and boiler capacity or output. This was also stressed, to a lesser degree, by the late F. J. Cole in the preparation of his ratios.

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[24] Vice-President in Charge of Engineering, Superheater Company, New York, N. Y. Mem. A.S.M.E.

[25] Consulting Mechanical Engineer, Ridgewood, N. J.

The boiler bears a vital part in the locomotive performance, and it is the uncertainty attendant upon such boiler performance in advance of test data affording knowledge of actual capacity that makes the prediction of engine output the uncertain and variable element pointed out by the author. Front-end design, together with grate design, plays a major part in determining boiler output, boiler design remaining the same. An article in the June, 1932, issue of *Railway Mechanical Engineer* clearly demonstrates this fact.

Regarding the aspect of draft under continuous or intermittent steam flow, it might be added that experiments made with different locomotives using continuous flow on standing tests compared with subsequent observations at similiar back pressures under road operation, where intermittent flow is found, confirm the conclusions cited as reached by Messrs. Goss and Fry. Except at low speeds, where a definite interval exists between exhausts, a close agreement was recorded. Some of these experiments are reported in the Front End Committee report of the International Fuel Association of 1930.

The Pennsylvania test-plant experience for the engines used by the author represents boiler output at the time deemed capacity, with front-end design generally superior to most in use at that time. Subsequent and recent improvements in front-end design have given much greater maximum capacity output; 72,000 lb actual evaporation per hour for the K-4s as contrasted with a former maximum of 55,000 lb; 96,000 lb for the M-1 (4-8-2) as compared with 66,000 lb. This naturally affects the engine or tractive-power output of the locomotive, and developments for improvement in draft conditions and back-pressure reduction have in more than one instance resulted in 200 to 250 hp increased output, due to greater boiler capacity. The great value of front-end improvement lies in the ability to support more efficiently high combustion rates, with consequent increase in available steam for engine output. In meeting the demands for modern operation this phase is important, as it results in higher sustained tractive power at speed.

Evaporations as high as 10 lb of water per square foot of equivalent heating surface have been attained in road operation. On such a basis the New York Central J-1, cited by the author, should be capable of delivering around 78,000 lb of steam per hour, instead of the 54,600 lb which would very materially influence tractive power and horsepower output. Experience with improved draft appliances on locomotives with boiler characteristics similar to this engine indicates that this expectation is not unreasonable. The author has recognized this phase to some degree by placing "the peak point only 20 per cent above the Cole figure."

It might be pertinent, with respect to the feedwater-heater factor, to inquire whether this is predicated upon observations with the same type of heater, inasmuch as the engines cited are predominatingly equipped with one type. However, the factor given corresponds closely with average results, under varying road operation.

In the final analysis, the utility of any method such as Messrs. Cole and Lipetz have devised lies in the comparability that it affords as to a reasonable locomotive performance, a logical standard of measure; and while actual results may be obtained in excess, the standard here proposed admirably serves as a comparometer of equitable value. Serving a useful purpose of pointing out what should be expected, it tends to improve performance or work for its improvement should it fall below this reasonable expectation.

R. Eksergian.[26] This paper is a very illuminating and valuable contribution on locomotive performance. The author has pointed out the many difficulties in accurately predicting locomotive tractive effort and horsepower characteristics, such as on the cylinder side of estimating and allowing for cylinder condensation and wiredrawing against speed, and on the boiler side the difficulties of estimating evaporation and likewise its variation with speed. His recommended method for the calculation of horsepower and tractive force against speed is unquestionably rational and probably the best approximation in a preliminary analysis. For estimating the evaporation, the Cole method has been retained, with a correction coefficient as a function of the speed. The data on steam consumption per horsepower-hour are found to concur with similar data that the writer had occasion to study a few years ago while at the Baldwin Locomotive Works.

However, in the application of the author's method we are immediately confronted with allowances that must be made for variations in boiler proportions and for different types of locomotives. These variations become sufficient to question the need of the Cole evaporation figure. This does not mean that the data supplied by the late F. J. Cole, on evaporative yields for firebox and particularly for tubes of different lengths, are not of extreme value in aiding in the calculation of the correct evaporation of a boiler, but such data become modified, with large grates, combustion chambers, etc., so that the data are perhaps antiquated except for a guide in a first approximation to modern power. The author himself has seen the necessity of this in his correction coefficient β.

The author has pointed out that the analytical method of estimating performance becomes extremely involved due to allowance for cylinder cooling, wiredrawing, etc., and on the boiler side due to proper heat-transfer calculations, combustion and draft considerations, and the effect of grate, firebox, and other proportions on the evaporative yield. However, even crude approximations along the analytical line give surprising results, and such calculations should be encouraged, as in them lies the possibility of improvements in basic proportions.

In other lines of engineering a great degree of success has been obtained by similarity methods; thus in hydraulics the introduction of functions of non-dimensional parameters, such as the Reynolds modulus, has made it possible to bring about an orderly interpretation of otherwise chaotic test data on pipe-line friction losses. The same method has been extended to interpretation of heat-transfer data. For this reason the writer feels that horsepower and tractive-force data could be put in the form of dimensional equations with parameters including important variables in design, and then the variation of the functions of these moduli could be studied from all available statistical data.

A few years ago the writer attempted a preliminary analysis of performance in a paper in *Mechanical Engineering*.[27] The horsepower and tractive-force curve were plotted against rpm and piston speed, and the adhesion range was limited to approximately 50 rpm and the power range beyond 50 rpm. While admitting a small drop in cylinder tractive effort in the adhesion range, the tractive force may be considered in this range to depend essentially on the adhesion limits, where

$$Z_i = \frac{A}{\alpha} = \frac{cW}{\alpha} \text{ (initial tractive force)}$$

A = adhesion weight on drivers

α = factor of adhesion

c = coupling ratio = $\dfrac{\text{weight on drivers}}{\text{total weight of locomotive}} = \dfrac{A}{W}$

[26] Engineering Department, E. I. du Pont de Nemours, Inc., Wilmington, Del. Mem. A.S.M.E.

[27] "Preliminary Power Rating and Tractive Force of Modern Locomotives," vol. 48, 1926, p. 1095.

The limit of the adhesion range was considered at a speed

$$N_a = \frac{9.16\, V_s\, q_a\, H}{d^2 S}$$

where

d = diameter of cylinder (in.)
S = stroke (in.)
N_a = rpm at end of adhesion range
H = heating surface (sq ft)
q_a = unit evaporation at low speed (lb/sq ft)
V_s = estimated specific volume at cut-off (cu ft/lb).

In the subsequent power range, the horsepower was estimated from the relation

$$hp = H\frac{q}{h} = H\frac{q}{f(\epsilon, N)}$$

ϵ = per cent cut-off
N = rpm

The unit or rated horsepower of the locomotive was selected at that corresponding to 200 rpm, and if q_o and h_o are the mean evaporation yield per square foot of total evaporative heating surface and h_o is the corresponding steam consumption at an economical cut-off at 200 rpm, then

$$(\text{hp})_{\text{rated}} = H\frac{q_o}{h_o}$$

and the horsepower at any other speed could be written

$$\text{hp} = H\frac{\beta q_o}{r h_o} = \frac{\beta}{r}(\text{hp})_{\text{rated}}$$

where β corresponds to the author's β, and $rh_o = h$ is obtained from the author's curves 5 and 6.

Now,

$$Z_i = \frac{p_i d^2}{D/S} = \frac{cW}{\alpha} \text{ (starting tractive force)}$$

at speed, $Z = xZ_i$ (or $p_m = xp_i$) and

$$\frac{xZ_iV}{375} = \text{hp}, \quad N = \frac{V}{D} \times 336$$

so that

$$x = \frac{126{,}000}{N}\,\frac{\alpha}{cD}\left(\frac{H}{W}\right)\left(\frac{\text{hp}}{H}\right)$$

Now $\dfrac{\text{hp}}{H} = \dfrac{\beta}{r}(\text{hp})_{\text{rated}}$

The per cent of initial tractive force x, therefore, is

$$x = \frac{126{,}000}{N}\,\frac{\alpha}{cD}\left(\frac{H}{W}\right)\left(\frac{\beta}{r}\right)\left(\frac{\text{hp}}{H}\right)_{\text{rated}}$$

This equation shows the effect on the percentage of the initial tractive force of certain basic proportions in the design of a locomotive. Thus a locomotive with a low factor of adhesion falls off quicker in mean effective pressures against speed than does a locomotive with a high factor of adhesion—that is, an under-cylindered locomotive.

Since the heating surface per pound of weight of locomotive does not vary greatly (i.e., H/W = const.), the falling off of tractive force depends principally on the coupling ratio. A Santa Fe type will fall off quicker than a Mikado, while an Atlantic type, with a low coupling ratio, will sustain a high operating-at-speed tractive force relative to its initial value.

There may be considerable objection of referring the evaporation on the basis of the total evaporative heating surface. This would be true for stationary practice where we find considerable variation in heating surface and grate proportions, with resulting good boiler efficiencies, but with the locomotive, due to construction limitations, the component elements bear a relatively constant proportion of the total heating surface. For this reason the evaporative heating surface appears a satisfactory dimensional unit for characterizing the evaporation of a locomotive boiler. In estimating the mean yield per square foot, however, we must resort to a similar coefficient as the author's β. This would be a function of the proportions of the boiler, a function of the speed, and a function of the rate of firing.

Following the pioneer work of L. E. Fry, assuming the boiler efficiency to fall off at a rate linearly with the rate of firing, it can be shown that the unit of evaporation can be expressed by the relation

$$q = \left[\frac{\text{Btu}_c}{(HA_v)\Phi}\right]\left[\frac{K_x - mx^2}{100}\right]$$

where

Btu_c = heat value of the coal
Φ = total heat of steam, including that for superheating
x = coal rate per hour per square foot of grate
H = evaporative heating surface
K = 75 to 85
m = 0.2 to 0.3.

It would appear from this relation that increasing the grate would increase the evaporative yield. This is true provided satisfactory firebox volume can be maintained. But the essential feature of the foregoing relation is that it gives the characteristics of firing rate and evaporation.

In conclusion, it is felt that the steam consumption per horsepower can be estimated much more accurately than the evaporative yield, and for this reason further study is necessary to evaluate such a coefficient as the author's β for different boiler proportions as a function of speed, rate of firing, etc., whether the yield is expressed in terms of the Cole unit or a unit yield per square foot of heating surface of the boiler.

THOS. C. MCBRIDE.[28] The system of locomotive ratios proposed by the author considers the feedwater heater and its effect in increasing the maximum evaporation obtainable from the boiler. The statement of this increase introduces a new and important ratio. A large amount of information that has been collected over a number of years on locomotives without heaters is made available in the consideration of present-day locomotives with heaters by the application of this new ratio. Careful consideration and accurate statement of this new ratio are therefore important.

The author reaches the conclusion that the Cole evaporation figures still hold good for boilers on modern locomotives without feedwater heaters, but when locomotives are equipped with feedwater heaters, the boilers generate 7 per cent more steam. In contrast with this conclusion, feedwater temperatures obtained from feedwater heaters on locomotives operating at the maximum rates of the Cole ratios indicate an average of approximately 15 per cent additional heat to the boiler through the heater, and this additional heat can get out of the boiler only through a like increase in evaporation. The difference between these two figures is too great to neglect, especially in developing a system of locomotive ratios.

The author has taken great pains to collect and carefully apply

[28] Consulting Engineer, Railroad Department, Worthington Pump & Machy. Co., Philadelphia, Pa. Mem. A.S.M.E.

a mass of data obtained partly from road trials of locomotives, but road trials necessarily involve large "probable errors." A few temperature readings taken simultaneously while the locomotive is known to be working at the desired maximum make it possible to calculate the extra heat going to the boiler through the heater with a probable error of but a few per cent. It is necessary only to read the temperatures of the water entering and leaving the heater and the steam pressure and superheat at the moment the indicator card or other check is made on the capacity of the locomotive. The process can be repeated every few minutes and more exact information gathered in an hour than could be obtained from a month of road trials.

The writer has shown that (assuming 200 lb boiler pressure, 150 F superheat, and 215 F feedwater temperature obtained, with alternate assumptions of water at 40 F from the tender to represent winter and 70 F to represent summer conditions) the additional heat supplied to the boiler through the heater is 13.6 and 11.1 per cent, respectively. That is, presuming exactly the same amount of fuel burned and exactly the same amount of heat transmitted to the water and steam in the boiler, with the consequent same efficiency of the boiler, the feedwater heater increases the heat to the boiler and consequently the evaporation from the boiler by 13.6 and 11.1 per cent, respectively, under the conditions assumed.[29]

These assumed conditions were considered representative of usual road operation in 1920. When locomotives are operated at the maximum contemplated in the ratios, much higher feedwater temperatures are obtained, generally 240 F and frequently 250 F.

If the conditions assumed in the paper of 1920 are again used for the sake of comparison, but with a feedwater temperature of 240 F, each pound of water will carry into the boiler from the heater 240 — 40, or 200 Btu, and will require 1284.6 + 32 — 240, or 1076.6 Btu from the fire for generation into steam. The heater increases the heat from the fire by 200/1076.6, or 18.6 per cent. But of the total of 118.6, 2 per cent, or 2.4, is required to operate the feed pump, leaving 118.6 — 2.4, or 16.2 per cent net increase in heat because of the heater. Similarly, with the alternate assumption of water at 70 F from the tender, each pound of water carries into the boiler 240 — 70 or 170 Btu, and requires from the fire the same 1076.6 Btu. The addition to the heat from the fire by the heater is 170/1076.6, or 15.8 per cent. Of the total of 115.8, the feed pump requires 2 per cent, or 2.3, for its operation, leaving 113.5, or a net of 13.5 per cent additional heat to the boiler through the heater.

The average of 16.2 and 13.5 is 14.8, or practically 15 per cent. It is thought that this figure, or some figure close it, depending on the conditions assumed to represent average present-day locomotive-operating conditions, should be adopted for the ratios instead of the 7 per cent obtained from road trials.

In constructing the system of locomotive ratios, the author rightly starts at the boiler. It is desired to determine the maximum evaporation that can be obtained from the boilers of locomotives on the road. The use that can be or is made of the steam is not considered until later. The limitation of the maximum boiler capacity obtainable is the amount of fuel that can be burned under the existing conditions of fire and fuel and the existing condition of the boiler, or both. Tests to determine the increase in maximum boiler capacity obtainable on the road from heaters should be made with the same locomotive without the heater and with the heater, the heater necessarily of sufficient capacity to supply the entire maximum capacity requirement of the boiler, and with fuel burned at exactly the same rate. Could such tests be made and all errors eliminated, there is no doubt that they would show the same increase in maximum boiler capacity because of the heater as is indicated by the temperatures.

The condition of the fire is a factor controlling the maximum evaporation obtainable from boilers of locomotives to an extent seemingly worthy of consideration. At one time it was thought that 12 lb of water per square foot of evaporative heating surface was the maximum that could be depended upon from boilers with an old (or older) fire, but that 15 lb could be obtained with a new fire. If this is true, the condition of the fire, not considered in the ratios, introduces a variable greater in extent than change in rpm at usual speeds or even the feedwater heater. Neglect of the ratios to consider condition of the fire brings about a condition in which the locomotive builder must furnish a boiler big enough to meet expected performance at the 12-lb rate, but appliances like the feed pump might be expected to be of sufficient capacity for the 15-lb rate, even though this 15-lb rate of the combination of maximum power requirement and fresh fire were called for but very occasionally. Feed pumps would then on the average have to be 25 per cent too big for anything but the very occasional requirement. This situation may be difficult to handle in the ratios, at least until much more road test data are available, but some recognition of it and its consequences seems important.

In summing up, the writer believes we are bound to recognize the differential of 15 per cent or thereabout increased evaporation in favor of the boiler with heater, in view of the direct and accurate evidence of the temperatures obtained. It will then be necessary to modify Fig. 4 and Table 2, spreading the "evaporation coefficient" to a differential of 15 per cent at high and medium rpm. But over all this looms the condition of the fire, new or old, demanding at least recognition.

The use of feedwater heaters on locomotives naturally tends to reduce the superheat, because more steam is generated for the same amount of fuel burned. At the same time there is a reduction in back pressure because of the exhaust steam taken by the heater. For a number of years as a result of tests these two effects were considered as counterbalancing with no change in steam consumption per indicated horsepower because of the heater. In that case the 15 per cent increase in maximum heat obtained through the heater resulted in 15 per cent increase in maximum indicated horsepower obtainable. Locomotives intended for operation with heaters are now provided with extra superheating surface; the intended superheat is obtained and the steam consumption per indicated horsepower is decreased because of the reduced back pressure obtained with the heater. The 15 per cent additional heat to the boiler therefore results in more than 15 per cent increase in maximum indicated horsepower obtainable from the same maximum fuel burned. Since the machine friction of the locomotive is not changed by the application of the heater, there will be a still greater increase in tractive effort, and since the locomotive resistances are also not changed by the heater, there is a still greater increase in maximum drawbar pull at speed because of the heater. These further increases in the maximum power obtainable on the road are substantial in amount, but vary with individual cases. They will appear as the method of calculation recommended by the author of this paper is pursued.

HARRY S. VINCENT.[30] The author has adopted the convention that the low-speed end of a tractive-effort graph can be represented by a horizontal line. The writer also accepted this convention, until his attention was called to the fact that all locomotive tests made at low speed prove that the tractive effort can

[29] "Locomotive Feedwater Heating," by Thomas C. McBride, Trans. A.S.M.E., 1920, vol. 42, p. 307.

[30] Retired. Late Chief Consulting Engineer, Franklin Railway Supply Company, New York, N. Y. Mem. A.S.M.E.

be most accurately represented by a slanting line. Those who have ridden the footboard of a freight locomotive know very well that a heavy train can be urged over the "sticking point" at a speed of 2 mph, when 8 or 10 mph would be impossible.

In establishing the rated tractive effort of a locomotive, it is the custom to select some point on the speed-effort curve, calling this the nominal effort of the locomotive. At this rating speed, the locomotive is assumed to develop in the cylinders a mean effective pressure of 85 per cent of boiler pressure. From this is developed the well-known tractive-effort formula:

$$TE = \frac{d^2 s 0.85\,P}{D} \qquad [16]$$

in which

TE = nominal or rated tractive effort, lb
d = diameter of cylinder, in.
s = stroke of cylinder, in.
P = boiler pressure, lb per sq in.
D = diameter of driving wheels, in.

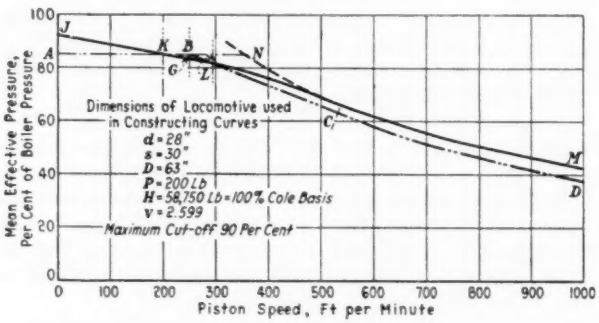


FIG. 17

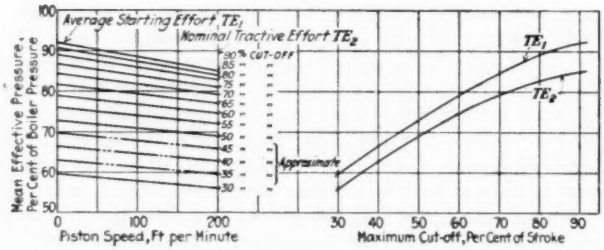


FIG. 18

The broken lines in Fig. 17 indicate by $ABCD$ a graph based on the speed factors shown in Table 14 of "Locomotive Handbook," American Locomotive Company, 1917, p. 24, generally known as the Cole factors. The horizontal scale represents piston speed in fpm and the vertical scale percentage of boiler pressure. It is characteristic of these tables that the graph constructed from them is composed of a horizontal straight line A-B, a slanting straight line B-C, and a hyperbola C-D. In making tractive-effort curves from these factors, it has always been the custom to join the two straight lines by a curve, indicated by the dotted curve G, this curve terminating at a piston speed of approximately 200 fpm. The writer's experience of many years has convinced him that the tractive effort of a locomotive at low speed cannot be indicated by a horizontal line, such as A-B. It can be shown more accurately by the slanting line J-K, in which K represents the nominal or rated effort and J the average effort at starting, or zero speed. The examples of actual road tests, shown by Figs. 19 and 26, would seem to substantiate the writer's contention.

The vertical position and slope of the line J-K depend entirely on the maximum cut-off for which the locomotive is designed. It is evident, that as the maximum cut-off is reduced, the nominal or rated effort, for any given engine constant, must be represented by a lower percentage of the boiler pressure. For instance, a locomotive having a maximum cut-off of 90 per cent, has a rated effort, based on $0.85\,P$, whereas one having a maximum cut-off of 60 per cent, will be rated at $0.745\,P$.

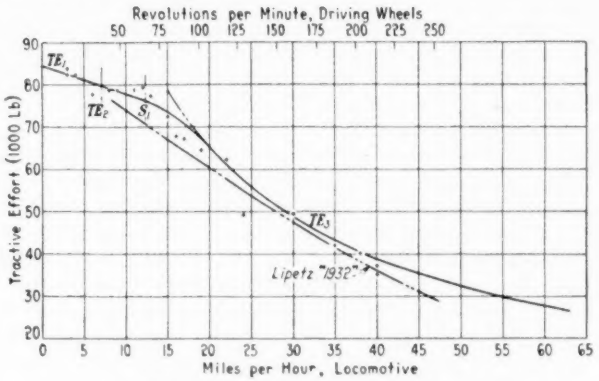


FIG. 19 TEXAS & PACIFIC 2-10-4

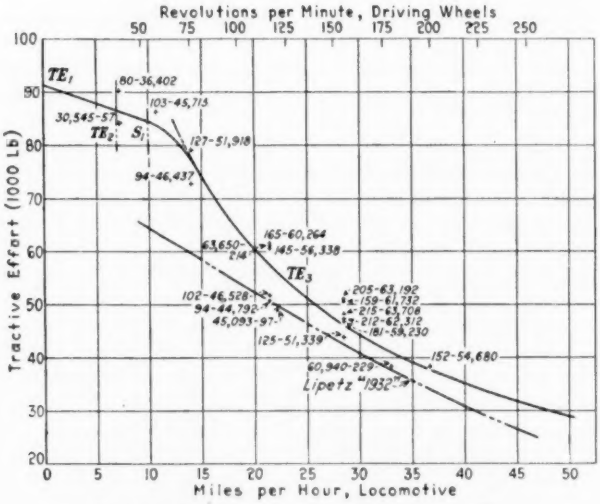


FIG. 20 PENNSYLVANIA RAILROAD CLASS I1s

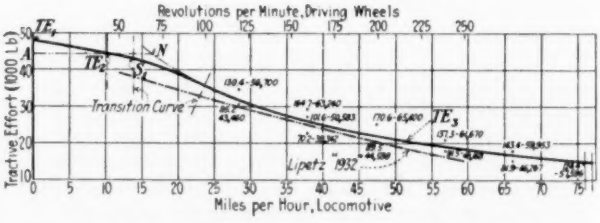


FIG. 21 PENNSYLVANIA RAILROAD CLASS K4s

Fig. 18 gives the position and slope of the line J-K, for maximum cut-off, varying by 5 per cent, from 30 to 90 per cent. These data have been used very successfully by the writer for several years, in representing the tractive effort of modern locomotives. One very important result of adopting the slanting line J-K in place of the horizontal line A-B, to represent tractive effort at low speed, and of making the point K the rating point, is that it establishes a norm by which all locomotives can be measured and compared. The adoption of the straight line A-B would permit any locomotive builder to claim, with reason,

a higher rated power for a locomotive than is justified, if a test shows that its tractive effort comes anywhere on the lines *A-B*, although it is self-evident that of two competing locomotives that one is most powerful which gives its rating nearer *B* than *A*.

The author's recommended method of expressing the tractive effort of a locomotive is dependent entirely upon the validity of the values shown in his Figs. 4 and 5. Like any other power plant, the locomotive is designed to deliver its rated effort at a predetermined efficiency. Like other power plants, it is often pushed to a rating far beyond the normal, involving a decrease in thermal efficiency, although perhaps increasing its operating efficiency as a transportation unit. This is well illustrated by the author's Fig. 2, in which the five Pennsylvania locomotives

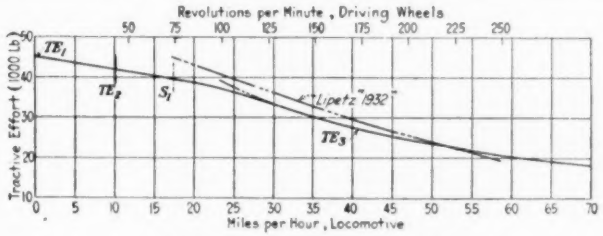


FIG. 22 NEW YORK CENTRAL 4-6-4

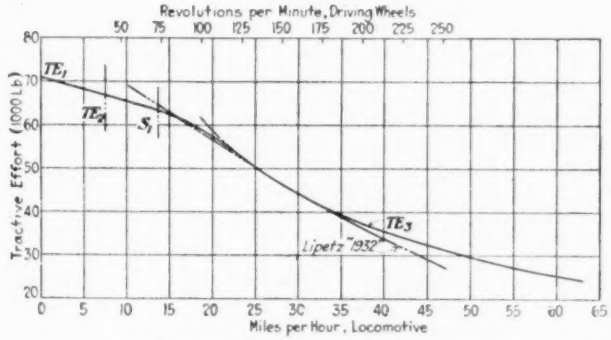


FIG. 23 BOSTON & ALBANY 2-8-4

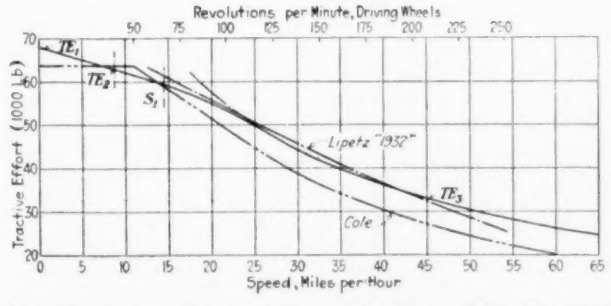


FIG. 24 TIMKEN ROLLER BEARING COMPANY 4-8-4 NO. 1111

shown were pushed to their limit on the test plant, in one instance developing an evaporation as high as 28 per cent above the mean for which it was designed. This sort of operation cannot be used as a basis for developing a tractive-effort formula. Such a formula should be based on the normal operation of a locomotive designed in accordance with the most approved engineering practice known.

In the writer's opinion the best available practice does not justify the drop in the evaporation curve beyond 200 rpm as shown in the author's Fig. 4. There is no apparent reason why a locomotive boiler, after it has reached its normal evaporation, should not continue indefinitely at the same rate, provided (*a*) it is properly proportioned, (*b*) the fuel burned does not exceed the normal hourly rate per square foot of grate surface, (*c*) it has a well-designed drafting arrangement, and (*d*) the steam-using apparatus (the cylinders, valves, and valve gear) are so designed and constructed that they will supply to the drafting apparatus

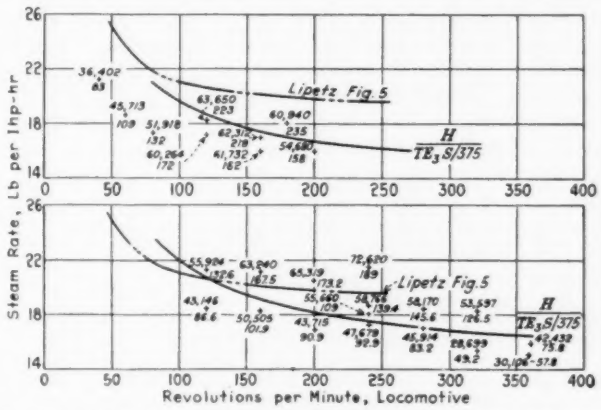


FIG. 25 (Above) PENNSYLVANIA RAILROAD I1s, FROM BULLETIN NO. 32. (Below) PENNSYLVANIA RAILROAD K4s, FROM BULLETIN NO. 29 AND LATER TESTS

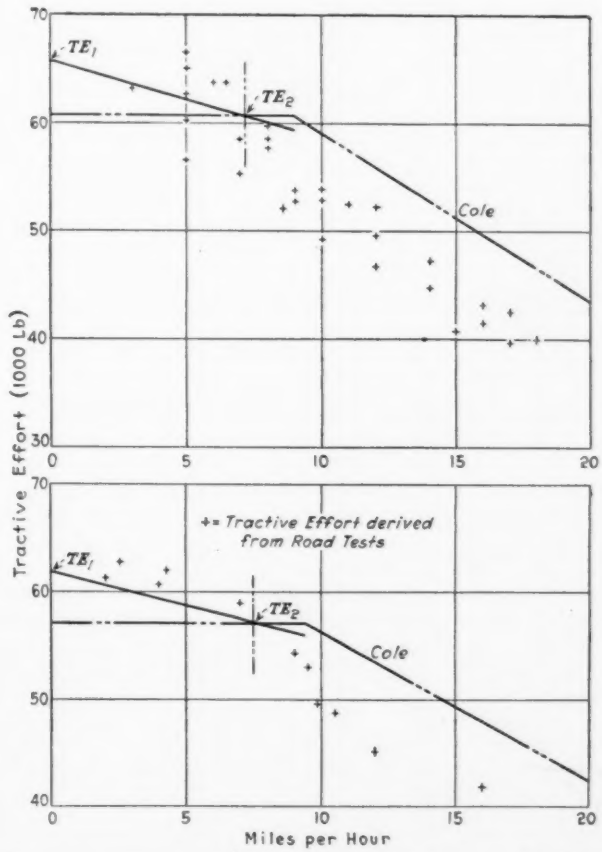


FIG. 26 (Above) FREIGHT LOCOMOTIVE 2-8-2, 29 × 28, 56-IN. DRIVING WHEEL. (Below) FREIGHT LOCOMOTIVE 2-8-2, 28 × 30, 63-IN. DRIVING WHEEL

equal weights of steam per unit of time. It is the requirement (*d*) that usually fails, causing the evaporation curve to assume the droop shown by Fig. 4.

The author's tractive effort modulus *Mt* is predicated on a

steam rate which is assumed to be constant for any given rpm or piston speed, illustrated in his Fig. 5. In discussing this curve, the author states that "only a portion of the curves, representing the most frequent average speeds of road tests, could be plotted"; for the remainder of the curves he uses an "inverted method" based on several assumptions. It has been the writer's experience that there are very few road tests of a locomotive which can be used for establishing a normal horsepower or tractive-effort curve, for the reason that the results obtained in such tests are average results; that is, the horsepower, the water and fuel consumption, the speed, etc. are given as the average over a run. It is only in those exceptional cases, where a test is made under constant conditions of grade and speed, that the results may be used in establishing a rating curve. Road tests are of course invaluable to a railroad, as they give information which can be used by the transportation department in fixing and adjusting tonnage.

The results obtained from operation of a locomotive on a test plant, when intelligently analyzed, give, in the opinion of the writer, far more useful information for the construction of rating curves than can be secured from road tests. In test-plant operation the uniform conditions under which each test is conducted can be continued as long as necessary to secure the desired result; this is impossible in a road test. The apparatus for recording the data can be operated more efficiently and accurately than is possible on a moving locomotive or car, and the test is conducted by a staff, each of which is thoroughly familiar with his duty.

In adopting the water rates given in his Fig. 5, the author makes no allowance for differences in maximum cut-off. Such differences materially affect water and fuel rates, especially at the lower speeds. They have an appreciable effect at all speeds. This will be discussed later.

There are six major factors involved in the calculation of the tractive effort of a locomotive; these are:

(*a*) The engine constant
(*b*) The normal evaporation of the boiler, lb of steam per hour
(*c*) the working pressure, lb per sq in.
(*d*) the maximum cut-off
(*e*) the average steam temperature in the cylinders
(*f*) the speed of the locomotive.

In the writer's opinion, the most practicable formula for combining these elements in one equation to form the basis for a correct speed-effort curve was devised by W. F. Kiesel, Jr., mechanical engineer of the Pennsylvania Railroad, and first published in "Locomotive Operation and Train Control," by A. J. Wood, 1915, p. 27. The formula has been successfully used by Mr. Kiesel for many years, being derived from results obtained on the Altoona test plant and modified to apply to road conditions. This formula, as slightly altered and transposed by the writer, is:

$$TE_3 = \frac{1.95\,P_1M}{1 + (36.66\,MS)/(Hv)} \quad \ldots\ldots\ldots [17]$$

in which

TE_3 = indicated tractive effort, lb
P_1 = average initial pressure in cylinder = boiler pressure less 10 lb
M = the engine constant = d^2s/D
S = linear speed of locomotive, mph
H = the normal evaporation of the boiler, in lb steam per hr
v = specific volume of steam at P_1 pressure, and 100 deg superheat, assumed to be the cylinder average.

The writer has availed himself of an article, "The Kiesel Train-Resistance Formulas," by A. J. Wood, *Railway Mechanical Engineer*, December, 1916, giving the complete derivation of the Kiesel formula. In addition to the notations given elsewhere, let

E = the expansion ratio
mep = mean effective pressure in cylinder
n = revolutions per minute of driving wheels.

Then

$$\frac{H}{60} = \text{lb of steam per min}$$

$$\frac{H}{60 \times 4 \times n} = \frac{H}{240\,n} = \text{steam per stroke}$$

$$\frac{\pi d^2 s}{1728 \times 4 \times v} = \text{weight of one cylinder full of steam}$$

$$E = \frac{\pi d^2 s}{1728 \times 4 \times v} \times \frac{240\,n}{H} = \text{expansion ratio}$$

But $n = \dfrac{1056\,S}{\pi D}$

therefore $E = \dfrac{\pi d^2 s \times 240 \times 1056\,S}{1728 \times 4 \times H \times \pi D \times v} = \dfrac{d^2 s \times 36.66 \times S}{H \times D \times v}$,

But $\dfrac{d^2 s}{D} = M$

therefore $E = \dfrac{36.66\,MS}{Hv}$

$$\text{mep} = \frac{1.95\,P_1}{1 + E} = \frac{1.95\,P_1}{1 + (36.66\,MS)/Hv}$$

since $TE_3 = \text{mep}\,\dfrac{d^2 s}{D}$

$$TE_3 = \frac{1.95\,P_1M}{1 + (36.66\,MS)/Hv} \quad \ldots\ldots\ldots [17]$$

The term E, called by the late Professor Wood the expansion ratio, is not what is usually known by that term, but is a ratio found by dividing the weight of a cylinderful of steam at the initial pressure by the weight of steam actually passed through a cylinder at each stroke. In terms of volume, this expansion ratio is found by dividing the cylinder volume by the volume which the steam passed through in one stroke would have at the initial pressure. The writer acknowledges his indebtedness to Mr. Lawford H. Fry for this definition of the term E.

A comparison of the tractive-effort curves derived from the author's recommended constants with those resulting from the formulas proposed by the writer is given in Figs. 19 to 24, inclusive. The specific dimensions of the locomotives used in making curves Figs. 20 to 24, inclusive, are given in the author's Appendix No. 1. The dimensions of the locomotive for Fig. 19 are appended.

In Fig. 19 the crosses represent the tractive effort derived from indicator cards taken at various speeds in a road test of the locomotive. The curves TE_1, TE_2, and TE_3 are constructed by the method described herein. It is evident that the curves closely approximate the actual tractive effort developed. Particular attention is directed to the low-speed portion TE_1, TE_2, and S_1, which represent the actual test much closer than would a horizontal line drawn through the rating point TE_2.

The moduli of the author, as given in his Table 7, were used

to construct the curves marked "Lipetz 1932." The curve is fairly consistent with the test results for speeds above 25 mph, but falls below the indicated effort at the lower speeds. This locomotive has a maximum cut-off of 60 per cent of the stroke; consequently, the steam consumption per ihp-hr is lower than contemplated in the author's Fig. 5. Therefore the tractive effort based on evaporation is correspondingly higher.

The test data, as given in the writer's Figs. 20 and 21, are taken from Pennsylvania Railroad Bulletin No. 29, record of test of class K4s, passenger locomotive, and No. 32, record of test of class I1s, heavy freight locomotive. The crosses represent the tractive effort as derived from the indicated horsepower developed at various speeds. The figures shown adjacent the crosses give the fuel consumption per square foot of grate surface and the evaporation rate of the boiler at the time of making the tests. The tractive effort curves TE_1, TE_2, and TE_3 are constructed by the method described herein. It is evident that they closely approximate the test results. The boiler of the I1s has a rated hourly evaporation of 61,718 lb, but due to the relatively small grate this rating was developed only at high unit fuel consumption.

The K4s locomotive has a boiler rating of 51,100 lb steam per hr, and has ample grate area to deliver this evaporation at a normal fuel rate. The curve TE_3 for this locomotive approximates a position consistent with the rated evaporation.

The curves marked "Lipetz 1932," as shown in the writer's Figs. 20 and 21, are derived from the author's constants. They approximate the test results at the higher speeds, but are evidently too low at the lower speeds. The I1s locomotive has a maximum cut-off of 55 per cent, therefore giving steam rates much below those shown in the author's Fig. 5. The K4s locomotive has a 90 per cent maximum cut-off, to which the author's moduli should apply. It is evident, however, that they are not suitable for the lower speed range in this locomotive.

Figs. 22 to 24 represent the same locomotives as do the author's Figs. 11 to 13. Tractive effort curves TE_1, TE_2, and TE_3 are drawn in conformity with the writer's method. The curves marked "Lipetz 1932" are constructed by the formula of the author from the constants given in his Table 7 for locomotives with feedwater heaters. It will be observed that they are not the same as the "1932" curves, shown in the author's Figs. 11 to 13. He has evidently used constants for locomotives without feedwater heaters, although the records indicate that heaters were used.

In the writer's Fig. 24 a curve is shown marked "Cole"; this is constructed in accordance with the Cole factors, in which, at the nominal effort, mep equals 0.85 P. The Timken locomotive has a maximum cut-off of 85 per cent; the nominal mep is therefore 0.84 P. For this reason the rated tractive effort TE_2 falls slightly below the Cole curve.

The author has given no data for locating the horizontal portion of his curves except his Formula [2]. It is evident from his curves Figs. 12 to 13 that he does not use 0.85 P for limited cut-off locomotives.

Referring to the writer's Fig. 21, the sloping line $TE_1 - TE_2$ is extended to S_1. At the latter point the straight line becomes tangent to the transition curve, connecting the straight portion of the curve with the hyperbola. The locus of S_1 is a function of the boiler capacity. Its location may be determined graphically by drawing a horizontal line through TE_2 intersecting the hyperbola at N. S_1, then, equals 0.84 AN. The position N is the point on the hyperbola where the cylinders would utilize the normal boiler capacity at maximum cut-off if the tractive effort remained constant from A to N. Stated as an equation,

$$S_1 = \frac{(1.95\ P_1 M) - TE_2}{(36.66\ MTE_2)/(0.84\ Hv)}$$

The transition curve between the straight portion of the graph $TE_1 - S_1$ and the hyperbola TE_3 is empirical. No error of any consequence is possible by this process.

The writer has shown in his Fig. 25 the steam rate as related to rpm of the driving wheels, developed in tests of the Pennsylvania Railroad locomotives, class I1s and K4s. The crosses represent the steam rates at the evaporation and unit fuel consumption shown by the figures adjacent them. The full-line curve is the steam rate obtained by dividing the rated evaporation of the boiler by the indicated horsepower corresponding to the tractive effort curves TE_3 of the writer's Figs. 20 to 21. These curves fit in with the test results, much better than do the curves in broken lines, which are constructed in accordance with the author's Fig. 5. This is especially true of the I1s locomotive, constructed with 55 per cent maximum cut-off.

The formula for constructing the hyperbola TE_3 involves somewhat more work than does the author's moduli. Usually all such calculations are made with a slide rule, so that the additional labor is negligible. In the writer's practice, a simplified form of calculation is used, giving the same results. An example of this is given in Table 11, using the dimensions of the Timken locomotive, as the basis of calculation. The tractive-effort curves resulting are given in Fig. 24.

TABLE 11 SIMPLIFIED METHOD FOR CALCULATING THE HYPERBOLA TE_3, FIG. 24

(Based on dimensions of Timken locomotive No. 1111.)

d = 27 in.	H = 72,540 lb	Constants	Maximum cut-off
s = 30 in.	r = 2.111	1.95	85 %
D = 73 in.	P_1 = 240 lb	36.66	

$$\frac{72{,}540 \times 2.111}{36.66} = 4175 \qquad \frac{27^2 \times 30}{73} = 299.5$$

$$1.95 \times 240 \times 4175 = 1{,}953{,}000 \qquad 4175/299.5 = 13.95$$

Mph		Hyperbola TE_3
10	1,953,000/23.95	81,600
15	28.95	67,500
20	33.95	57,550
25	38.95	50,200
30	43.95	44,450
35	48.95	39,900
40	53.95	36,200
45	58.95	33,150
50	63.95	30,550
55	68.95	28,350
60	73.95	26,400
65	78.95	24,750

$$TE_1 = \frac{27^2 \times 30 \times .908 \times 250}{73} = 68{,}000 \text{ lb}$$

$$TE_2 = \frac{27^2 \times 30 \times 0.84 \times 250}{73} = 62{,}900 \text{ lb}$$

$$S_1 = \frac{(1.95 \times 240 \times 299.5) - 62{,}900}{\dfrac{36.66 \times 299.5 \times 62{,}900}{0.84 \times 72{,}540 \times 2.111}} = 14.4 \text{ mph}$$

TABLE 12 PRINCIPAL DIMENSIONS OF LOCOMOTIVE FOR FIG. 19

(Texas & Pacific Railroad.)

Class	I-1
Wheel arrangement	2-10-4
Cylinder diameter, in.	29
Cylinder stroke, in.	32
Wheel diameter, in.	63
Boiler working pressure, lb per sq in.	250
Weights in working order, lb:	
On drivers	300,000
Total locomotive	448,000
Heating surfaces, sq ft:	
Firebox	473
Tubes and flues	4640
Total evaporative surfaces	5113
Superheating surfaces, sq ft	2100
Superheater type	E
Grate area, sq ft	100
Feedwater heater	Elesco

The writer's tractive-effort formulas, as given here, have been in successful use for a number of years. They are applicable in principle to all simple multi-cylinder locomotives. From examples shown, it is evident that they are more general in their application than those submitted by the author; therefore it

would appear that there is no good reason for the adoption of a new method.

W. A. POWNALL.[31] The author has given something of much value in trying to arrive at a method of determining the horsepower and tractive effort of the modern steam locomotive. The writer has done considerable work on the rating of locomotives for fast freight trains, using a new method of dividing the resistance per ton on a so-called "equivalent grade" for each district into the theoretical drawbar tractive power at the desired speeds. In actual service the modern engine made a better performance than what had been predetermined by calculation, bringing out the fact that the tractive-power curve determined by Cole's method needed raising somewhat for modern engines, just as pointed out by the author. The revised methods proposed should produce approximately correct horsepower and tractive-power curves for the modern locomotive of conventional design with type-"E" superheater, feedwater heater, etc. However, there seems to be a way to materially increase the horsepower and tractive power at high speeds through increased evaporative capacity and lower steam rate through reduced back pressure. This can be done with a different design of exhaust nozzle, front-end appliances, and a larger stack. Recent experience with such designs has shown that a considerable increase in evaporative capacity of the boiler can be obtained, and although this of course means an increase in the rate of coal burned per square foot of grate area, and which is attended by a falling off in boiler efficiency, the general result is a considerable increase in locomotive capacity.

Fig. 13 shows the Timken engine with a performance curve much higher than the Cole or the proposed method. The author states that this may be due to a good quality of coal, but it is believed that this exceptional performance at high speeds is due to the type of front end used on the Timken locomotive. The exhaust nozzle is a six-ported star-shaped design, of larger area than the customary round nozzle tip, but which at the same time produces a steam jet with greater entraining power due to its greater periphery. The usual draft plates and deflector plates in the front end are dispensed with, and in their place is an inside stack with flare, the lower edge of which is about 12 in. above the bottom of the smoke arch, and the netting consists of a straight cylinder extending from the lower edge of the stack to the bottom of the smoke arch. The stack itself is somewhat larger than the conventional stack. This arrangement produces very strong draft, and with less back pressure on the cylinders, and because of the strong draft, grates with reduced air opening have to be used.

The Wabash has equipped a 4-8-2 type locomotive with similar front-end arrangement, and has obtained as high as 4900 ihp at a speed of 55 mph. The maximum cylinder horsepower by the Cole method would be 3215, and by the Lipetz method 3237. This engine in actual service evaporated at times water at the rate of about 80,000 lb per hr, whereas the maximum evaporation from the Cole method would be 63,764. The evaporation based on the Cole method is figured with coal burned at the rate of 120 lb per sq ft of grate area per hour, whereas this rate was very much exceeded during the actual performance of the engine in question. Tests of Pennsylvania Railroad engines equipped with this front-end arrangement (this design of front end was developed on the Pennsylvania) have shown similar high evaporative rates and increased horsepower. The high horsepower developed is undoubtedly contributed to by a lower water rate due to reduced back pressure, and the writer would point out this low water rate on the Timken engine as indicated in Figs. 7 and 8 of the paper.

[31] Wabash Railway Company, Decatur, Ill.

At the present time the majority of the fast-freight business in the country is being handled by locomotives of modern design, and the method developed by the author will be very helpful in establishing fast-freight tonnage ratings for these locomotives. However, in view of the increased capacity that the experience given above indicates, the writer believes we will in the future obtain still higher tractive-power and horsepower curves for moderately high locomotive speeds.

JOHN E. MUHLFELD.[32] The author's reference to Cole's ratios recalls the writer's association with Mr. F. J. Cole, on many occasions, in connection with the designing of different steam locomotives, including the adaptation of the Mallet articulated type to American practice, for various railroads in the United States and Canada. The writer is heartily in accord with respect to the usefulness of the Cole formulas and that they still hold good in many cases.

Concerning the author's proposed modernization of the Cole ratios for the purpose of meeting present-day requirements of steam locomotive design and operation, the writer, while agreeing that there is a need for this, questions the differentiation based on the so-called types "E" and "A" superheaters. In his conclusion, the author states: "the object of this paper is to suggest a simple method for figuring horsepower and tractive effort for modern locomotives. To this class belong locomotives with type 'E' superheaters, feedwater heaters, and valve motions with about $8\frac{1}{2}$ in. of valve travel. This presupposes that the superheater heating surface assures sufficient superheat, which in locomotives with the type 'E' superheater is about 250 F. . . . For locomotives of older design, with type 'A' superheaters, new constants could be worked out similar to those given in the paper, but it is suggested that for these latter locomotives the Cole formula should be used."

Any number of modern steam locomotives equipped with type "E" superheaters, with from 200 to 250 lb boiler pressure, are not averaging superheated steam of over 600 F, or much over 200 deg of superheat, when working steam. This performance can be duplicated by many of the older locomotives, originally constructed without superheaters, but which since have been equipped with type "A" superheaters and, with 200 lb boiler pressure, can raise total steam temperatures of from 700 to 750 deg, and average from 600 to 650 F, thereby producing as high as 350 deg superheat as a maximum and from 200 to 250 F as an average.

For example, during 1914 some Consolidation freight locomotives, built in 1906, were equipped with the conventional type "A" superheaters, piston-valve steam chests, brick arches, and automatic fire doors. These locomotives have the following characteristics:

Tractive power, lb	44,880
Weight on drivers, loaded, lb	186,000
Size of cylinders, in	22 × 30
Diameter of driving wheels, in	55
Boiler pressure, lb	200
Grate area, sq ft	33.5
Firebox heating surface, sq ft	231
Tube and flue-heating surface, sq ft	2080
Superheater heating surface, sq ft	520
Valves	Piston
Valve motion	Stephenson
Valve travel, in	$5\frac{1}{4}$
Boiler feed	Live-steam injectors

On test runs made with these locomotives over 1.0 per cent ruling grades, which were accompanied by representatives of the railroad, superheater, and arch-brick companies, the superheated-steam temperatures ranged from 612 F to 632 F, with a maximum range of from 700 F to 740 F. When operating over 1.0 per cent grades, both injectors were used to supply the

[32] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

boiler, the water level in the boiler was increased 3 in., the maximum steam pressure was maintained, and the total temperature of the superheated steam ranged from 670 F to 740 F. Indicator diagrams were taken during this performance. During this trip the center row of arch brick melted away while the 740 F temperature of the steam obtained, indicating approximately 350 deg of superheat. This was an unusual performance for a coal-burning locomotive, but is cited to demonstrate what can be accomplished with a locomotive equipped with a type "A" superheater of the description as given, and that superheated steam temperatures obtainable with type "E" superheaters can be readily duplicated with type "A" superheaters.

The particular feature of the type "E" superheater has been its increased superheating surface, which reflects favorably on the boiler capacity, and, in combination with the higher velocity draft, has been set up as the principal advantage over the type "A." However, as compared with the type "A," the type "E" has many operating disadvantages, due to the overheating, swelling, and burning out of the torpedo type of forged return bend; the more restricted gas area through the superheater flues; the stopping up with cinders, ash, soot, and other foreign matter of superheater flues resulting in stuck units; the increased gas velocities through the superheater flues tending to cut out crown-sheet staybolt heads and the beads on the firebox end of the flues; and to a more rapid cutting action of the cinders against the return bends and the element necks, necessitating application of innumerable shields to these parts to prevent such action. All of these conditions mean increased maintenance troubles and expense and of operating inefficiencies and failures, which do not obtain with the type "A" superheaters.

Summing the locomotive superheater situation, it is the writer's opinion that the author's method of calculating tractive power should be used on locomotives having 100 per cent or larger boilers, modern long-travel valve gear, and a sufficient degree of superheat, whether equipped with a type "E" or a type "A" superheater.

The author emphasizes the type "E" superheater as the most outstanding improvement for increasing the power of the steam locomotive, on the basis that it will produce about 250 F or higher superheat. He also considers that boiler evaporation is the controlling factor for locomotive horsepower and that to determine the tractive power it is necessary to refer horsepower to locomotive speed. In the writer's opinion, boiler pressure, grate area, firebox volume, firebox evaporation surface, factor of adhesion, cylinder back pressure, cylinder clearance space, and various other major details are all most important in the development of the modern steam locomotive hauling capacity, or drawbar pull, or drawbar horsepower, and its efficiency and economy in operation.

(1) Gage pressure, lb	(2) Superheat, deg F
200	200
350	200
500	200

Gage pressure, lb per sq in.	Superheat, deg F	Total temp., deg F
200	310	698
400	250	698

The tractive "power" or "force" or "effort" of a steam locomotive is the pressure exerted by the action of the steam against the pistons in the cylinders, through the medium of connecting mechanism, to turn the driving wheels and cause the locomotive to advance. It is measured at the tread of the driving wheels, the internal friction or resistance of the engine being neglected. It is closely related to "hauling capacity," and in practice both are influenced by such factors as weight on and diameter of driving wheels, arrangement of leading and trailing truck wheels, factor of adhesion, steam-generating and superheating capacity, steam pressure and temperature, diameter and stroke of cylinders, mean effective pressure, and uniformity of crank effort, or torque.

The hauling capacity or drawbar pull or drawbar horsepower of a locomotive, which is the real measure of its ability to start and accelerate a train and move tonnage at sustained speed, can only be determined when its tractive power, and the locomotive, grade, and uncompensated curvature resistances which tend to prevent the movement of the locomotive, are known, provided the locomotive has sufficient adhesion to develop this power, without excessive slipping.

With respect to high superheat temperatures, it is questionable as to the practical advantage or economy in superheating steam to higher temperatures than what is required for minimum cylinder condensation, in view of the useful heat that will be exhausted to the atmosphere. To produce 70,000 lb tractive power in a single expansion-cylinder locomotive, carrying 200 lb pressure, with 63-in. driving wheels, would require the admission of live steam into two $28^1/_2$ in. in diameter by 32 in. stroke high-pressure cylinders. At 250 lb pressure the diameter of the cylinders would be about $25^1/_2$ in., at 275 lb pressure about $24^1/_2$ in., and at 300 lb pressure about $23^1/_4$ in. If 500 lb pressure is used, in combination with single-expansion cylinders, it would only be necessary to put live steam into two 18 in. in diameter by 32 in. cylinders, with 63 in. in diameter driving wheels, in order to develop 70,000 lb tractive power, and in which case, with the conventional amount of superheat, from 75 to 100 deg of superheat would be wasted in the exhaust steam.

By increasing the conventional 200 to 250 lb boiler pressure, assuming 200 F superheat as a constant, regardless of the pressure, Table 13 shows what can be accomplished and that only 28.5 additional Btu are required to produce steam at 500 lb as compared with 200 lb.

The ideal condition of the Rankine cycle, as illustrated in Table 14, is also illuminating.

It will be noticed that the 400 lb pressure, 250 deg superheated steam will do about 25 per cent more work per pound of steam than the 200 lb pressure, 310 deg superheated steam. This would result in the use of about 20 per cent less fuel per horsepower hour. The 200 lb pressure, 310 deg superheated steam will still have about 30 deg superheat when exhausted, while the 400 lb pressure, 250 deg superheated steam will be of about 96 per cent quality, saturated, when exhausted.

If locomotive designers and builders in the United States are right in limiting modern locomotive superheated-steam pressures at from 200 to 250 lb, in combination with stayed crown and side sheet fireboxes of relatively small evaporative capacity, then all of the modern central power station and marine boilers, as well as the German State Railway and other European locomotive designers, who are making use of from 350 to 1500 lb and higher superheated-steam pressures, must be wrong.

TABLE 13

(3) Total Btu in steam, per lb	(4) Total Btu in steam rejected from cylinders in exhaust at atmospheric pressure	(5) Btu transformed into useful work in cylinders (3 — 4)	(6) Theoretical efficiency as per Rankine cycle (5 ÷ 3)
1309.7	1083	226.7	17.3%
1325.5	1058	267.5	20.2%
1338.2	1042	296.2	22.1%

TABLE 14

Entropy increase in unavailable energy	Total Btu in steam, per lb	Total Btu in steam rejected in exhaust at 20 lb gage cylinder back pressure	Total Btu transformed into useful work	Efficiency Rankine cycle
1.7066	1364.0	1182.0	182.0	13.3%
1.6318	1365.1	1130.0	225.1	16.6%

The thermal efficiency of the conventional steam locomotive, carrying from 200 to 250 lb boiler pressure, when operated in road service, averages from 4 to 6 per cent at the tender drawbar, in terms of the heat value of the coal as fired. Road service tests of the D. & H. No. 1400 locomotive, which uses 350 lb pressure, show it to have a thermal efficiency at the tender drawbar of 8.75 per cent; the D. & H. No. 1401, with 400 lb pressure, develops 9.3 per cent, and the D. & H. No. 1402, with 500 lb pressure, develops 10.4 per cent. The last locomotive therefore has about double the thermal efficiency of the average conventional steam locomotive, and in the hope of substantially increasing this percentage, the Delaware and Hudson Railroad now has another locomotive under construction at the Schenectady shop of the American Locomotive Company, D. & H. No. 1403, which will carry 500 lb boiler pressure and will be the first triple-expansion type of steam locomotive that has ever been built.

The author has contributed interesting and valuable information concerning calculations for the designing of the modern steam locomotive which will be of great benefit for the guidance of railway and builder's engineers. At the same time we should not overlook the practical operating and maintenance factors, which can only be determined upon by dynamometer-car tests, for the purpose of obtaining actual information relating to indicated horsepower, sustained drawbar pull and drawbar horsepower, fuel and water consumption, boiler and superheater capacity and efficiency, and mechanical and thermal efficiency of the locomotive as a whole—these data to be scientifically utilized to develop empirical formulas.

W. F. Kiesel, Jr.[33] The stated object of the paper is to establish a method of figures for calculation of indicated tractive effort as a function of speed. Throughout the paper references are made to the difficulties of establishing accurate formulas applicable to various types and variations of locomotives, and detail equipment, establishing the need of adopting empirical data.

The author points out that Cole's ratios are no longer applicable to modern locomotives and submits his method of corrections necessary to make the Cole formulas more closely conform to modern power results. Tests of Pennsylvania Railroad locomotives, classes K4s, E6s, K2sa, and L1s, made in 1914 and 1915, and I1s made in 1919, used as examples by the author, are rather too old, as much improvement in locomotive design has been developed since then.

The Cole ratios have been of use in establishing comparisons between locomotives of different design, and it is agreed that they are obsolete. There seems to be no good reason to attempt to resurrect them at this late date, in view of the fact that closer comparisons can be made by other methods of calculation, equally or more simple.

The question of first importance is, "How much steam can any given locomotive make, available for use in the cylinders?" The author is rather vague on this point. He calls attention to boiler tractive effort, as distinguished from rated tractive effort. The latter is dependent on boiler pressure and proportions of cylinders and drivers, and its calculation has been standardized by a simple formula. The designer is mainly interested in maximum boiler tractive effort, which is dependent on maximum evaporation possibilities of the boiler. Conventional fire-tube boilers, with staybolted firebox, vary very little in important proportions.

From various tests and writings, published in the past, it is not far wrong if we assume that the evaporation per square foot of the combustion-space heating surface, compared with that of the flues and superheater (with flue sheets spaced about 20 ft apart) is 6 to 1.

Assuming that equivalent heating surface is the sum of superheater, flue, and six times the combustion-space heating surfaces (steam and water side), it has been developed from tests that the evaporation limit to date is closely $11^1/_2$ lb per hr per sq ft of equivalent heating surface, which permits the empirical assumption that the steam available for use in cylinders may be as high as 10 lb per hr, for each square foot of equivalent heating surface, for the whole range of boiler tractive effort. When test results of a locomotive fall materially short of this, it is advisable to make an investigation to determine reasons for the shortage.

The second phase of this problem is to determine the results in cylinder tractive effort of the use of various amounts of steam in the cylinders up to the available limit. The author determines drawbar horsepower, but it would seem preferable to find the cylinder tractive effort and subtract engine and tender resistance, the result being drawbar pull at rear of tender. Piston speed, or crank speed, suggested in the paper, can be used, but the writer prefers to use speed (V) in miles per hour, because that value is more generally used. He also prefers to determine cylinder tractive effort (T), which can readily be transformed to indicate horsepower, if that figure is desired.

Assume:

Initial pressure, P = 10 lb less than boiler pressure
Engine constant, $C = d^2S/D$
Steam constant, $M = 3W/110w$

d = cylinder diameter, in.
S = piston stroke, in.
D = driver diameter, in.
W = amount of steam used, per hr
w = average weight of steam per cu ft at 100 deg superheat and pressure P.

Since modern locomotives, with few exceptions, use boiler pressure between 200 and 300 lb per sq in., the steam constant M may be written $M = xW$, the values of x being

Boiler pressure	200	225	250	275	300
Coefficient x	0.0716	0.0643	0.0571	0.0532	0.0493

Having determined P, C, and M, the formula for tractive effort becomes

$$T = \frac{2PM}{(M/C) + V}$$

The writer submits that this formula, in use for the last 20 years, though to a certain extent empirical, is based on a rational theory, gives fairly close results, and permits deriving more information, to determine preferential locomotive designs, with less effort than by the use of modified Cole ratios.

For more explicit information as to the theory on which the foregoing is based and for locomotive and tender resistances, see "Principles of Locomotive Operation," by Arthur Julius Wood, McGraw-Hill Book Company.

G. T. Wilson.[34] The writer thoroughly appreciates the fact that the results as obtained are subject to a factor of correction where the performance of the locomotive to which it is applied varies to an appreciable extent from the average results upon which the method of calculation is based.

The application of this boiler-performance calculating method to two of the most modern types of New York Central Lines locomotives, the J-1 4-6-4 and L-2 4-8-2 indicates that the results

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as shown for tractive effort and indicated horsepower in respect to speed are not representative of capacity rating for these locomotives as shown from dynamometer road tests.

It is the practice of the New York Central Lines to rate a locomotive on capacity test results represented by the drawbar pull-speed curve for the respective locomotive or class. The capacity test results represent the maximum sustained drawbar pull and horsepower for all speeds for the respective class of locomotive as governed by the existing boiler ratios, drafting arrangement, valve setting, and cylinder characteristics.

For the last 20 years the capacity results have been used for tonnage rating with success. In the last few years we have made it possible by means of a device applied to the locomotive to provide the engineman with a visible indication of cut-off correlated to speed, so that with full-open throttle the engineman may select a cut-off to produce maximum drawbar pull for that incidental speed. The cut-off indication of this device is based upon the capacity test results for that class of locomotive to which it is applied. The incidental cut-off in terms of speed corresponds to the cut-off used when maximum sustained capacity was developed during dynamometer road tests.

Experience has proved that we may duplicate in every-day performance the actual capacity test results by selection of the same incidental cut-off as used to develop the drawbar pull-speed curve.

Application of the performance test results for boiler evaporation and steam consumption per indicated horsepower in calculating the tractive effort, as proposed in this paper, results in a reduced rating of $2^1/_2$ per cent at driver speed of 50 rpm to 42 per cent at 300 rpm for the J-1 4-6-4 type and $7^1/_2$ per cent at 50 rpm and 17 per cent at 225 rpm for the L-2 4-8-2 type of locomotives.

The pronounced effect of the personal element in operation, drafting arrangement, and wide differences in track profile and operating demands causes a wide variation in both the boiler evaporation and cylinder performance when based upon the results of performance tests.

It is regretted that boiler and cylinder performance are not observed when capacity tests are made, so that more definite results would be available in a study of this kind. For the older designs of locomotives the grate area was the controlling factor in boiler performance. For the performance tests of the J-1 4-6-4 and L-2 4-8-2, the average coal fired per square foot of grate area per hour was 62 and 66 lb, respectively.

For operation of our J-1 class 4-6-4 type locomotives in regular revenue service where maximum capacity has been required for the major portion of continuous service over two divisions, the estimated coal consumption indicated a firing rate of 125 lb of coal per square foot of grate per hour. Our locomotives are not normally worked to maximum capacity in every-day operation except during acceleration and on grades, but conditions often require maximum sustained capacity.

From stationary locomotive boiler evaporation tests we have found that for a boiler of somewhat earlier design and proportionally smaller grate area the evaporation rate was reasonably efficient up to a firing rate of 180 lb of coal per square foot of grate per hour.

Based upon our observations from dynamometer tests and average every-day operation we do not believe that the results for tractive effort and horsepower for a modern design of locomotive may be consistently based upon average performance tests as representative for the tractive effort and power of the locomotive. We contend that the capacity performance of a locomotive is representative of the true characteristics of the boiler and engine because such results eliminate the human variable, common in performance results, and show the true performance as governed by the design of the boiler, feedwater heater, superheater, drafting arrangement, and steam distribution to cylinders.

A. Giesl-Gieslingen.[35] The writer would like to take up Mr. Wilson's remarks dealing with locomotive "capacity" versus "maximum performance." Certainly a railroad is interested in the utmost limit of capacity of a locomotive, since it is of importance on frequent occasions. However, this limit is subject to many influences, the most imponderable of which is probably the boiler capacity, as it is governed by the accidental quality of the coal and the ability of the fireman. These latter influences are of course always felt, whether the engine is working light or hard; but for every locomotive there is a reasonable maximum of performance which can safely be developed whoever may handle it and whatever conditions may be, and this is what the author primarily determines. The boiler is then called upon to work only at a conservative rate.

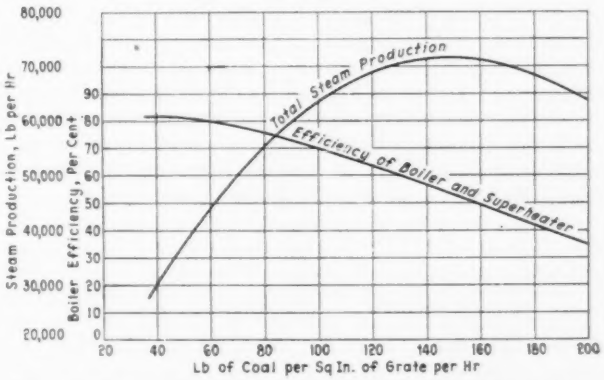


Fig. 27 Boiler Performance of Hudson Type Locomotive

A good idea of what happens when this "safe maximum performance" is being exceeded and the engine is being forced to capacity may be obtained by studying combustion rate and steam production. The writer had an opportunity of observing road tests with the Hudson type locomotives of the New York Central, during which, at times, the capacity limit was being reached. A carefully developed diagram (see Fig. 27) shows the total steam production of the boiler plotted against combustion rate, obtained under the specific conditions of these tests, with a very good fireman and high-grade coal. These engines have a rather large grate area, and therefore a combustion rate of 90 lb produced 60,000 lb of steam, which is about the limit used by the author in his calculation of "maximum performance." The utmost limit of evaporative capacity lay close to 72,000 lb, or about 20 per cent higher, but in order to produce the 20 per cent more steam about 65 per cent more coal had to be fired, bringing the combustion rate up to 150. An interesting point is that a further increase in the firing rate resulted in a diminished output of steam. This is easy to understand if it is considered that, as an extreme example, choking the firebox with coal would kill the steam production altogether; there must be a limit somewhere, but this limit is often being reached at lower rates of combustion than one would assume.

In the neighborhood of the boiler limit, the curve of steam production plotted against combustion rate becomes nearly horizontal, as shown in the diagram, meaning that about the same amount of steam is being generated by widely varying amounts of coal. With the forced operation of present days, this condition is quite often being reached and accounts for much

[35] New York, N. Y.

of the seemingly strange fluctuations in boiler efficiency. The writer has observed cases where the fireman, in an attempt to quickly meet the increasing steam demand during speeding up, unknowingly exceeded that firing rate where the steam generation reaches a maximum; thus instead of producing, say, 66,000 lb of steam with a combustion rate of 123 lb, he did the same job firing 173 lb, or 40 per cent more.

If it is further kept in mind that the absolute value of steaming capacity and the corresponding combustion rate are widely influenced by local conditions, it is apparent that the maximum capacity of a locomotive is something beyond calculation and should be obtained from special capacity tests to suit the specific requirements of the railroad. On the other hand, what the author calls maximum performance, as determined by his interesting and simple method, is a function of locomotive dimensions and can be used efficiently to calculate safe performance figures and to compare various engines with each other.

Finally, a word about train resistance: Like the locomotive formulas, those for the running resistance of cars should be revised to meet modern conditions. The writer has found that the resistance values obtained by the University of Illinois at the beginning of this century are at least 30 per cent and sometimes 50 and 60 per cent too high for our modern freight trains. Here is a very important field for cooperative study, and now that the author has shown how to obtain the tractive power of our locomotives, it would be desirable to determine with equal accuracy what we can do with this tractive power.

W. G. KNIGHT.[36] It is our experience, from carefully observed tests using our dynamometer car, that some changes are necessary in the Cole formula with respect to cylinder horsepower, especially on engines of the super-power type. We have noticed particularly at ranges of speed from 30 to 35 mph that the superpower type of engine produces a higher cylinder horsepower than that provided for in the Cole ratios.

F. E. RUSSELL.[37] The author's analysis of horsepower and tractive effort of steam locomotives is very interesting and enlightening, in view of the fact that he attacks this problem from a different angle, and confirms the accuracy of Cole's evaporating capacity of locomotive boilers. The writer has used Cole's evaporating figures for a number of years and finds them to give very accurate results. The author has endeavored to determine a simplified method for figuring horsepower and tractive effort of modern locomotives. In his endeavor to do so he has introduced "revolutions per minute" in preference to piston speed as the basis of his calculations, and justifies this method by stating that there is not a wide range in crank radius or stroke in modern locomotives, which range from 28 in. to 32 in. stroke.

Prior to 1921, when the Southern Pacific Company introduced a 30 in. stroke in passenger service, the common practice was to use strokes of from 24 in. up to 28 in. Since 1921 long strokes have been adopted for both freight and passenger service in use with superheated locomotives, and some have been placed in service with 34 in. stroke. It must also be borne in mind that there is a wide range in diameter of drivers from 51 in. up to 81 or 84 in. Therefore we should not adopt an average or approximate method, but stick to fundamentals and use piston speed which can be as easily converted into miles per hour as can revolutions per minute, besides being more accurate. The final result is generally desired at a specified speed in miles per hour.

To arrive at the most correct results we must consider the two primary elements separately—the boiler, the unit that generates the steam, and the engine, the unit that converts the heat units in the steam into tractive power and horsepower, either of which can be converted into the other by taking into consideration the time or rate of doing work. Therefore, for most accurate results, we should determine the efficiency and capacity of the boiler to supply the engine, and then consider the capacity and efficiency of the engine itself. In other words, by determining the capacity of the boiler and by using the correct mean effective pressure and pounds of steam per horsepower hour dependent upon the quality and temperature of steam supplied, the maximum horsepower or tractive power can be easily calculated for any speed in miles per hour.

At the time Cole established his ratios and adopted 27 lb. of saturated steam per horsepower-hour and 20.8 lb of superheated steam per horsepower-hour, locomotive superheaters might be considered in their infancy in this country and were mostly of the type "A" construction, developing low superheat as compared to temperatures developed in modern type "A" and "E" superheaters, with the result that steam consumption per horsepower-hour has decreased from 20.8 lb to 19 lb or less per horsepower-hour with type "E" superheaters. Also, both the mean effective pressure and the amount of heat in the steam will vary on different roads and in different territories on the same road, due to character of operation, such as high speed in comparatively level territory or slow speed on heavy grades in mountain territory.

Certain devices applied to locomotives, such as superheaters and feedwater heaters, materially affect the capacity of the locomotive as a whole as well as its efficiency. The superheater, by increasing the temperature of steam, not only has an effect equivalent to increasing boiler capacity, but also increases the mean effective pressure on account of reduction in back pressure; hence it increases both the capacity of the boiler and the engine

The feedwater heater, when of proper capacity, increases the evaporating capacity of the boiler about 10 per cent and also further affects the horsepower or tractive effort by increasing the mean effective pressure due to the reduction of back pressure. Back pressure is also materially affected by the area and shape of exhaust passages and the size of the exhaust nozzles.

Apparently some engineers have spent so much time to prevent wiredrawing of steam on its way to the cylinders that they have paid little attention to the even more important feature of easy curves and large areas for passage of steam from the cylinders to the atmosphere.

It is simpler to determine horsepower and tractive power by means of mean effective pressure based on piston speed, which takes into consideration both stroke and wheel diameter.

L. K. SILLCOX.[38] The various formulas which have been submitted from time to time to assist in the approximate predetermination of locomotive operating characteristics prior to construction have as a rule produced remarkably similar results. These equations assume a variety of forms, depending upon the method of approach to the problem, and each involves certain assumptions with respect to prospective ability. The Cole analysis and the tractive-effort equation have been widely adopted by reason of the simplicity of application, the apparent logical arrangement of factors, and the close agreement which has been observed between road test results and the calculated performance. No marked deviation from the Cole results were ever witnessed so long as locomotives were constructed in general accordance with the proportions upon which the Cole ratios were based. For a number of years, locomotives have been designed

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for highly specialized services; the former standardized ratios have been disregarded, and various improvements, involving both internal modifications and the addition of auxiliaries of proved merit, have been incorporated in the construction to produce higher efficiencies and to adapt modern steam motive power to its required functions.

High speeds, sustained high speeds, sustained high speeds produced economically—these three factors, or the evolution of a single factor, have been largely responsible for the obsolescence of past practices in locomotive design, and when we add to speed the elements of high capacity and the required degree of availability, we observe a locomotive which quickly illustrates the advance which has been made in locomotive building since the Cole ratios were derived. The basis of the Cole locomotive ratios is the cylinder horsepower of the locomotive at or near its maximum power, using this to proportion the firebox area and tube heating surface. Assuming that 120 lb of coal of good quality is about the maximum burned per square foot of grate area per hour, the proportions of the grate can be computed. The grate area being determined, the firebox volume within certain limitations cannot be altered, since it is determined largely by the length of the grate and the diameter of the boiler. The basic factor in boiler efficiency is heat production, and this depends on the fuel fired per cubic foot of firebox volume per unit of time, together with the ratio of firebox surface to firebox volume—in other words, to the cooling influence of the water jacket. In a large firebox the surface/volume ratio is low. The cooling of the flame depends on the water-cooled surface exposed to radiation and convection, and this is less in proportion to the volume and weight of burning gases and solids as the linear dimensions of the firebox are increased. Consequently, in a large firebox, the flame loses less of its heat to the walls, and combustion is more rapid and therefore more complete than in a smaller firebox at the same rate of firing per square foot of grate or per cubic foot of volume. Hence, the usually accepted ratios of grate area to the various heating surfaces are really of little value unless the firebox volume also be known. The boiler can hardly be made too large provided the firebox is also increased. Locomotives with large boilers have a reserve of power and capacity, and are thereby enabled to get over difficult sections of the road or give high sustained capacity without seriously reducing the boiler pressure.

The important function of the locomotive cylinders in converting the heat energy in the steam into mechanical work, and the many modifying influences due to variable conditions under which the work must be performed, together with the limits imposed by the loading gage, are some of the more important points which must have consideration in determining the most efficient arrangement of cylinders when designing a modern high-power locomotive. Since it is governed almost entirely by the steam-producing power of the boiler, the wheel diameter, the speed at which a given load must be hauled over the ruling gradient, and the working steam pressure, cylinder capacity, per se, is generally determined beforehand. The work to be performed therefore becomes a problem involving the weight of the steam available and the efficiency with which the steam can be employed in the locomotive cylinders, the maximum tractive effort accordingly being determined by these and other less important governing factors.

Since the term maximum tractive effort is only an expression involving cylinder volume, wheel diameter, and boiler pressure, it is quite useless as an index of the normal horsepower of a locomotive or the load that a locomotive can haul at any speed above a very low rate. Tractive effort depends solely upon the size of the cylinders and the mean effective pressure, whereas horsepower depends upon both the mean effective pressure and the speed. The so-called maximum tractive effort can only be exerted at exceedingly low speed with nearly full-stroke admission, and in many cases with the feedwater supply shut off in order that the boiler may produce an abnormal amount of steam, which conditions of course cannot be continuously maintained for any serviceable period of time. The utilization of the greatest possible percentage of the heat energy in the steam delivered by the boiler can only be obtained by using the steam expansively, and since there are definite limits to the steaming capacity of the boiler, it will be obvious that the term "maximum tractive effort" may be very misleading indeed. The volume/surface ratio of a locomotive cylinder is equally as important as in the case of the firebox. If the volume in proportion to the metallic surface exposed to the steam is great, the percentage of cylinder condensation and of surface radiation are minimized, although probably the latter is of relatively small moment. For any particular ratio of diameter to length of a hollow cylinder, the volume increases more rapidly than the surface as its linear dimensions are increased, and for any particular cylinder volume the surface is at a minimum and the volume/surface ratio at a maximum when the diameter and length are equal. Such a cylinder is the most efficient from the thermodynamic point of view, and the larger such a cylinder is made the more efficient thermodynamically it becomes.

The author, in arriving at a basis upon which to construct his suggested method of calculation, acknowledges the importance of speed which introduces the time element into all calculations. He would abandon the elements of cylinder size and piston speed as subordinate to boiler evaporation and crank speed. In the first case, he replaces a fixed dimension with a rate of transforming energy. In the second case, both factors involve a time interval, and he defines his reasons for preferring to base his calculations upon crank speed, stating that either method is equally applicable, but since the majority of factors involved bear a fixed relation to and are normally expressed in their relation to crank speed, his final equations are to that extent simplified.

At low rates of firing it has been found that 5 to 10 per cent of the fuel (bituminous coal) and 25 per cent of the oxygen which enters the firebox through the grates are lost through the stack due to imperfect combustion. The principles of complete combustion are simple, although not always easy to carry out, especially in the case of bituminous coal. Combustion comprises the introducing of the coal into the firebox, raising its temperature to the burning point and supplying sufficient air for the combustion of the gases which are given off during this process. It is also necessary that the air be thoroughly mixed with the combustible gases and that these be maintained at a high temperature during the process. With bituminous coal, in some cases, 50 per cent or more of its weight constitutes volatile matter which is driven off when heated. When fresh coal is fired on top of the fuel bed, the gases are expelled at a point where the air supply does not easily reach them, and the result is that a large proportion of them are apt to be carried out of the stack unburned. The introduction of fresh coal tends to shut off the air supply when it is most needed, and, in addition to this, it must be remembered that all coal contains more or less moisture, which in the process of evaporation cools the firebox and thus further interferes with combustion. The next consideration has regard to the production of water gas by the water in the steam uniting with the carbon to form carbon monoxide and hydrogen. This absorbs still more heat from the firebox, which, however, would be given back if sufficient air at a high temperature could be supplied above the fuel bed to burn the gases. It is not possible to do this where the air admitted is too cold; hence a large proportion of these gases passes off unconsumed. After the

moisture has been driven off, distillation of the hydrocarbons begins; part of these are burned, while the remainder, owing to lack of air and low temperature, are wasted. This entire cycle results in the formation of smoke and the waste of combustible gases.

We may inquire into the wisdom of constructing locomotives which will at times be required to operate under conditions of maximum rates of combustion—that is, with an area of evaporating surfaces which taxes the capacity of the grates in order that the heating surfaces may be fully utilized. Under such conditions we know the boiler efficiency is very low. However, if we consider a district made up, for the most part, of a uniform grade, with a maximum grade of short duration, a relatively small grate area is justified and an excess amount of coal is fired with a known loss in utilization for a short interval, and the net efficiency over the entire run may yet be very high. Locomotives are designed with the conditions under which they are to operate fully analyzed, and thus the variation in relative proportions is explained. Such variation limits the application of a single equation which has been employed heretofore in the calculation of expected performance results. Merely cylinder diameter and stroke are of but little service for guidance as to the actual amount of steam a locomotive will require, under different conditions of operation, and such ratios as cylinder volume to grate area, cylinder volume to heating surface, and the like have no meaning whatever in this respect. Further, the futility of such refinements as varying the cylinder diameter by $^1/_4$ in. in order to meet an assumed greater or lesser capacity of a boiler by reason of a few square feet more or less of tube heating surface, will be obvious. The more important features that really do matter, when considering locomotive proportions, may be listed as follows:

(*a*) The speed at which the locomotive will be required to perform the major portion of its work should be retained, if possible, somewhere between these prescribed limits:

Diameter of driver	Revolutions per mile	Revolutions per minute	Speed, miles per hour
63 in.	320.1	120–180	23–57
70 in.	288.1	120–180	42–63
73 in.	276.2	120–180	44–65
79 in.	255.3	120–180	49–71
85 in.	237.3	120–180	51–76

(*b*) The cylinder volume should be sufficient to enable the locomotive to perform its work at an actual cut-off of from 30 to 35 per cent of the stroke, with full boiler pressure in the steam chest, or as near thereto as liberal steam-pipe and throttle area will allow.

(*c*) The provision of ample area in the steam and exhaust ports, with a valve travel as long as possible, in order to avoid wiredrawing, both at admission and exhaust.

(*d*) Smokebox, smokestack, and blast-pipe arrangements to be so designed that a maximum draft efficiency and facility of free release of the exhaust steam is obtained. This is of vital importance at high speeds, when the indicator diagram is necessarily restricted, under which conditions back pressure has a far greater detrimental effect than when the diagram is large at lower speeds.

Much may be said in explanation of the manner in which the mean effective pressure and steam consumption of a locomotive may be expected to vary with changes of speed and cut-off. Unfortunately, the results of such explanations are not amenable to representation in the form of a compact formula; and, indeed, except as a rough guide, curves obtained from the performance of one particular locomotive cannot be relied upon for reliable indicatives relative to the probable performance of another locomotive.

The variation of speed factors with the piston speed for superheated locomotives are shown in Table 15, as derived by Cole in March, 1910, at which time it was found that the average maximum horsepower was reached at 1000 ft piston speed per minute and remained constant at higher speeds. Recent tests with locomotives indicate a distinct rise in percentage of about 10 per cent beyond the figures shown from 1000 ft piston speed and up, and herein lies the advantage which is observed in the locomotive constructed in 1933 compared with 1910, when sustained capacity at speed is to be judged.

TABLE 15

Piston speed, ft per min	Speed factor	Modified speed factor—1910 locomotive (Cole) mod. factor = 0.85	Modified speed factor—1933 locomotive (test) mod. factor = 0.93
0–250	1.000	0.85	0.930
300	0.955	0.812	0.888
500	0.770	0.655	0.716
700	0.605	0.514	0.562
1000	0.445	0.378	0.414
1200	0.371	0.315	0.345
1400	0.318	0.270	0.296
1600	0.278	0.236	0.258

One can readily subscribe to the accuracy of the author's method of approach to the problem of predetermining locomotive characteristics. So long as authentic data are available for guidance in estimating the probable relation between the expected and Cole evaporation, no marked discrepancies should be introduced in the results. Much of the familiar Cole formula is retained. In fact, the effect of the addition of a superheater might be similarly expressed by constructing a curve to indicate the relative performance of a locomotive so equipped with respect to one employing saturated steam. Conversely, the Cole constants may be extended to include factors which take into account the benefits derived by the preheating of feedwater, by the partial elimination of throttling losses effected by long-travel valve gears, and by any improvement which serves to increase the availability of the energy contained in the fuel fired. In like manner, the Cole speed factors may, by the application of proper modifications, be rendered applicable to any change in the relation between boiler and cylinder capacity, anticipating probable change in the shape of the horsepower curve.

Whether the author's method is adopted or the Cole analysis modernized, the results should bear marked similarity. Locomotives must be grouped, in either case, on a basis of relative proportions and the extent to which the latest refinements are incorporated in their design. Both are dependent upon the accumulation of adequate test data for the degree of accuracy experienced. While simplification of calculation is claimed for the author's analysis, familiarity, confidence, and universal acceptance recommend retaining the Cole method in principle. The many tests already catalogued in their agreement with calculated performance using the Cole equations suggest the desirability of continuing the practice so well established in order that a common basis of comparison of the old and new may be preserved. The introduction of any method which represents a departure from an accepted standard will likely be strenuously opposed so long as the familiar process can be revised to produce dependable results.

Ching Pong Pei.[39] It is correctly stated in the paper that no one formula can express the tractive effort of all locomotives. It is equally difficult to express all the factors affecting the development of the tractive effort of the locomotive under the varying operating conditions in any single, simple formula or equation. In other words, a single, simple equation for the calculation of locomotive tractive effort cannot be all-inclusive, and any all-inclusive expression for the locomotive tractive effort cannot be simple in nature or in its application. Such being the

[39] Champaign, Ill. Mem. A.S.M.E.

case, in the formulation of any one simple equation for the tractive effort of all locomotives, one is forced to make a choice in the selection of the more important items and the elimination of the lesser ones.

The author, in arriving at a basis upon which his suggested method of calculation is constructed, used the most direct method of first determining the horsepower output of the locomotive and then obtaining the tractive effort from the well-known equation $P = (T \times V)/375$ or $T = (P \times 375)/V$.

The horsepower output of the locomotive is to be determined by the direct method of dividing the total boiler evaporation by the steam consumption of the locomotive cylinders and the auxiliary devices. The relation between the horsepower and the tractive effort of the locomotive as defined by the latter equation is mathematically exact. Hence the whole problem is reduced to the determination of horsepower, which entails only two factors —the total boiler evaporation and the total steam consumption.

In the determination of the boiler evaporation, the author proposed the equation $E = \beta E_c$. The evaporation coefficient β is made a function of the rotative speed of the locomotive. Strictly speaking, there cannot exist any relation between the speed of the locomotive and the capacity or the output of the locomotive boiler. It is undoubtedly true that the demand on the boiler is necessarily and primarily a function of the rotative speed of the locomotive. However, the boiler output of the locomotive may be more or less than the required amount of steam at the corresponding rotative speeds, for the locomotive boiler and the locomotive engines are, after all, two separate units.

The amount of steam generated by a locomotive boiler depends primarily upon the amount of fuel fired in the firebox and upon the efficiency of the boiler. Due to the falling off of the boiler efficiency with the increase in the firing rate, a point may be reached at which the boiler evaporation is actually diminished in the face of increased firing rate. Such a point is seldom reached if ever at all, in locomotive road performance. The amount of fuel that can be efficiently burned in the firebox depends upon the draft in the front end, which, in turn, is a function of the steam passing through the exhaust nozzle. It is in this respect that the performance of the locomotive boiler is interlocked with the performance of the locomotive engine, and the connecting link between the two is the performance of the locomotive front end.

The justification of correlating the boiler evaporation with the rotative speed of the locomotive, as expressed by the formula $E = \beta E_c$ is, as the writer sees it, premised on these assumptions:

1 The Cole evaporation figures E, as derived from the performance of the locomotives of the pre-war period, still hold good on modern locomotives.

2 The draft efficiency is uniform on all locomotives, being only a function of the amount of steam passing through the exhaust nozzle, which implies that the same amount of steam is exhausted from the cylinders at a certain rotative speed of all locomotives.

3 The firing rate in the firebox and the boiler efficiency are each only a function of the draft produced in the front end.

There is no way to prove or disprove the validity of the first assumption, since no mention has ever been made about the firing rate in connection with the Cole evaporation figures. Likewise, the author avoided specifically stating the firing rate or the boiler efficiency at which the evaporation figures were derived. There is no doubt that at some firing rate an evaporation figure as shown by the Cole figure can be obtained. It is equally true, at some other firing rates, that the evaporation rate can be either higher or lower than the Cole figures. In other words, the original Cole evaporation figures are indefinite and ambiguous, and the question naturally arises as to the propriety of using those figures as the basis for the estimate of boiler evaporation. In the derivation of boiler-evaporation figures, the author stated that the Cole evaporation figures still hold good on modern locomotives. Whatever significance may be attached to the Cole evaporation figures, it is clear from the foregoing statement, according to the author, that there has been no material improvement in the locomotive boiler performance, with the exception of the increase in boiler capacity made possible by the addition of feedwater heater, during the last 15 or 20 years. Does this not offer a challenge to the railroad mechanical department in general and the locomotive designer in particular?

It is rather difficult to subscribe to the second assumption in that the same amount of steam is always exhausted from the cylinders at a certain rotative speed of the locomotive. This would be true only when the locomotive is operated with a definite and exacting relationship between the cut-off and the rotative speed. In a general way, the locomotive is operated with longer cut offs at low speeds and with shorter cut-offs at high speeds, and this is as far as the relation between the cut-off and the rotative speed goes. Hence, this assumption cannot hold true for any one locomotive, let alone for all locomotives.

Whether or not the intermittent character of the locomotive exhausts has any effect on the draft produced in the front end, there must exist a certain definite relationship between the amount of steam passing through the exhaust nozzle and the amount of air admitted into the firebox. We may choose any arbitrary ratio, such as the pounds of exhaust steam per pound of air admitted into the firebox, as the draft efficiency. In this case, it can be readily seen that the design of the front end as well as the design of the ashpan, grates, firebox, and the boiler itself exert a large influence on the draft efficiency. In spite of the similarity of the modern locomotives, we know from experience that the draft efficiency is not uniform on all locomotives, even if it were possible to always exhaust the same amount of steam from the cylinders of all locomotives at the corresponding rotative speeds.

From the test results of locomotives, both in road service and on testing plants, we find that neither the firing rate nor the boiler efficiency is a function of the draft in the front end only. In locomotive road operation, it is perfectly clear there is no such thing as a constant firing rate, as the amount of fuel burned in the firebox will have to depend upon the load and the speed of the train and the profile of the road. Within the operating range of a locomotive, it may be generally stated that the tractive effort of the locomotive increases with the increase in the amount of fuel burned, though not necessarily in the direct proportion. There is no denying the fact that one of the major factors, if not the major factor, affecting the boiler evaporation of the locomotive is the firing rate. It is exceedingly difficult to follow the logic of any estimate of the boiler evaporation without due regard to the amount of fuel burned.

The steam consumption of the locomotive cylinders is naturally mainly a function of the percentage of cut-off at which the engine is operated. It is also a function of the pressure and the temperature of the steam, both at the steam chest and at the point of cut-off. Test results of locomotives on testing plants seem to indicate that there is always a narrow range of cut-offs to be operated with a certain rotative speed, the combination of which results in the minimum steam consumption, other conditions remaining unchanged. On the other hand, if we were interested only in the maximum capacity of the locomotive, it may be operated with a much longer cut-off than that at which the minimum steam consumption is obtained, limited only by the capacity of the boiler to supply steam. At any stated boiler capacity, which is mainly a function of the amount of fuel burned, the locomotive can be operated with a number of different cut-offs at one rotative speed, in which case the locomotive would develop

different values of tractive effort, according to the cut-off, at the same rotative speed. It is therefore not sufficient to merely correlate the tractive effort and speed, as represented by the ordinary tractive-effort-speed curves, without specifying the manner in which the particular tractive effort figure is obtained.

As previously pointed out, it is not possible to express the tractive effort of locomotive by any single, simple formula, however desirable it may be, and it is also almost meaningless to merely state that the locomotive would develop so many pounds of tractive effort at a certain speed without specific qualifications. In view of these difficulties, would it not be more logical to present the locomotive tractive-effort data in the form of a series of tractive effort-speed curves, each one representing a certain firing rate and each point on a curve representing the tractive effort obtained with the most suitable cut-off at the corresponding rotative speed? It thus presents a complete picture of the locomotive tractive effort to the railroad operating department, enabling them to determine in which way the locomotive can be most efficiently operated to meet the individual need of the road, whether in the obtainment of the maximum capacity of the locomotive at the expense of economy or in the selection of a most economical performance with some sacrifice in the locomotive capacity.

S. S. Riegel.[40] We worked out the horsepower and tractive-effort curves for our most recently acquired fast-freight locomotives by both Mr. Lipetz's simplified and the Cole method, and in our dynamometer tests, when the engines were first received, we developed horsepower and tractive-effort curves in harmony with the results secured by the Lipetz formula. We naturally are satisfied with the correctness of the deductions as stated in the paper.

Author's Closure

It is very gratifying to see that a paper on locomotive ratios evoked so much discussion, not only on the subject to which the paper was directly devoted, but also on other related subjects. The majority of these latter comments does not call for a reply, as no criticism was voiced by these discussions with respect to the proposed method of figuring horsepower and tractive effort of steam locomotives. I shall therefore restrict myself principally to discussions on the method itself, and later, in the conclusion, I will briefly reply to all other remarks. A brief recapitulation of the paper would be of use, as this may in part answer some questions of the discussers. At the beginning of the paper it was pointed out that there are two possible methods of evaluating the horsepower and tractive effort of a locomotive. The power of the locomotive is derived from steam through the means of the cylinders. In the first method, by using mathematical analysis, it would be quite possible to follow through the various processes taking place in the cylinders for a certain amount of steam of known quality (pressure and temperature). This analysis would be very intricate, as a great number of complex factors has to be taken into consideration. Furthermore, as such an analysis would represent a consecutive chain of premises and sequences, the final result would depend upon the validity of each individual premise; and there would be no assurance that the method would lead to reliable results.

In order to eliminate all these difficulties and offer a practical quick method for evaluating the horsepower of a locomotive, it was suggested in the paper to apply an empirical method, by which the power is figured on the basis of the total evaporation of the locomotive boiler and the approximate steam rate per horsepower-hour at various speeds. Thus only two variable factors are introduced in the calculation instead of a multitude of factors, and in addition both factors are such that they can be figured fairly accurately. The evaporation of a locomotive boiler is more or less known. It has been measured and evaluated hundreds of times, and the Cole figures seem to represent very closely the amount of steam which can be generated by a locomotive under normal conditions without excessively forcing the boiler. As to the steam rate, it is also known that a well-proportioned and properly designed locomotive has fairly definite limits for steam consumption per horsepower-hour for various conditions of work. It would seem, therefore, that a method by which these two variables are tied together should give more accurate results than complex formulas.

Second, it must be borne in mind that the objective of the method is not to predict with accuracy the power-and-tractive-effort curve of any locomotive of any design. As Mr. G. W. Armstrong puts it rightly, the object is to provide a "comparometer" for evaluating locomotives of known designs. One or another locomotive having an unusually large superheater, or firebox, or combustion chamber, or cylinders, or a special draft arrangement, or some other novelties which are being tried out from time to time, can not be evaluated by the suggested method. Only the average well-proportioned and properly designed locomotive can be served by this method with a reasonable degree of accuracy, and the locomotive with the enumerated or other improvements has to be evaluated by comparing its performance with the predictions of the formula.

Third, as it has been already pointed out in the paper, there are great many kinds of tractive effort which are of interest and are being considered in different cases. It is therefore always possible to find fault with one or the other method of figuring tractive efforts and to point out that some tests show higher or lower figures than those obtained by the method. Tractive effort is not a definite thing, and it is not the *maximum* tractive effort which is being sought. It has been explained in the paper that the method permits to determine the *performance* curve which is usually obtained under normal conditions of work; and I should like to add here—and with a reasonable degree of efficiency. This will be explained later.

As regards the paper itself (not the method), it should be remembered that the theories and various statements made in the paper which may be open to criticism have no influence on the method in the final recommended form. Neither the theory of the variation of evaporation with speed, nor the evaluation of coefficient β, nor the actual steam rate as obtained from tests, has any bearing on the final results. These theories and statements had been brought out in order to make clearer the way by which the method was developed. After formulas [12] and [13] and Table 7 have been established, they were checked for all modern locomotives for which test data were available, and the results were shown in Figs. 10 to 15 of the paper. The preliminary theories and statements were thus no more than stepping stones, which can be removed after the final formulas and empirical constants had been established and verified, and the criticism of these theories, by purely theoretical considerations, cannot undermine the method itself. The author will not try to defend them as such, unless it is shown that they affect the method.

As a whole, all discussers agreed that cylinder dimensions alone cannot serve as a basis for figuring tractive efforts. No one disputes the fact that the tractive effort, especially for higher speeds, is determined by the boiler capacity. Dr. R. Eksergian, using more or less the same premises as the author's, tries, however, to point out the way by which an analytical method may lead to final results. I do not dispute his way of reasoning, but whether a practicable method can be worked out by following his lines, remains to be seen. Only Messrs. Kiesel and Vincent

[40] Mechanical Engineer, Delaware, Lackawanna & Western Railroad Company, Scranton, Pa.

dispute the correctness of figuring boiler tractive-effort curves on the basis of the boiler only, and devise formulas in which both the cylinders and boiler evaporation are included.[41] Mr. Kiesel made several general remarks and offered his well-known formula to which I will return later, but Mr. Vincent presented a very complete theory of an analytical method which is very interesting, and tried to prove by charts that the method recommended in the paper does not give in application to locomotives, even those which were considered in the paper, as good results as his method. In the case of the locomotives cited in the paper, Mr. Vincent made errors in calculating tractive efforts in accordance with the method submitted in the paper. In other cases, especially in his statements, he made several mistakes, which the author will try to correct. He will therefore start his replies with Mr. Vincent's criticism.

Formula [17], which is based on Mr. Kiesel's theory and is a modification of his formula, is actually a simplified form of Professor Strahl's method explained on the third page of the paper. Mr. Vincent neglects a great many things as compared with Strahl. He pays no attention to the cylinder clearance volume, to the release and compression volumes, to the exhaust pressure, to the distortion of indicator cards on account of speed, to the heat exchange between steam and cylinders, and to many other things. He further assumes constant evaporation through the whole range of boiler tractive effort. The only variable which he takes into account is the variation of cut-off with speed and the resulting change in expansion.

As it is clear from the derivation of formula [17], it has a meaning only if the second member in the denominator of the right-hand side, designated by E, is actually the expansion ratio, and $1.95P_1/1 + E$ is the mean effective pressure in the cylinders (mean indicated pressure), equivalent to p_i of formula [1] of the paper. Prof. A. J. Wood called E the "expansion ratio" and figured it by dividing the weight of a cylinderful of steam to steam per stroke, exactly as Mr. Vincent defines E. Further, Professor Wood assumed that the mean effective pressure is approximately equal to $2P_1/1 + E$,[42] very close to $1.95P_1/1 + E$ as Mr. Vincent assumes. Thus E for all intents and purposes is the expansion ratio, differing from the real expansion ratio only by the neglect of clearance and release volumes and the effect of cylinder cooling; in other words, it is the expansion ratio approximately estimated. Likewise, the mean effective pressure is roughly expressed by a simple formula $1.95P_1/1 + E$, as the real formula is much more complicated even if the distortion of indicator cards due to speed is not taken into consideration.

Because of this distortion effect, no formula can express the mean indicated pressure. Tests with the I1s locomotive on the Altoona testing plant (Pennsylvania Bulletin No. 32) give results (for representative indicator cards with full throttle) which are compared in Table 16 with values according to the foregoing formula.

TABLE 16

Test No.	Rev. per min	Cut-off	Mean eff. pressure, lb per sq in.	E (inverse of cut-off)	$\frac{1.95\ P_1}{1\ +\ E}$
5928.....	40	0.30	118.3	3.33	108.0
5932.....	40	0.40	142.6	2.50	133.7
5940.....	40	0.55	186.0	1.81	166.5
5926.....	80	0.30	98.6	3.33	108.0
5927.....	80	0.50	150.1	2.00	156.0
5933.....	80	0.55	163.1	1.81	166.5
5915.....	120	0.30	79.7	3.33	108.0
5917.....	120	0.50	124.7	2.00	156.0
5929.....	120	0.55	126.2	1.81	166.5
5921.....	160	0.30	68.3	3.33	108.0
5937.....	160	0.47	104.9	2.13	149.4
5938.....	160	0.50	101.0	2.00	156.0

[41] Mr. L. K. Sillcox, while not disputing the correctness of the author's method, prefers for practical considerations the Cole method. The author presents his reply to Mr. Sillcox at the end of his closure.

[42] *Railway Mechanical Engineer*, Dec., 1916, p. 627.

Thus the discrepancy between the actual and estimated mean effective pressure, fourth and sixth columns, especially at higher speeds, is enormous; the actual at 160 rpm is 36.7 per cent lower than the estimated, while at 40 rpm the actual is 12 per cent higher than according to the formula. It is true that E is not exactly the inverse of the cut-off, but whatever method is chosen for figuring the E, it will not make the foregoing comparison much better, as for the same cut-offs the mean effective pressures vary considerably with the speed.

It is inconceivable why v in formula [17] should correspond to the average pressure during *admission* with only 100 F of superheat as an average for the whole cylinder (including expansion ?), unless it is an empirical correction of errors caused by making the other assumptions. Nor is it easily understood why such an important factor as cylinder back pressure is not taken into account at all, and why auxiliaries, which on a road locomotive absorb in some cases as much as 10 per cent of the total steam, are disregarded altogether. In view of all these approximations, which affect the ultimate result in opposite directions, the formula may accidentally give sometimes fairly good tractive-effort figures for certain ranges of speed and boiler forcing.

Considering now more specifically Mr. Vincent's curves, which should prove that the recommended method is not applicable to some of the locomotives cited by me as examples, I will take up first his contention that at low speeds the cylinder tractive effort (TE_1) should not be represented by a straight horizontal line, but by an inclined line. In Fig. 1 of the paper, I have shown cylinder tractive efforts at different cut-offs. They actually represent curves obtained from tests with the New York Central 4–8–2 locomotive. The curves have a sloping shape, and up to a certain speed can be assumed to be straight inclined lines. The rated tractive effort of this locomotive, which has 85 per cent maximum cut-off, is, with 0.85 mean indicated pressure factor, 60,620 lb. It can be seen from Fig. 1 that the top curve for 85 per cent cut-off lies above the rated tractive effort up to a speed of about 12 mph, which is slightly more than the speed of the intersection point of "1932" curve with the horizontal rated tractive effort line on Fig. 10 of the paper. We can thus afford to draw the rated tractive effort with 0.85 factor as a horizontal line, but we could also draw an inclined line, starting it from a higher point, as long as this point is below R on Fig. 1 and the inclination is properly chosen. It seems to be simpler to draw the rated tractive effort as a horizontal line and extend it to the intersection with the boiler tractive effort, as this will be less, by a certain margin, than all cylinder tractive points at low speeds obtained at 85 per cent cut-off. In other words, if we estimate the rated tractive effort with the usual factor 0.85, which leaves a certain margin for the peak tractive-effort points, the line can be horizontal, but if, as it is sometimes being done, a factor of 0.90, or even 0.92 to 0.93, is assumed, it is necessary to draw an inclined line.

The foregoing figures are true for long cut-off locomotives. It would be appropriate to state here what the conditions would be for locomotives with limited cut-offs. In this case the factor in formula [1] should be taken in accordance with the limitation of the cut-off as shown in Table 17.

TABLE 17

Cut-off, per cent	Mean indicated pressure ratio(α)[43]
90	0.86
85	0.85
80	0.84
75	0.83
70	0.82
65	0.80
60	0.78
55	0.75
50	0.71

[43] α in formula [1] of the paper.

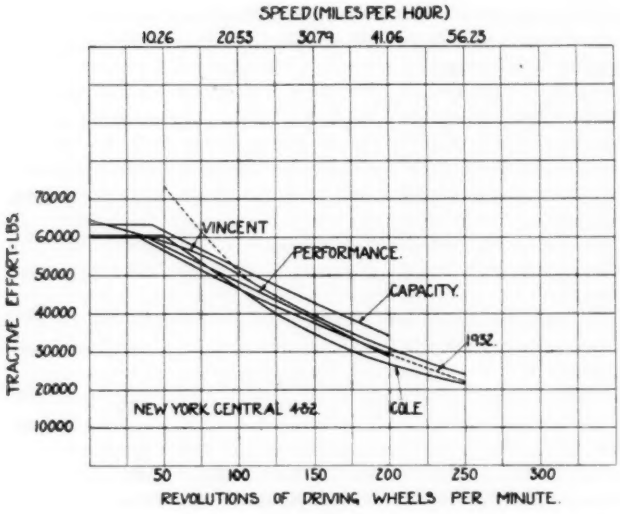


FIG. 10-A

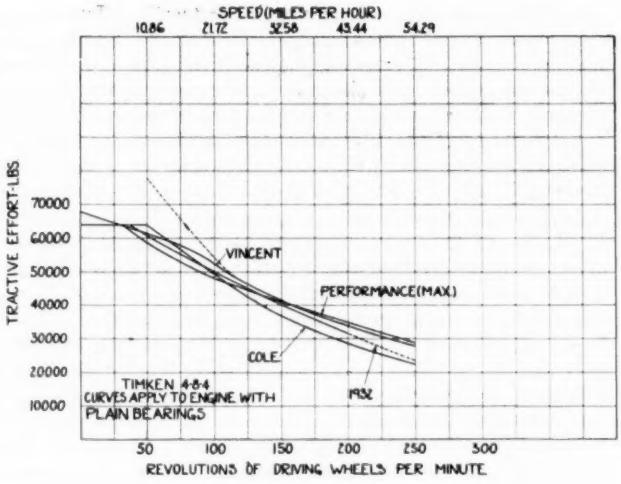


FIG. 13-A

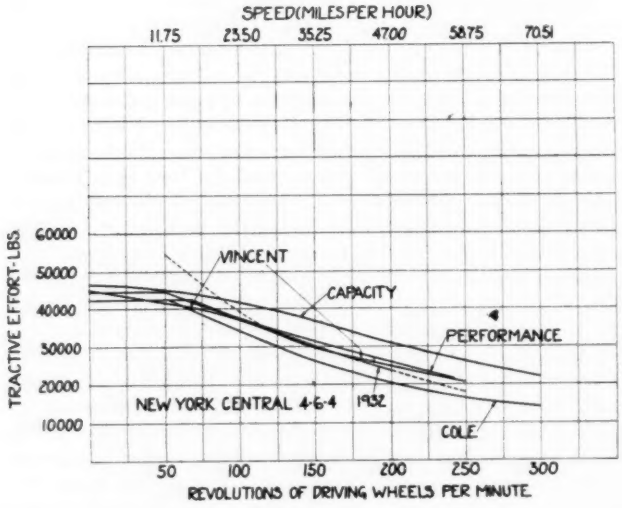


FIG. 11-A

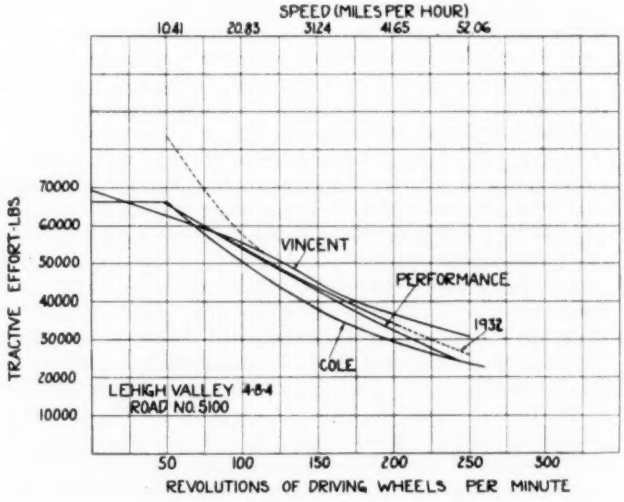


FIG. 14-A

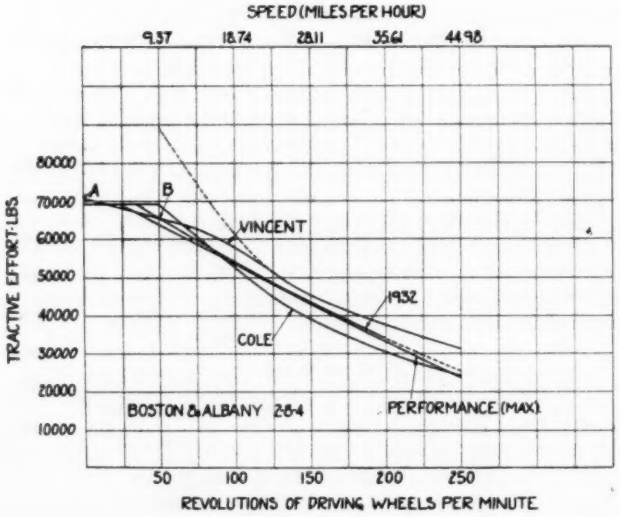


FIG. 12-A

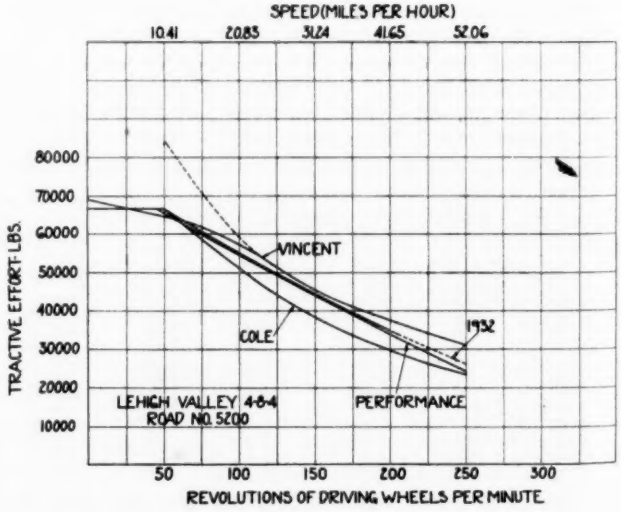


FIG. 15-A

These figures represent more closely the actual maximum values for the mean indicated pressure ratios, and therefore it would seem that there is more justification in this case for plotting the rated tractive effort as an inclined line. However, as every limited cut-off locomotive has either auxiliary ports, or some other arrangements for starting, the straight line may still be applied because these ports will permit an additional margin above the real mean indicated pressure. So, for instance, in Fig. 12 the rated tractive effort is represented by a straight horizontal line drawn at a height of 69,600 lb, corresponding to a factor of 0.78. The Boston & Albany locomotive has a 60 per cent limited cut-off. This is a reply to one of Mr. Vincent's remarks.

Moreover, in a well-proportioned locomotive, either long or limited cut-off, the rated tractive effort corresponds to the limit of adhesion, and the straight line represents not only the rated tractive effort, but also the adhesion limit. It is for this reason that I prefer to put it as a straight horizontal line. When Mr. Vincent uses the inclined line, beginning at 92 per cent mean indicated pressure ratio, part of his cylinder tractive-effort line may very often be above the adhesion line, as can be seen from the examples of the locomotives which are shown later with Mr. Vincent's curves.

Leaving now for a moment Mr. Vincent's transition curve (TE_2) and going over to his main curve (TE_3), it can be seen from Figs. 10-A to 15-A, which represent Figs. 10 to 15 of the paper with Mr. Vincent's tractive-effort curves added and horsepower curves eliminated, that his curves are, in all cases except one, higher than the "1932" curves and do not, when applied to these locomotives, represent the test results any better. This also applies to Fig. 12-A, which shows the Boston & Albany limited cut-off locomotive, as to all other examples which are long cut-off locomotives, except New York Central 4–6–4 (Fig. 11-A), for which it is slightly lower for speeds below 25 mph.

The reason for the author's replotting these curves is that Mr. Vincent made an error in figuring the "1932" values for his Figs. 22, 23, and 24, which are supposed to show the same curves as Figs. 11-A, 12-A, and 13-A. Although an example was given in the paper showing how the "1932" method should be applied, and the tractive efforts for the New York Central 4–6–4 locomotive in that example were calculated, Mr. Vincent made the error of giving credit for the feedwater heater twice: first, by adding 7 per cent to the Cole evaporation figure (E_c), and second, by using from Table 7 those factors which are supposed to be used for locomotives with feedwater heaters and the Cole evaporation figure (E_c) without any additions. However, this led Mr. Vincent to state, referring to his replotted curves for the three locomotives, Figs. 22 to 24, including the New York Central 4–6–4: "It will be observed that they are not the same as the '1932' curves, shown in the author's Figs. 11 to 13. He has evidently used constants for locomotives without feedwater heaters, although the records indicate that heaters were used." The discrepancy is due to Mr. Vincent's use of 1.07 E_c instead of E_c, together with factors for feedwater heaters shown in the second and third lines of Table 7. The right procedure calls for the use of E_c in all cases—this in accordance with formula [13]—and for different factors for locomotives with or without feedwater heaters, as the case may be, in accordance with Table 7. The curves of Figs. 11 to 13 were found to be correct, and therefore they have been replotted on Figs. 11-A to 13-A. Mr. Vincent's curves have been recalculated and added to the charts. In addition, new charts have been also made for the remainder of the locomotives cited in the paper, Figs. 10-A and 14-A to 15-A, which show Mr. Vincent's curves and the "1932" curves correctly.

Mr. Vincent's curves, although practically in all cases higher than those of the author, follow the same trend as the "1932" curves for speeds above 20 to 25 miles, below which Mr. Vincent's curves have to be replaced by an arbitrary transition curve. If his curves were to follow his formula below these speeds, they would be as shown by the broken line on Figs. 10-A to 15-A, and much above either the "1932" curve or test results. It is only because they are cut off and replaced by an arbitrary curve, which is made to bend down so as to meet the low-speed line (TE_1), that Mr. Vincent's curves represent the results more or less correctly; in other words, although his formula [17] depends both on cylinders (machine part of the locomotive) and evaporation, with a predominating influence of cylinders for low speeds and evaporation for high speeds, his formula is actually recommended by him only for high speeds, at which the boiler and not the cylinders are of importance. Thus, at speeds from 0 to about 10 mph, Mr. Vincent uses a chart of inclined straight lines, depending upon the size of the machine part only—this is at least logical in principle. From 10 to about 20 to 25 mph, he makes use of his transition curves, with no formula, which is rather arbitrary, and only for speeds higher than 20 to 25 mph he offers a formula, depending on sizes of both machinery and boiler, whereas in this case only the boiler is of importance. This is not logical even in principle. Moreover, the rational, mathematically derived formula turns out to have a limited application. It could not be otherwise, since so many essential factors have been neglected without even attempting to compensate for them, except by introducing various unjustified approximations.

The reason that Mr. Vincent's figures, nevertheless, come so close to those of the author at speeds above 20 to 25 mph, and therefore represent more or less correctly test results, is that at those speeds the formula is actually of the same form as formula [13] of the paper, as can be easily shown by following mathematical manipulations. Mr. Vincent's formula [17] is:

$$TE_3 = \frac{1.95\ P_1 M}{1 + 36.66\ MV/Ev} \qquad \text{[17-A]}$$

if the paper's notation for speed (V) and evaporation (E) are used. For locomotives with feedwater heaters, Mr. Vincent assumes $E = 1.07\ E_c$. Consequently,

$$TE_3 = \frac{1.95\ P_1}{1/M + 36.66\ V/1.07\ E_c v} = \frac{1.95\ P_1 E_c v}{E_c v/M + 34.25\ V}$$

If steam were a perfect gas, P_1v would be constant at constant temperature, in accordance with the law of Boyle. For superheated steam varying between narrow limits of pressures and temperatures, P_1v is approximately constant; namely, for boiler pressures 215 to 265 lb (gage) per sq in. and 100 deg superheat, the figures are as given in Table 18, and the product P_1v varies

TABLE 18

P_1 in lb per sq in. (gage)	T_1 in deg F	In cu ft per lb	P_1v
215	493.7	2.328	500.5
240	502.7	2.111	506.6
265	511.1	1.931	511.7
Average			506.3

very little. Substituting in the formula the average figure 506.3 for P_1v in the numerator, and the value of V from formula [3] of the paper for V in the denominator, we obtain:

$$TE_3 = \frac{1.95 \times 506.3\ E_c \times 336.134}{E_c v\ 336.134/M + 34.25 \times nD} = \frac{1.95 \times 506.3 \times 336.134/34.25}{336.134\ E_c v/34.25\ M + nD} E_c,$$

or, carrying out the arithmetical calculations,

$$TE_3 = \frac{9689.3}{9.814\ E_c v/M + nD} \times E_c \ldots\ldots [17\text{-B}]$$

Formula [17-B], in view of its derivation as given above, is thus identical with Mr. Vincent's formula [17]. Comparing it with formula [13] of the paper and neglecting the difference in constants, it will be seen that the structure of the formulas in both cases is very similar: the tractive effort varies directly with evaporation E_c and inversely with diameter D, with this difference, however, that instead of M_t/D of formula [13], a more complicated factor for E_c must be used in order to make their product equal the tractive effort, namely:

$$\frac{M_t'}{D} = \frac{9689.3}{9.814\ E_c v/M + nD} \ldots\ldots [19]$$

In this expression M_t' has been introduced as a modulus similar to modulus M_t of formula [15]. Comparing it with [15], it will be seen that both M_t and M_t' vary inversely with speed n, although the law of variation is different in view of the presence of the first member of the denominator in formula [19], which member is missing in the denominator of formula [15]. The first member introduces the influence of the machinery in the modulus, and as at low speeds its value is proportionately larger than that of the second member of the denominator, dependence of tractive effort on the evaporation is not so direct as in the method proposed in the paper. The effect of this member is reflected in the broken portions of the Vincent curves on Figs. 10-A to 15-A. The effect of the cylinders (M) is too great. At high speeds, on the other hand, the second member of the denominator gains in value, and the influence of the machinery on the results of formula [17-B] diminishes, while that of the boiler increases. Therefore, the Vincent formula comes closer at high speeds to the formula of the author than at low speeds.

As to Vincent's transition curve, the reason for its existence is not clear. If a locomotive had an unlimited boiler supply, there would be no boiler tractive effort, and the limitation would be by the maximum cylinder tractive effort, which would be the only one to consider. If, on the other hand, the boiler capacity would be limited, but the cylinder dimensions were so large that the decrease of the cut-off would have to begin almost from the start, the boiler tractive effort would be the only one to consider practically through the whole range of speeds. If, as it actually happens in real locomotives, both the cylinder dimensions and boiler capacity are limited, the tractive effort must consist of two intersecting curves, as it can be only either one or the other. Therefore, there should be no justification for a transition curve, although I can see a reason for a transition curve in the case of the Vincent formula, because at intermediate speeds it gives exaggerated results for all locomotives (broken lines on all previous figures) which have to be corrected by a curve of some sort interposed between two other curves (TE_1 and TE_3), which are approximately correct—one for the reason that it corresponds to the limitation of the cylinders, which is more or less simple and definite, and the other for the reason that it comes close to the method suggested by the author.

In addition to the curves which have been discussed, Mr. Vincent also shows in Figs. 19 to 21 the application of his and the author's methods to three other locomotives. Fig. 20 refers to the Pennsylvania I1s, which is a locomotive with a very limited cut-off (maximum 55 per cent).[44] Mr. Vincent shows that at low speeds test figures are above the boiler tractive-effort curve based on "1932" factors. He explains it by the fact that the author's formulas have been derived from data of full cut-off locomotives, which at low speeds, when long cut-offs are used, show higher steam consumption than locomotives with shorter cut-offs. In his opinion, for this reason higher powers and tractive efforts can be obtained from the I1s locomotive than the author's formulas would indicate. In substance it should be so for very limited cut-off locomotives, but Mr. Vincent's explanation does not prove the point in so far as the application of the method to locomotives, or even to limited cut-off locomotives, is concerned.

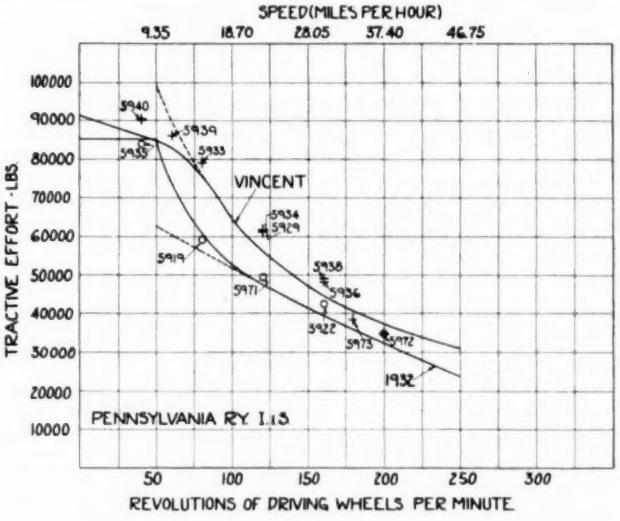


FIG. 28

In order to make a better comparison, the author replotted the "1932" curve of Fig. 20, as Mr. Vincent's plotting was not quite accurate. It is shown on Fig. 28 with Mr. Vincent's curve. He also repeated the most representative test points shown on Fig. 20 by crosses. The numbers which accompany the crosses are test numbers as they appear in the Pennsylvania Bulletin No. 32, in which the test results of the I1s locomotive were published. Particulars about these tests are given in Table 19.

TABLE 19

Test No.	Rev. per min	Indicated tract. eff., lb	Dry coal per sq ft per hr, lb	Steam per ihp-hr, lb	Boiler eff., per cent	Overall locomotive eff., per cent
	Bulletin's column No.					
	14	49B	33D	46	41	52
5940.....	40	90,257	79.9	21.2	56	4.7
5939.....	60	86,072	103.0	18.6	55	5.1
5933.....	80	79,135	127.1	17.3	54	5.6
5929.....	120	61,219	214.0	18.3	42	4.0
5938.....	160	48,978	204.8	17.0	41	3.6
5936.....	160	48,121	214.7	17.4	39	3.8
5973.....	180	38,417	228.5	18.0	35	3.1
5972.....	200	34,827	158.0	16.0	49	4.4

It is doubtful that on account of the lower steam consumption of the locomotive the test figures are higher than what the "1932" curve would indicate. By analyzing the figures from the table, it can be seen that all the tests (except one, No. 5935, discussed below), which, according to Mr. Vincent, gave tractive efforts much above those recommended by the author, especially those at speeds below 120 rpm, have been obtained at steam consumptions from 17.3 to 21.2 lb per ihp. If steam consumed by auxiliaries on a road locomotive, which are not included in the Pennsylvania test results, are added, fairly high steam consumptions (up to 23 lb per hp-hr) will be obtained for these speeds, which are

[44] Maximum tractive effort corresponding to this cut-off is 90,024 lb, factor 0.75. On Fig. 28 rated tractive effort of 85,222 lb, corresponding to a usually cited cut-off of 50 per cent, is shown (factor 0.71). The higher figure is correct.

not far from those assumed by the author in his Figs. 5 and 6. The same applies to other speeds with the exception of test No. 5972, but this test point is below Mr. Vincent's curve. In the opinion of the author, the real reason for the test points lying above his curve is that all these points have been obtained by excessively forcing the boiler. In order to get 86,072 lb tractive effort at 60 rpm (test No. 5939), it was necessary to burn 103 lb of coal per sq ft of grate area per hour, which, at this low speed, is rather high. Likewise, in order to get the results of tests Nos. 5936 and 5938 at 160 rpm, it was necessary to burn 214.7 lb of coal per sq ft per hr, which is enormous even if the fact of the small grate of the locomotive is considered—after all, the ratio of the outside evaporative heating surface to grate area is only 68.4.

A good indication of forcing a boiler and a locomotive are the boiler and overall efficiencies shown in the last two columns of Table 19. They are very low—the latter varies from 3.1 to 5.6 per cent, and more often it is closer to the lower limit than to the upper. In order to make this point clearer, test points with results which come closer to "1932" curves are shown as circles on Fig. 28. Table 20 gives some data of these tests.

TABLE 20

Test No.	Rev. per min	Indicated tract. eff., lb	Dry coal per sq ft per hr, lb	Steam per ihp-hr, lb	Boiler eff., per cent	Overall locomotive eff., per cent
	Bulletin's colomn No.					
	14	49B	33D	46	41	52
5935.....	40	83,980	56.7	19.2	67	6.4
5919.....	80	59,188	70	15.9	65	7.3
5971.....	120	49,290	101	15.6	59	7.1
5922.....	160	42,261	132	15.4	54	6.2
5972.....	200	34,827	158	16.0	49	4.4

It will be seen from this table that the boiler was forced considerably less than in tests of Table 19, with the result that the efficiency was much higher—6.2 to 7.3 per cent, except only one case when it was 4.4 per cent.

Point 5935 is very interesting and is put on Fig. 28 on purpose—it is removed far from the boiler tractive-effort curve "1932," and at the same time the forcing of the boiler does not seem to have been excessive. This (probably incidental) point would bear out Mr. Vincent's contention, if there were at least one more point like it in this, or a similar point in another, limited cut-off locomotive. Then it would be necessary to revise the new moduli for the speed of 50 rpm for the 55 per cent limited cut-off locomotives. It would be premature to do it now, but when this is done, the method itself will not be affected, if for locomotives with 55 per cent cut-off a new modulus for $n = 50$ is established. The boiler tractive-effort curve would be drawn in that case as shown in Fig. 28 in full line marked "1932."

Commenting on his Fig. 21, which represents the Pennsylvania K4s long cut-off locomotive, Mr. Vincent states that "it is evident that they (the author's moduli) are not suitable for the lower range in this locomotive." This is far from being evident. An analysis similar to that made for the I1s locomotive proves just the opposite. Table 21 gives results of tests with locomotive K4s, marked by crosses in Mr. Vincent's Fig. 21. The locomotive has a ratio of outside evaporative heating surface to grate area of 58.3. The data have been taken from Pennsylvania Bulletin No. 29.

TABLE 21

Test No.	Rev. per min	Indicated tract. eff., lb	Dry coal per sq ft per hr, lb	Steam per ihp-hr, lb	Boiler eff., per cent	Overall locomotive eff., per cent
	...	Bulletin's column No. 394	339	381	350	399
4068.....	120	34,815	130.4	21.2	57.1	5.0
4062.....	160	29,578	164.7	21.2	50.3	4.5
4054.....	200	25,245	170.6	20.5	49.9	4.5
4025.....	240	20,814	137.3	18.7	56.6	5.0
4049.....	280	17,870	143.4	18.4	54.5	5.0
4070.....	120	30,884	85.2	18.5	67.0	6.7
4067.....	160	27,422	101.0	18.3	65.6	6.5
4012.....	160	22,854	70.2	17.0	69.8	7.4
4013.....	200	20,407	89.5	17.0	62.2	6.2
4014.....	240	18,115	91.5	17.4	66.2	6.5
4048.....	280	15,870	81.9	17.0	73.0	7.5

The first five lines represent higher points; the last six lines refer to the lower points, which are close to the "1932" curve. It can be seen that for the same speeds the tractive efforts differ sometimes very little, but the difference in the amount of burned coal and in efficiencies is very considerable. The overall locomotive efficiencies vary for the first group from 4.5 to 5.0 per cent, but for points which lie close to the "1932" curve they vary from 6.2 to 7.5 per cent. As it has been stated in the paper and repeated in the foregoing, the formulas recommended by the author have the objective to give average figures for *normal* operation, and therefore do not include cases where the locomotive is excessively forced. The recommended method gives curves with overall locomotive efficiencies of 6 to 7 per cent.

Fig. 19 shows the Texas & Pacific 2–10–4 locomotive, which is a locomotive with cut-off limited to 60 per cent. The author's curve is shown to be very close to Mr. Vincent's for higher speeds, and agrees with the test results given on the chart, in the transition curve region (speeds between 12 and 20 mph), even if the author's curve is lowered as it ought to be. The test results have not been published, and it is therefore impossible for the author to analyze the crosses given by Mr. Vincent, as it was done with the I1s and K4s locomotives. If those tests were available, the author is confident that the differences could be explained.

It is true that the "1932" factors were established mainly on the basis of tests with full cut-off locomotives. They seem, nevertheless, to be applicable to limited cut-off locomotives with moderate size cylinders. As it can be seen from Fig. 12-A, they correspond better to the test results than the Vincent formula. The Texas & Pacific tests show a fairly good agreement. The only locomotive which shows decidedly higher figures than the "1932" curve, and this only in the low-speed region, is the I1s locomotive, which has very large cylinders. It is possible that when more information on the performance of limited cut-off locomotives is available, the moduli for the ultra-limited cut-off locomotives may have to be revised, most probably for one speed only, 50 rpm, as the influence of limited cut-off at speeds where short cut-offs are used anyway, will hardly be noticeable. The method itself will not suffer from such a modification—just the opposite; this will only prove the flexibility of the method. As soon as for locomotives of a certain new type the steam rate, S_a, shown on Figs. 5 and 6, requires revision, it can be easily done, provided sufficient reliable experimental information is available, and new moduli can be established in accordance with formulas [14] and [15]. I do not see how Mr. Vincent can improve his mathematically derived formula on the basis of new experience.

If transition curves should be tolerated, the author's method could be easily adjusted for the time, until the new moduli are established. For ultra-limited cut-off locomotives, for instance, the following procedure may be suggested:

In cases when the boiler tractive-effort curve, based on the "1932" moduli, lies so low that it cannot be intersected with the rated, or maximum, cylinder tractive effort (Fig. 28), draw an arbitrary transition line from the intersection point of rated tractive effort with the 50 rpm ordinate, so as to merge with the "1932" curve.

The necessity for plotting the transition lines will be very infrequent; it may happen only in cases where the locomotive is decidedly over-cylindered. In cases where limited cut-off locomotives are built with moderate cylinder sizes, such a case will hardly happen, as can be seen from the example given of the

Boston & Albany 2–8–4 locomotive, as shown in Fig. 12-A.

There are several other remarks made by Mr. Vincent which are not correct and which call for a reply. In discussing the author's method, Mr. Vincent states that "the author's recommended method of expressing the tractive effort of a locomotive is dependent entirely upon the validity of the values shown in Figs. 4 and 5." Mr. Vincent further states, referring to the possibility of pushing locomotive evaporation to the limit: "This sort of operation cannot be used as a basis for developing a tractive-effort formula. Such a formula should be based on the normal operation of a locomotive designed in accordance with the most approved engineering practice known."

But this is exactly what the author did. As it was shown in comparing Mr. Vincent's curves with the author's in application to the Pennsylvania I1s and K4s locomotives, the Vincent curves, not the author's, were near the limit of boiler capacity with the lowest boiler and overall locomotive efficiency. And it was further shown that the author's curves agree very well with normal road-test results, while the Vincent curves were practically in all cases higher.

Reverting to the first statement regarding Figs. 4 and 5, it has been explained in the foregoing that by checking formula [6] with road test results, Figs. 5 and 6 were established as averages of Figs. 7 and 8. This was done on the basis of Fig. 4, and therefore its accuracy, so to speak, is included in Figs. 5 and 6; in other words, if values of β were taken erroneously smaller or larger than what they ought to be, the error is correspondingly reflected in values S_a, and the moduli figured on the basis of both β and S_a will be correct, because in their ratio the error will be eliminated. This was explained in the paper.

Likewise, as to the discrepancy in steam rates as shown in Fig. 25, the same thing should be remembered—namely, that the ultimate recommendations of the paper, as given in formulas [12] and [13] and Table 7, do not depend upon Figs. 4, or 5 and 6 separately. They are interconnected, and therefore should be used together or not at all. Table 7 should be considered jointly with formula [12] for power, or [13] for tractive effort. If the Cole evaporation figure E_c should differ from the actual figures, curves 5 and 6 will correspondingly differ. In such a case, moduli of Table 7 will deviate in the same sense, but the ultimate results (P_i and T_i) will be, nevertheless, correct, because the moduli are established by checking of test results on the basis of *calculated* figures E_c. That the actual steam rates do not differ much from S_a is an indication that, even taken separately, the various rates and coefficients used in the method are more or less correct.

Furthermore, it should be remembered that the steam rates which are shown on Fig. 25 are taken from the Pennsylvania test results, and, as Mr. Vincent says himself, they do not include auxiliaries, whereas curve 5, as it is plainly shown in the designation of the abscissas, represents steam rates including auxiliaries.

Mr. Vincent is of the opinion that the drop in the evaporation curve for speeds above 200 rpm, as shown in the author's Fig. 4, is not justified. He says that there is no apparent reason why a locomotive boiler, after it has reached its normal evaporation, should not continue indefinitely at the same rate. If Mr. Vincent had in mind "indefinitely" in relation to time, then he would be right, but he refers to speed above 200 rpm, and in this case the author's opinion is borne out by the Pennsylvania tests shown in Fig. 2. Unfortunately, road tests as they are usually performed in this country cannot prove the point, but as far as the conclusions of the paper go, it is immaterial whether or not the author's statement can be strictly proved with respect to evaporation, because as regards horsepower this is unquestionably correct in the majority of cases.

All performance curves given in the paper, except those for the Timken locomotive and the capacity test for the New York Central 4–6–4 locomotive, show for the horsepower a tendency to drop after a certain speed. The same applies to all modern German locomotives very thoroughly tested by Professor Nordmann, to whom reference was made on the fifth page of the paper.[45]

The Pennsylvania tests also show such a maximum point for locomotives; so, for instance, the I1s maximum horsepower (3863 ihp) had been reached at 160 rpm, and for the K4s, the maximum powers (3150.0 and 3155.3 ihp) had been obtained at speeds of 240 and 280 rpm, respectively, after which the powers dropped. It is true that conditions (*c*) and (*d*) of those enumerated by Mr. Vincent are the most important in extending the maximum horsepower beyond the highest speed, but so far, the conventional drafting arrangements and valve motions have not proved to give the desired effect in all cases. The high evaporation capacity of the Timken locomotive, as Mr. Pownall pointed out, may be due to the star-shaped nozzle and the very efficient drafting arrangement of the locomotive. It is further possible that improved valves of the poppet type and other improvements will also contribute to the rising of the power curve at high speeds, but neither of these improvements has been universally adopted. When these improvements will become regular features of locomotives and test data will be available, the moduli for 250 rpm will be revised, and new moduli will be established for higher speeds, but for the present there is no justification for increasing them beyond the author's recommendation.

Further, Mr. Vincent states that the construction of his curve TE_3 involves somewhat more work than does the author's moduli, but thinks that the additional labor is negligible. He probably overlooked that the moduli, which have been already prepared and are given in the paper, apply to all types of locomotives, and that there is no need of figuring them at all. Formulas [14] and [15] are given simply in order to permit the reader to check Table 7, or to establish moduli for intermediate speeds, if he wishes. Consequently, there can be no question that Mr. Vincent's method calls for more work than that of the author.

Thus, in conclusion and in addition to what has just been said about the amount of work, it would seem that Mr. Vincent's method gives approximately correct figures for a short portion of the speed range, gives exaggerated values for speeds below that range, and therefore requires in all cases plotting of arbitrary transition curves, whereas the author's method recommended in the paper makes use of a clearly defined curve and a straight line, and their intersection, without the necessity of resorting to transition curves. This is true for locomotives of all types, including limited cut-off engines, except when the limit of the cut-off is very low, in which case a transition curve might be of use until the new moduli for 50 rpm are established.

Everything that was said in the foregoing in relation to the principle of the Vincent boiler tractive-effort curve (TE_3) also applies to the method of Mr. W. F. Kiesel, Jr., of which method Mr. Vincent's formula is only a slight modification. At the end of his discussion Mr. Kiesel refers to Professor Wood's book, "Principles of Locomotive Operation." In Fig. 13 of this book, 1915 edition, Mr. Kiesel's curve is plotted in application to Pennsylvania E6s locomotive. It has exactly the same tendency of giving too high values at low speeds as the Vincent method. This portion of the curve between 23 and 18 mph is shown by Professor Wood with an upturn (broken line), similar to what has been found in Figs. 10-A to 15-A with the Vincent curves. If Mr. Kiesel's formula should be extended for speeds below 18 miles, the curve would go up still higher and the values would be exaggerated. It is evident that some transition curve has to be

[45] Theorie der Dampflokomotive auf Versuchsmässiger Grundlage, by Prof. H. Nordmann, Organ für die Forstchritte des Eisenbahnwesens, May 15, 1930, pp. 267–269.

used in this method too, in application to all locomotives, and such a curve is actually shown by Professor Wood.

There is a difference, however, between Mr. Kiesel and Mr. Vincent in figuring the boiler evaporation. While Mr. Vincent accepts the Cole evaporation figure E_c as equivalent to H in his formula [17], Mr. Kiesel does not think it necessary to figure the evaporation of a locomotive separately for various parts of the boiler as established by Mr. Cole. He recommends to figure the evaporation on the basis of *equivalent* heating surface, assuming the equivalent heating surface equal to the sum of superheater, flue, and six times the combustion-space heating surfaces, and the average evaporation equal to $11^1/_2$ lb per hr per sq ft of so-defined equivalent heating surface. The firebox heating surface seems to be a part of the combustion-space heating surface. This designation for the equivalent heating surface is based on an empirical law found by Mr. Kiesel—namely, that the evaporation of a combustion-space heating surface is six times as high as that of the flues and superheater, if the length of the flues is about 20 ft. Mr. Kiesel further assumes that 10 lb per sq ft of equivalent heating surface represents steam available for use in cylinders.

I do not think that such a rough method of calculating the evaporation would satisfy us, with all the knowledge that we have now, especially the inclusion of superheating surface into the total surface for figuring evaporation. This can be considered only as a necessary correction for the increase of power due to superheating, although one correction for the effect of superheating is already included in the Kiesel formula by referring w to pressure P with 100 deg superheat. Moreover, such a method offers no advantages over the more accurate Cole formula, which has been proved to be satisfactory by many years of use, and to hold good even now for modern locomotives, as it has been shown in the paper and corroborated by many discussers. A simple calculation of evaporation for any of the locomotives cited in the paper, for which we have data, will show that Mr. Kiesel's empirical rule gives highly exaggerated figures. They are not evaporation figures any more, but simply values of W to be substituted in Mr. Kiesel's formula, which thus becomes a purely empirical formula. Mr. Kiesel concedes this, but is of the opinion that the formula is based on a rational theory. This may be true, but the rationality of it is more than offset by the omission of a great number of important factors which cannot be expressed by mathematical formulas, and therefore must be compensated by such figures as evaporation of a superheating surface and others, analyzed in the author's discussion of Mr. Vincent's formula.

Mr. Kiesel did not show any examples of the application of his method to those locomotives for which he says that his formula proved satisfactory for over 20 years of experience, and therefore the author could not check his method, as he did it in discussing the Vincent method, but there can be no doubt that if applied to locomotives cited in the paper, his formula will not agree with the known test results because the structure of Mr. Kiesel's formula is the same as that of the Vincent formula. It may, though, agree with the results of the Pennsylvania I1s locomotive, for the same reasons that were stated in discussing the Vincent method, and, most probably, for low speeds only.

There are several things in Mr. Kiesel's comments which call for a reply. He states that "tests of Pennsylvania Railroad locomotives classes K4s, E6s, K2sa, and L1s made in 1914 and 1915, and the I1s made in 1919, used as examples by the author, are rather too old, as much improvement in locomotive design has been developed since then." The test figures of the enumerated locomotives have been used only in connection with boiler evaporation figures, and as far as the evaporation of the boiler is concerned, very few changes in the conventional design of the boiler have been made. Furthermore, the results of these tests have not been used directly in the examples given by the author, by which he tried to prove the correctness of his method, and still further, they have no bearing on the method, as finally recommended, for reasons advanced in the paper and in this closure.

Neither can the author agree with the statement that he is vague on the point of how much steam can any given locomotive make available for use in the cylinders. There is a definite formula in the paper for the amount of steam available for the cylinders—namely, $E(1 - x)$, just preceding formula [6]. This is made use of in formulas [6] and [6-A]; evaporation E is explained and figured in the paper, x is included in Figs. 5 and 6. Mr. Kiesel's statement that the author "determines drawbar horsepower" while he prefers "the cylinder tractive effort" is evidently a misunderstanding, as the paper is about the indicated horsepower and tractive effort, with hardly mentioning the drawbar horsepower at all.

The question of *maximum* versus *performance* tractive effort was discussed by Messrs. G. T. Wilson and A. Giesl-Gieslingen The latter's discussion may be considered as a reply to Mr. Wilson, which makes a further reply unnecessary. The author will only add that he stated in the paper his reasons for working out a method by which the performance, and not the maximum horsepower and tractive-effort curves, would be plotted. It would not be desirable to develop a method by which only capacity curves, such as shown on Figs. 10 and 11, would be obtained. These two locomotives, both of the New York Central Railroad, and tested under similar conditions, give maximum capacity figures, which, according to Mr. Wilson, differ from those obtained by the author by 17 per cent in one case and 42 per cent in the other. Similar relation of discrepancy would be obtained if the Cole, or the Vincent, or the Kiesel formula were applied, although the figures might be different. No method would satisfy both capacity curves, and therefore these high power figures must be partly ascribed to local conditions, such as quality of fuel and method of operation, which may be different on different roads, and partly to the design of the different parts, which cannot be expressed by a formula. It would not be advisable to have a yardstick formula for comparison based on the capacity of the 4–6–4 J1 locomotive. These capacity curves every railroad has to find out for itself.

As to the value of capacity tests, it has been also pointed out in the paper that unless there is complete assurance that the capacity tests have been made without exceeding the sustained evaporation of the boiler, the capacity figures are likely to be exaggerated and cannot be taken as basis of locomotive performance.

Mr. Ching Pong Pei raises several interesting points. He seems to be somewhat puzzled that no reference has been made in the paper to the rate of firing. He refers to Mr. Cole's evaporation figures as "indefinite and ambiguous" because they are not tied up with a certain rate of firing.

As far as Mr. Cole's figures are concerned, the author is to blame for not having stated definitely that Mr. Cole was considering the rate of firing of 120 lb per sq ft per hr. As the author made reference to Mr. Cole's publications and the Alco Handbook, he did not think it necessary to repeat Mr. Cole's firing rate. Personally, he does not attach much significance to the rate of firing in the case of modern locomotives. When Mr. Cole devised his figures, the grate areas were comparatively small. Modern locomotives have much larger grate areas, and as stated in the paper on the fifth page, "in well-designed locomotives the proportions are such that the two factors more or less balance," the two factors being the limitation of the amount of coal which can be burned on a certain grate and the amount of heat which can be absorbed by a certain heating surface. In modern locomotives the second is more often than the first the ruling limitation, and it is therefore permissible to consider the evapora-

tion as determined by the heating surface, rather than by the grate area.

In view of the foregoing, smaller firing rates than what Mr. Cole figured on are now actually being used. Mr. Wilson states that for the performance test of the J-1 and L-2 locomotives the average[46] firing rate was 62 to 66 lb, and during their capacity tests, 125 lb of coal per sq ft of grate area per hour was to be the limit, without stating, though, how this was determined. Dr. Giesl-Gieslingen believes that in some tests on the New York Central, when the boiler was forced to the limit, the firing rate was 150 lb. In his paper the author was not interested in firing rates in view of the indefiniteness of the figures and also for the reason that they fluctuate too much. Moreover, they cannot serve as indications of forced or normal operation of a locomotive, as this depends upon the relative size of the grate. Therefore, the author would not be in favor of Mr. Ching Pong Pei's suggestion to plot a series of tractive-effort curves for various rates of firing. This would be a very indefinite basis for comparison, although it may be an interesting method of scientific investigation of locomotives of certain types.

The author already pointed out in the paper that, strictly speaking, tractive effort becomes a definite conception only when the conditions at which the tractive effort is produced are specified, as for instance the maximum tractive effort, the most economical tractive effort, the constant-evaporation tractive effort. Mr. Ching Pong Pei's suggestion is along these lines—he would have constant-firing-rate tractive efforts. Much more reliable conclusions can be drawn if these curves are plotted for different constant-evaporation rates. Professor Lomonossoff was the first to develop this method, and he used it in all his tests, giving curves for different evaporation rates expressed per unity of heating surface per hour.[47] Professor Nordmann, as it has been mentioned in the paper, refers his test results to one constant evaporation rate, which is 57 kg of steam per square meter of fireside heating surface. A still better way of comparing locomotives, at least for some purposes, would probably be on the basis of equal overall efficiencies.

However, the difficulty with all these methods, in which a system of curves is considered, is that during actual operation the locomotive is not burning a constant amount of coal per unity of grate area per hour, nor is it generating a constant amount of steam per unity of heating surface per hour, nor does it work at a constant efficiency. These factors are varying all the time, and in the opinion of the author, the higher rates of firing and evaporation are obtained at higher speeds. If anything, the locomotive is more likely to work at constant efficiency at certain modes of operation—performance, maximum performance, capacity. If a tractive-effort curve obtained from test performance, or capacity, be plotted over a chart consisting of a series of curves for different constant rates of firing or evaporation, probably it will be found that the actual tractive effort intersects all the other curves. If, therefore, a chart as suggested by Mr. Ching Pong Pei is presented to a railroad operator, as he recommends, it would be necessary to give him a table, or chart, indicating the speeds at which the different rates are actually materialized. This would enable him to determine points, one on each of the curves, and plot the actual tractive-effort curve.

The author preferred, therefore, not to be bound by such purely theoretical considerations and to consider the curves which from actual tests are known to be obtainable in every day's service. These, called *performance curves*, can be plotted as shown in Fig. 16 of the paper for the New York Central 4–6–4 locomotive. These curves correspond to reasonable overall efficiencies (about 6 to 7 per cent), as has been shown in the discussion of Mr. Vincent's method, and to firing rates for modern locomotives up to 90 to 110 lb per sq ft of grate area per hour, as can be figured out from the total evaporation (E_c), average evaporation of coal (7 lb of water per 1 lb of coal), and grate area of any of the foregoing examples.

The author is not in agreement with the second and third premises, quoted by Mr. Ching Pong Pei, as those on which the correlation between boiler evaporation and rotative speed of the locomotive is based. Neither of the two premises has ever been mentioned by the author; as a matter of fact, the meaning of the second is not clear to him. The correlation referred to is for him a matter of observation of innumerable locomotive tests and conclusions drawn from stationary tests. As it has been stated elsewhere, the ultimate recommendations of the method do not depend upon the correctness of the assumed correlation.

Mr. J. E. Muhlfeld takes issue with the author in considering the type "E" superheater as a part of the "modern locomotive," asserting that type "A" superheaters are able to give just as high superheats as the type "E." In this latter respect Mr. Muhlfeld is quite right. In his conclusion, which was partly cited by Mr. Muhlfeld, the author stated also that "this [the use of the Cole method for all locomotives with type 'A' superheaters] should not imply, however, that type 'A' superheaters can never develop the horsepower recommended by the new method. Locomotives are known that have given very high performance figures with type 'A' superheater on good coal."

It is obvious that the locomotive power depends upon the proper superheat, and that if type "A" superheater is so dimensioned that it can insure as high a temperature as type "E" superheater, the performance of a locomotive equipped with this type of superheater should be just as good as that with the type "E." It has also been stated by the author that it would be better to refer the performances not to the type of the superheater, but to the temperature of superheat. However, it was pointed out that "it was not thought advisable to give constants for various superheats for the reason that before a locomotive is tested, its superheat is not known, and therefore these constants would not be helpful in calculating the horsepower of a locomotive beforehand." It was also mentioned that it would be more logical to base such constants on the relation of the superheating surface to the evaporative heating surface. The author tried to find a law for such a dependence, but so far such comparisons did not give conclusive results.

There is one important thing which justifies considering the type "E" superheater a feature of the "modern locomotive," especially American; namely, it is difficult, even impossible, to build a type "A" superheater with a sufficiently large heating surface in a large locomotive boiler, while the type "E" superheater permits doing it. The ratio of the superheating surface to the evaporative heating surface in the largest type "A" superheater is only about 0.32, whereas it is possible to design a type "E" superheater with a ratio of 0.45. The majority of locomotives with type "E" superheaters has a ratio of 0.41 to 0.42. The examples which Mr. Muhlfeld cites in favor of the type "A" superheater refer to small locomotives. Likewise, all European locomotives, which have mostly type "A" superheaters, are of comparatively small size. The difficulty occurs only when locomotive heating surfaces reach 4000 to 4500 sq ft, for which the type "E" superheater seems to be indispensable.

Mr. Muhlfeld is aware of this advantage of the type "E" superheater, because he says: "The particular feature of the type 'E' superheater has been its increased superheating surface, which reflects favorably on the boiler capacity." This is

[46] Presumably the average for a trip, as this is the only firing rate that can be determined on a road test.

[47] See his numerous publications; for instance, "Lokomotivversuche in Russland," Berlin, 1926.

exactly the reason for singling out this type of superheater by the author, because "boiler capacity" is the main consideration in the problem analyzed by him in the paper. As to the disadvantages of the type "E" superheater enumerated by Mr. Muhlfeld, they are of no import to the author from the point of view of the paper—as long as, notwithstanding these disadvantages, type "E" superheaters are used in the majority of road locomotives of recent build ("modern locomotives"), the author's duty is to evaluate these superheaters.

The author notes with pleasure that Mr. Muhlfeld has no criticism to offer regarding the suggested method. He probably realizes that this method can have a broader application than the Cole method. The power and tractive effort of the Delaware & Hudson locomotives with high-pressure water-tube boilers, mentioned in Mr. Muhlfeld's discussion, cannot be readily figured by the Cole method, because their ratios of cylinders and heating surfaces are quite different; but as soon as the evaporation of these boilers, and the steam rates of compound or triplex expansion engines of the locomotives are known, the evaluation of a locomotive of this type will be just as easy as of any other locomotive.

Mr. T. C. McBride is of the opinion that the increase in evaporation capacity of a boiler equipped with feedwater heater, found by the author to be 7 per cent, is underestimated, and that 15 per cent would be a more correct figure. He may be right in that the extreme 15 per cent figure calculated by Mr. McBride on the basis of heat saving is correct when everything is in first-class condition and the locomotive is working at the peak of its capacity. For average conditions 7 per cent is a more acceptable figure, although it may seem somewhat conservative.

Mr. McBride does not believe that road tests for which the author's figures were taken can be sufficiently accurate. But tests made on the Pennsylvania testing plant with I1s locomotive equipped with a feedwater heater of the open type showed a saving in heat fluctuating between 4.7 and 10.2 per cent, of which 7.45 per cent is the average. Therefore, when the evaporation figures found by the author were about 7 per cent above the Cole evaporation figure, he thought that the most simple way of introducing the feedwater heater into the ratios would be leaving the Cole figure unchanged and adding 7 per cent to that for locomotives equipped with feedwater heaters—this being in good agreement with stationary test results. The author believes that 7 per cent is a fair figure for average operating conditions, for the performance curve on the basis of available experimental data.

Mr. G. F. Starbuck raises the question of the application of the recommended method to limited cut-off locomotives and locomotives equipped with a booster. As it has been stated in the discussion of Mr. Vincent's method, the recommended method is applicable to the moderate limited cut-off locomotive. This has been proved by the example of the Boston & Albany A-1 engine and other locomotives discussed there. For the ultra-limited locomotive of the I1s type it has been shown that a correction, either by the temporary use of a transition curve or by a change in the modulus for the lower speed (50 mph), might be needed.

As to the booster, the tractive effort of the latter can be added directly to the rated (cylinder) tractive effort, but only up to the intersection point with the boiler tractive effort. This additional tractive effort will have its full value at starting and at very low speeds (several miles per hour), and will gradually have to come down to nothing at the intersection of the rated and boiler tractive efforts for lack of steam. There can be no advantage in using a booster for speeds above the adhesion speed (the highest speed at maximum tractive effort, corresponding to the intersection of rated tractive effort with boiler tractive effort), because when the tractive effort is limited by the boiler capacity, it is more advantageous to make use of the main engine only, which has a higher efficiency than the booster.

In connection with these questions, the author would like to mention that the method is also entirely applicable to switching locomotives, and although in some cases the tractive effort of such a locomotive will be lower than that obtained by the Cole method, the smaller figure of the method recommended in the paper is the correct one. This figure will be smaller than Cole's only for locomotives where the boiler percentage is below a figure close to 100. The Cole method in this case gives exaggerated figures.

I do not quite agree with Mr. H. B. Oatley when he says that "the new method offers formulas somewhat more complicated than have been used hitherto." This would be true if we exclude the figuring of boiler evaporation from the Cole method, and compare only the Cole figuring of cylinder horsepower and tractive effort on the basis of the Cole factors with the recommended method, in which, in addition to the moduli, the boiler evaporation must be also figured. However, ordinarily the Cole method requires the figuring of the boiler percentage, and in this case the knowledge of the boiler evaporation is also required.

Mr. George W. Armstrong is right in calling my attention to the discrepancies which might be found if the curves according to my method are compared with test results of locomotives equipped with improved drafting arrangements—and this is true whether the improvement be a more efficient smokebox arrangement, or the annular-ported exhaust nozzle, or other improvements. Mr. W. A. Pownall also called attention to this fact, giving figures obtained from the Timken locomotive, which gave exceptional results at high speeds. This is true, as it has been stated at the beginning of the discussion; the recommended method cannot take care of such improvements which have not yet become a composite part of a conventional locomotive. When all locomotives, or at least a great majority of modern locomotives, will be equipped with an improved standard drafting arrangement, the figures will have to be revised, but this will not affect the method itself.

Mr. L. K. Sillcox's criticism is expressed in the last three paragraphs of his discussion. It is preceded by a very interesting and valuable analysis of conditions of good combustion, essentials of good design, and elements of correct proportioning of locomotives, and in conclusion it is suggested to increase the Cole factors by about 10 per cent. This suggestion has no connection with the preceding analysis, and is even in contradiction with some of his statements, as for instance, with the following just remark: "Merely cylinder diameter and stroke are of but little service for guidance as to the actual amount of steam a locomotive will require under different conditions of operation, and such ratios as cylinder volume to grate area, cylinder volume to heating surface, and the like have no meaning whatever."

Table 15 is given by Mr. Sillcox in which new factors are suggested, differing from Cole figures by 9.2 to 9.5 per cent. They are marked "test;" it is claimed that they are based on recent tests and that the difference between them and Cole figures represents the advantage of the locomotive "constructed in 1933 as compared with 1910," when Mr. Cole devised his method.

The author is not familiar with the tests to which Mr. Sillcox refers, and as no particulars of the tests or any data are given, the author is not in a position to dispute or confirm the suggestion of raising the Cole factors by 9.2 to 9.5 per cent for all speeds. He may only state this—when the need for a revision of Cole factors became apparent, the first thought was to increase Cole factors in a certain proportion. This was tried, but it was impossible to establish a uniform rate for all cases. This can be easily concluded from Figs. 10 to 15, in which the Cole curves,

together with test performance curves, are shown. At low speeds (50 to 80, or 110 rpm; piston speeds approximately 250 to 400, or 440 fpm) they were found in some cases above performance curves, and their further increase was not desirable (Figs. 10, 12, and 13), while in others the whole curve lay below the performance curve (Fig. 11), and still in others the curves touched each other at 50 rpm. The reason for this was evident—these locomotives had different boiler percentages. The next thing the author tried was to change the Cole curves in relation to the boiler percentages. He got more consistent results, but then it occurred to him that by doing so he eliminated the Cole values altogether, because the cylinder horsepower was thus introduced twice, in the numerator and the denominator, and these two values canceled each other, leaving only a value proportional to the evaporation. Then the natural thing to do was to consider the evaporation only—what he did in his method.

The author explained in his paper the reasons for preferring to figure the locomotive characteristics on the basis of boiler dimensions rather than cylinder sizes. It is not necessary to repeat here the arguments, but it suffices to say that if we agree to limit ourselves to locomotives of a certain period and type, any essential part of a well-proportioned locomotive may be chosen as a yardstick for measuring its power. Superheating surface, weight, even length of a locomotive, could be selected for that purpose, and it would be found that the ratios of power to each of the enumerated dimensions fluctuate very little in modern locomotives. Nobody would seriously consider measuring the power of a locomotive by its length, but in the author's opinion, there is more justification in measuring it by the heating surface of the superheater (of a certain type) than by the size of cylinders. In Table 22 horsepowers of all locomotives cited in the paper are referred to the product $p_b \times A$ (boiler pressure times piston area),[48] which represents the main factor in figuring the power of a locomotive according to Cole and also to Mr. Sillcox. The only difference between them is that in Cole's formula these products are multiplied by certain factors, while Mr. Sillcox would increase these factors by 9.2 to 9.5 per cent.

The ratios of column 3 of Table 22 vary from 0.0230 to 0.0294, or 28 per cent. In the same table Cole evaporation figures E_c (not counting the increase due to feedwater heaters) and the ratios of power to these figures are also given for comparison.

TABLE 22

Locomotive	Maximum ihp (performance)[49] 1	$pb \times A$ 2	Ratio of col. 1 to col. 2 3	Total evaporation, lb per hr 4	Ratio of col. 1 to col. 4 5
New York Central, 4–6–4	3250	110,447	0.0294	54,662	0.0594
New York Central, 4–8–2	3240	128,825	0.0252	59,514	0.0544
Lehigh Valley, 5100	3750	143,139	0.0262	70,530	0.0531
Lehigh Valley, 5200	3800	135,587	0.0271	71,694	0.0530
Timken, 1111	3650	143,139	0.0255	67,370	0.0542
Boston & Albany, A-1	3400	147,780	0.0230	62,958	0.0541

[48] A. L. Co. Handbook, 1917, p. 54.

[49] Not capacity; read from charts, and may slightly differ from recorded figures.

They fluctuate only 12.1 per cent—much less than ratios in column 3.

The dependence of Cole's method upon cylinder sizes will always make the correctness of his figures a function of boiler percentages. For switching and limited cut-off locomotives with comparatively small boilers and large cylinders, any revised Cole figures will be exaggerated, while for high-speed locomotives with small cylinders, such as in the New York Central 4–6–4 engines, the opposite will be the case. At the same time, the real source of power, the boiler, will not be taken into account. Mr. Sillcox's reasons for his recommendation to preserve the Cole method and modernize his figures are "familiarity, confidence, and universal acceptance" of the method. In the author's opinion, if it is agreed that the Cole method is not based on correct premises, the enumerated advantages of the Cole method are of little importance. As the art progresses, Cole figures will have to be revised from time to time, and while this in itself is not a handicap, the difficulty is that test results give no indication how to proceed with the revision of Cole factors. A summary rise of all, or even of several, factors will always be a very approximate solution of the question. Tests will always determine the two fundamentals: improvement (1) in boiler evaporation and (2) in steam consumption, irrespective of the boiler and engine design—whether Stephenson or water-tube firebox, high-pressure or low-pressure steam, simple-expansion or compound- or triple-expansion cylinders, or even a turbine instead of cylinders. As soon as these data are known, they can be immediately and directly applied to the suggested method and new constants determined, because the method is based on the same two fundamentals: boiler evaporation and steam consumption. It can not be done with the Cole method. This makes the method suggested in the paper flexible and really universal, although, maybe, not universal in Mr. Sillcox's sense.

Mr. F. E. Russell is of the opinion that by referring mean effective pressure to piston speed we would "stick to fundamentals," whereas using rotary speed we introduce an approximation. The author, by referring to Sir J. Alfred Ewing and W. E. Dalby, greatest living authorities on steam-engine theory, tried to prove that piston speed is not a fundamental in steam-engine performance. He has never seen anything in favor of this, except Mr. Cole's supposition, and does not find in this factor anything more than the "familiarity" due to long usage, as pointed out by Mr. Sillcox. Furthermore, in his paper he showed that the suggested method can be used on the basis of piston speed if desired, although it involves some more work.

The author appreciates the reception given by the various discussers to the author's paper and their valuable discussions. He is very grateful to those who, like Mr. Riegel, stated the results of checking the recommended method against actual test results, and would welcome statements of other railroad engineers regarding the application of the author's method of figuring horsepower and tractive effort to modern locomotives in comparison with actual test results.

Canadian Pacific Railway's Multi-Pressure Locomotive 8000

Its Steam-Generating System

By F. A. SCHAFF,[1] NEW YORK, N. Y.

The conventional type of locomotive boiler with stayed surfaces is generally considered as not suitable for pressures above approximately 300 lb per sq in. Therefore some sort of water-tube construction is essential for pressures materially higher. On the Canadian Pacific locomotive 8000 the Elesco multi-pressure system seems to adequately meet the high-pressure requirements, but offers corollary advantages contributing to ease of maintenance. The system consists of three units: (1) the closed circuit, (2) the high-pressure boiler, (3) the low-pressure boiler. An outstanding feature of the water-tube construction is the use of distilled water in the closed circuit, so that no scale can be formed in the water tubes which are in contact with the fire.



THE development of steam generation has been characterized by a steady trend toward higher steam pressures. At the beginning of the century, locomotive boiler pressures had risen to 180 or 190 lb from 150 or 160 lb per sq in., which had been considered high pressure during the preceding decade. By 1910, 200 lb was generally regarded as the limit for simple-expansion locomotives, although during the compound era 215 and 225 lb per sq in. had been used to some extent. With the introduction of superheated steam there was a lull in the advance of pressures as the advantages gained by the use of superheat were adequate to meet the then requirements of added capacity and fuel economy. In more recent years the insistent demands for greater economy and higher sustained locomotive capacity, with heavier loads and faster schedules, caused designers to again increase the working steam pressure and also to provide for higher steam temperatures.

The locomotive boiler in all its major features has changed but little, save in size, since the days of the "Rocket," "Old Ironsides," and "John Bull." It is true that proportions have been changed and many devices have been added which have materially increased the efficiency, but in the main, the boiler is as it was a century ago. Brotan in Europe and others, following in his wake, built water-tube fireboxes in order to better provide for higher working pressures. This trend evidenced itself in America through the introduction of the Jacobs-Schupert firebox on the Santa Fe, the Baldwin locomotive No. 60,000, the McClellan firebox on the New Haven, and more recently the water-tube fireboxes on the Delaware & Hudson. The Gresley-Yarrow water-tube boiler on the London & Northeastern in England and the Winterthur locomotive in Switzerland represent constructions which aimed at suitability for higher working steam pressures.

With the present state of the art it is generally considered that the conventional type of locomotive boiler with stayed surfaces is not suitable for pressures above approximately 300 lb per sq in. It is essential therefore to utilize some sort of water-tube construction for pressures materially higher than this figure.

The Elesco multi-pressure system as used on Canadian Pacific No. 8000 seems to adequately meet not only the requirements for substantially high pressure, but offers corollary advantages contributing to ease of maintenance. The system consists of three separate units:

(1) The closed circuit
(2) The high-pressure boiler
(3) The low-pressure boiler.

Closed Circuit

The shaded part of Fig. 1 shows the closed circuit, a water-tube boiler, forming the firebox and combustion chamber of the locomotive. The bifurcated seamless steel tubes are 2 in. and $2^1/_2$ in. in outside diameter. The cross-section shows that the furnace walls are formed by one straight riser tube and one bent crossover tube changing alternately.

The tube ends are rolled into the firebox ring at the bottom and at the top into the steam separator drums. A special feature of the boiler is that the downcomer tubes are not located within the gas path, but outside of the wall of riser tubes, thus insuring a positive water circulation without the dangerous reversals of flow encountered on some designs of water-tube boilers. The tube system is filled to approximately the center of the steam separator drums with distilled water, which serves as a heat carrier.

The steam generated is taken from the steam space of the separator drums and carried by short riser pipes to the heat-transfer elements located in the main steam drum on top of the firebox. During its flow through these elements the steam, by losing its latent heat to the lower temperature feedwater of the high-pressure boiler, condenses, thus generating indirectly the high-pressure live steam used in the high-pressure cylinder. The condensate flows out of the lower part of the heat-transfer elements through the downcomers or condensate tubes to the water collectors at the bottom of firebox and combustion chamber. There it again enters the riser tubes, thus completing the circuit. As distilled water is used and the system is sealed, no scale can be formed in the water tubes which are in contact with the fire.

This is probably the outstanding feature of this design, as one of the principal difficulties with water-tube construction in locomotive boilers has been the necessity for careful and frequent

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

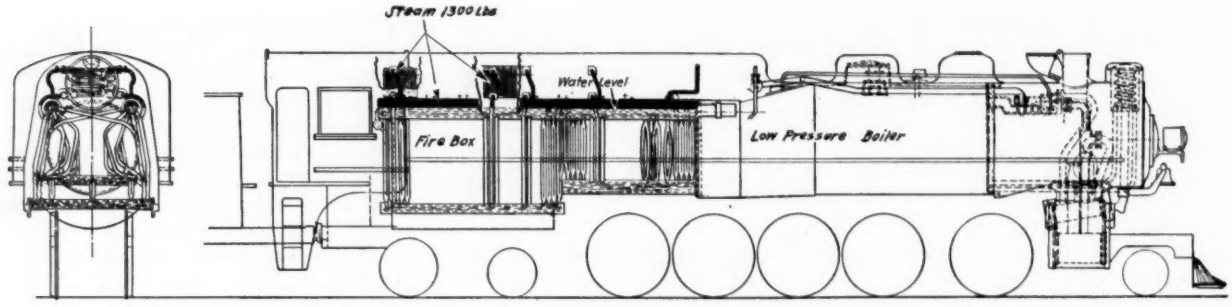


FIG. 1 MULTI-PRESSURE LOCOMOTIVE, STEAM-GENERATING SYSTEM
(The 1700-lb pressure closed system.)

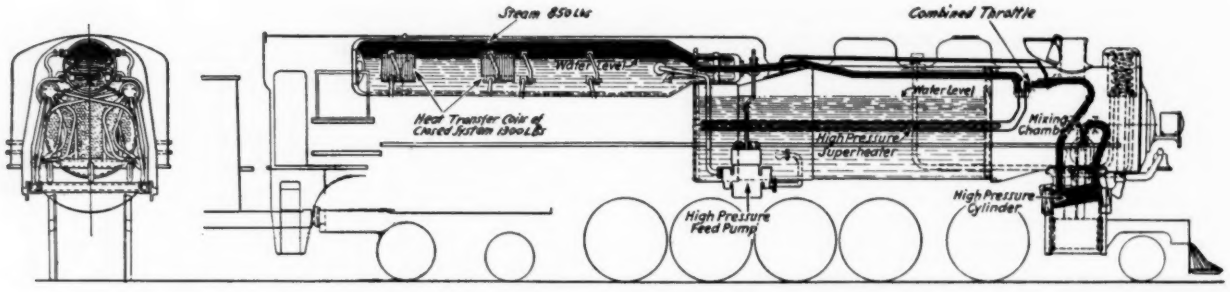


FIG. 2 MULTI-PRESSURE LOCOMOTIVE, STEAM-GENERATING SYSTEM
(The 850-lb high-pressure boiler, with water and steam piping.)

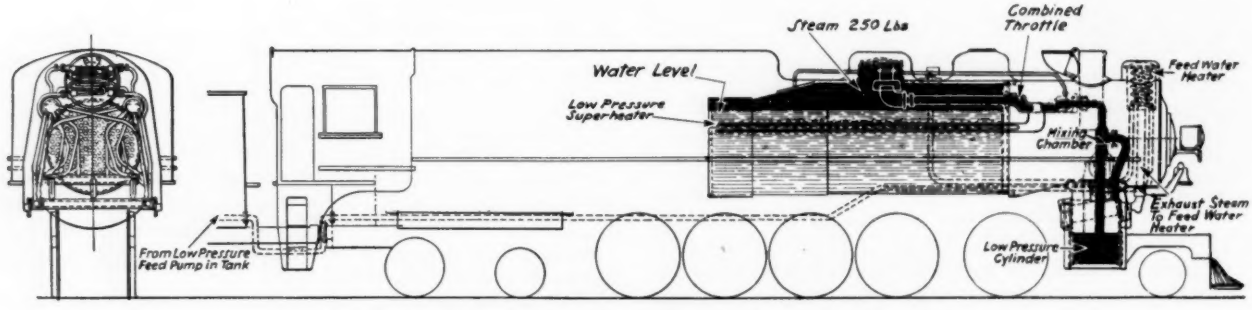


FIG. 3 MULTI-PRESSURE LOCOMOTIVE, STEAM-GENERATING SYSTEM
(The 250-lb low-pressure boiler, with water and steam piping.)

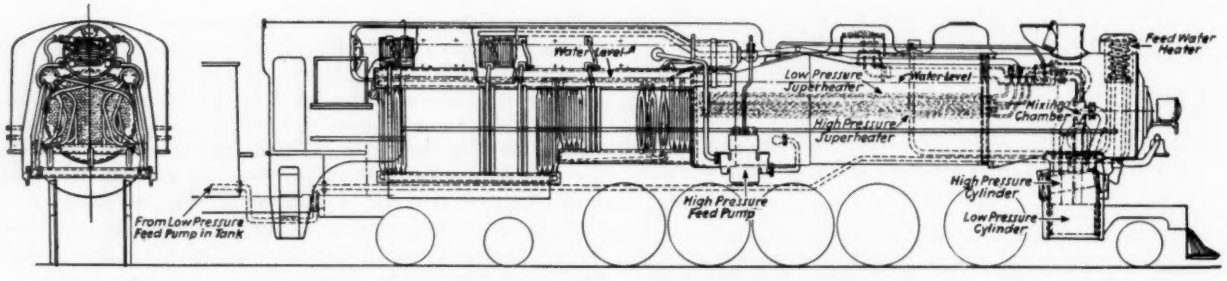


FIG. 4 MULTI-PRESSURE LOCOMOTIVE, STEAM-GENERATING SYSTEM
(Combination of the three systems.)

cleaning of the tubes to avoid accumulation of scale. This cleaning is a slow and expensive operation, requiring the locomotive to be out of service a considerable portion of the time. With the indirect system as used on the 8000, the water tubes will never require cleaning. Thus aside from the time and labor saved, there is the further advantage of avoiding the breaking and remaking of numerous high-pressure joints at the washout plugs etc.

The flow of water and steam in the system is a natural circulation depending upon the hydraulic head of the liquid and unassisted by mechanical means. There is no fixed steam pressure in the closed circuit, but it depends upon the steam output of the locomotive. Under normal conditions the boiler carries about 1350 lb per sq in.; under peak loads it may rise to 1600 lb.

The pressure in the closed circuit is such that the required temperature differential is obtained and the heat flow from the closed circuit steam to the high-pressure boiler water produces and maintains the 850-lb pressure in the high-pressure boiler. The two safety valves of the closed circuit are set at a pressure of 1700 lb per sq in., in order that there shall be an adequate range which will preclude the loss of water.

Fig. 2 will make clear the arrangement of the high-pressure boiler. The high-pressure steam of 850 lb per sq in. is generated in a seamless forged nickel-steel drum 39 in. in inside diameter, located above the closed circuit and protected from contact with flames and firebox temperatures by the cross-over tubes of the closed circuit, by a plate of heat-resisting steel, and by lagging. The high-pressure steam is generated indirectly by means of the heat-transfer elements in the boiler drum. The average water level in the drum is about 7 in. above center line, so that from 80 to 90 per cent of the elements are covered by water. The steam flows through two dry pipes located at the highest part of the boiler and perforated at their upper circumference. They emerge as one pipe at the front end of the drum, lead to a shut-off valve, and from there to the high-pressure superheater header in the smokebox of the locomotive.

Fig. 3 shows the third part of the system, the low-pressure boiler, carrying 250 lb per sq in. pressure. It is similar to the barrel part of an ordinary boiler with a circular rear flue sheet riveted to the boiler shell instead of to the firebox. The boiler is fitted with flues $3^1/_2$ in. in outside diameter, containing superheater units of the type-E design $1^3/_{16}$ in. in outside diameter. The units are divided into two groups near the vertical center line, those on the right-hand side composing the high-pressure superheater and those on the left-hand side forming the low-pressure superheater. Both superheaters are made of the same material, and corresponding units have like dimensions. Maximum interchangeability is therefore provided.

Fig. 4 shows a combination of the three systems as an illustration of the water and steam flow in the boilers. Ordinary feedwater is drawn from the tender tank by a standard feed pump, and forced through the exhaust-steam feedwater heater in front of the stack to the low-pressure boiler, which it enters at a temperature of 200 to 220 F. The low-pressure boiler serves partly as an evaporator for the low-pressure steam and partly as an economizer for the high-pressure boiler.

The high-pressure feedwater is drawn by the high-pressure feed pump from the low-pressure boiler at approximately 250 lb per sq in. pressure and 400 F temperature, and delivered to the high-pressure boiler drum. Besides using the less expensive and lighter heating surfaces of the low-pressure boiler for preheating the high-pressure feedwater, this procedure has the advantage that most of the scaling matter remains in the low-pressure boiler, where it can be easily cleaned out. Only a small amount of foreign matter is carried over into the high-pressure drum. Inasmuch as the high-pressure drum is not exposed to flame, any foreign matter which may be deposited in the drums or on the heat-transfer elements is of a soft nature and can be readily washed off through the clean-out holes provided along the top of the drum.

The steam generated in the high-pressure boiler at 850 lb per sq in. pressure flows through the outside dry pipe to the superheater header in the smokebox, then through the units of the type-E superheater through the multiple-valve throttle to the high-pressure cylinder located between the frames underneath the smokebox. Steam from the low-pressure boiler at 250 lb per sq in. in pressure passes through a tangential drier and a conventional dry pipe to a low-pressure superheater header in the smokebox. From there it passes through the low-pressure type-E superheater to the multiple-valve throttle, then to two mixing chambers located in the smokebox.

These mixing chambers are formed by increasing the diameter of each low-pressure steam pipe. The low-pressure live steam is mixed with the high-pressure exhaust steam by means of a perforated nozzle located in this chamber. The mixture flows to the two outside low-pressure cylinders. By this means the high-pressure exhaust steam is reheated in a simple manner, and the difficult problem of oil separation, so vital for all types of separate reheaters, is avoided. The exhaust from the low-pressure cylinders is utilized in the usual way for drafting the engine and preheating the low-pressure feedwater.

The high-pressure and low-pressure superheater headers are of the type-E throughbolt design. They are separate, but cast integral. Each header is fitted with a multiple-valve throttle, the high-pressure and the low-pressure differing only in so far as the high-pressure throttle has valves of considerably smaller diameter on account of the increased pressure and greater density of the high-pressure steam. The two throttle camshafts are linked together in such a manner that each low-pressure valve opens ahead of the corresponding high-pressure valve. The camshafts are operated by one throttle lever in the cab, the throttle operation being exactly the same as with a normal locomotive.

The boiler heating surfaces and the cylinder sizes and cutoffs are so calculated that full pressure is maintained in high-pressure and low-pressure boilers during operation. In order to prevent popping of the high-pressure safety valves when the throttle is closed, and for accelerating the steaming of the low-pressure boiler at firing-up, a crossover line is arranged which allows superheated high-pressure steam to be passed into the low-pressure boiler. Flow of steam through this line is controlled by a cone-seated valve.

There are three water gages on the boiler. The one for the closed circuit is placed outside of the cab, to be observed only at the beginning and end of each run to ascertain that no water has been lost in service. During the run the system is controlled indirectly by two thermocouple pyrometers, which show the temperature of the steam and give an indication of the performance of the closed circuit.

Inasmuch as the low-pressure boiler tubes are only in contact with gases of relatively low temperature, together with the absence of the crown sheet of the normal type of boiler, the safety of the design is apparent.

As before stated, the water in the tubes forming the firebox and combustion chamber is scale-free. Overheating of tubes due to scale formation, a quite common failure on ordinary high-pressure boilers, is therefore entirely eliminated.

An additional safety factor is the relatively low energy stored in the multi-pressure boiler, in spite of the higher steam pressures. All three units together contain only approximately 70 per cent of the Btu found in a normal pressure locomotive of equal capacity. The separation of the boiler into three units increases the

safety still further, as all three units are not likely to be damaged at the same time. In case of a failure of a water tube in the closed circuit due to faulty material or leakage, the energy of the escaping steam will be only about 15 to 20 per cent of that liberated by a corresponding failure on a standard-type boiler. As considerable care has been taken to prevent steam from entering the cab, in case of a tube rupture, the damage therefore will certainly be negligible, compared with a crown sheet or arch tube failure in an ordinary locomotive.

The engine is expected to be more economical than the present type of locomotive. The savings are derived from the higher working capacity of the high-pressure steam which can be generated even at a slightly lower expense of fuel than normal pressure steam, since for 1 lb of steam at 250 lb pressure, 1201 Btu are required, against only 1195 Btu at 850 lb pressure. Furthermore, the boiler is designed for steam temperatures at the superheater headers of about 750 F, which also contributes to the increased economy.

Table 1 shows some of the characteristic dimensions of five locomotives of this type. Three of these are for fast passenger service. The New York Central locomotive is designed for fast freight operation, and the Canadian Pacific engine is specifically intended for heavy grade work.

The Canadian Pacific engine 8000 is the largest of the five locomotives which have thus far been constructed, and in which the multi-pressure indirect steam-generating system has been employed.

Fig. 5 is a view looking forward through the firebox to the combustion chamber. The formation of the firebox roof by the alternately crossing riser tubes is shown. The tubular floor of the combustion chamber is also clearly depicted. These floor tubes take water from the front of the firebox ring and deliver it to the combustion-chamber crossheader shown in the background. The steam and water mixture from this header rises through tubes to the separator drums in the same way as from the other riser tubes.

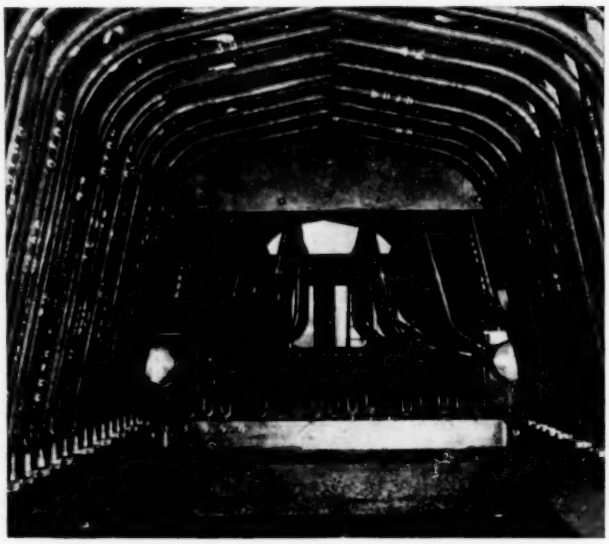


FIG. 5 INSIDE OF FIREBOX, LOOKING TOWARD COMBUSTION CHAMBER

Fig. 6 is a view looking through the combustion chamber to the back head. The combustion-chamber cross-header is shown in the foreground, and the riser tubes, just referred to, are clearly seen. In this view also the front end of the separator drums can be seen. The shutoff valve on the front end of the high-pressure boiler is prominent.

Fig. 7 is a quarter side view of the assembled closed circuit and high-pressure boiler. Attention is called to the condensate

TABLE 1 COMPARISON OF MULTI-PRESSURE LOCOMOTIVES

Road	C.P.R. (Canada)	N.Y.C. (United States)	P.L.M. (France)	L.M. & S. (England)	G.-St. (Germany)
Boiler pressures:					
Closed circuit	1300–1700 lb	1300–1700 lb	1300–1560 lb	1300–1700 lb	1300–1700 lb
High-pressure boiler	850 lb	850 lb	850 lb	900 lb	850 lb
Low-pressure boiler	250 lb	250 lb	200 lb	250 lb	200 lb
Firebox:					
Grate area	77 sq ft	65 sq ft	41.8 sq ft	28 sq ft	26.6 sq ft
Volume	353 cu ft	318 cu ft	236 cu ft	175 cu ft	212 cu ft
Fuel	Oil	Coal	Coal	Coal	Coal
Heating surfaces:					
Closed circuit, fire touched	520 sq ft	430 sq ft	322 sq ft	200 sq ft	218 sq ft
Low-pressure boiler, water touched	3746 sq ft	3229 sq ft	1830 sq ft	1335 sq ft	1370 sq ft
Total evaporating	4266 sq ft	3659 sq ft	2152 sq ft	1535 sq ft	1588 sq ft
High-pressure superheater, steam touched	941 sq ft	835 sq ft	400 sq ft	285 sq ft	323 sq ft
Low-pressure superheater, steam touched	1102 sq ft	1070 sq ft	415 sq ft	385 sq ft	320 sq ft
Total superheater, steam touched	2043 sq ft	1905 sq ft	815 sq ft	670 sq ft	643 sq ft
Total combined	6309 sq ft	5564 sq ft	2967 sq ft	2205 sq ft	2231 sq ft
Heat-transfer coils, inside	750 sq ft	660 sq ft	460 sq ft	295 sq ft	315 sq ft
Boiler dimensions:					
Low-pressure boiler, inside diam	82 in.	80 in.	63 1/4 in.	66 1/8 in.	61 3/4 in.
Low-pressure boiler, length over tube sheets	19 ft 1 3/8 in.	18 ft 2 1/8 in.	16 ft 3 in.	13 ft 2 1/2 in.	13 ft 10 3/4 in.
Low-pressure boiler, no. of flues	214	194	132	130	116
Low-pressure boiler, outside diam of flues	3 1/2 in.	3 1/2 in.	3 1/4 in.	3 in.	3 1/4 in.
High-pressure boiler, inside diam	39 in.	39 in.	37 3/8 in.	36 in.	36 in.
High-pressure boiler, length overall	25 ft 2 in.	23 ft 9 1/2 in.	20 ft 4 1/8 in.	14 ft 11 in.	16 ft 11 in.
Steam separator drums, inside diam	12 in.	12 in.	10 1/4 in.	10 1/4 in.	11 in.
Combustion chamber drums, inside diam	7 1/4 in.	7 1/4 in.	3 in.	3 1/4 in.	3 1/8 in.
Closed circuit tubes, outside diam	2 and 2 1/2 in.	2 and 2 1/2 in.	2 in.	2 in.	2 in.
No. of superheater units—high pressure	49	44	32	24	30
low pressure	61	55	66	32	56
Superheater tubes, outside diam	1 3/16 in.	1 3/16 in.	13/16 in.	1 in.	13/16 in.
Water content:					
Closed circuit	2660 lb	2360 lb	1630 lb	1250 lb	1360 lb
High-pressure boiler	5180 lb	4665 lb	4000 lb	2500 lb	3150 lb
Low-pressure boiler	16500 lb	16750 lb	8900 lb	6775 lb	7500 lb
Ratios:					
Closed circuit heat. sur. ÷ grate area	6.75	6.62	7.7	7.15	8.2
Total evap. sur. ÷ grate area	55.4	56.2	51.4	54.8	59.7
Total heating sur. ÷ grate area	82.0	85.7	70.9	78.8	83.8
Closed circuit heat. sur. ÷ total evap. heat. sur.	12.2	11.3	15.0	13.0	13.7
Heat. trans. coil heat. sur. ÷ closed cir. heat. sur.	1.44	1.61	1.43	1.48	1.45
High-press. suphr. sur. ÷ closed cir. heat. sur.	1.81	2.03	1.24	1.43	1.48
Low-press. suphr. heat. sur. ÷ low-press. heat. sur.	0.295	0.33	0.227	0.288	0.233

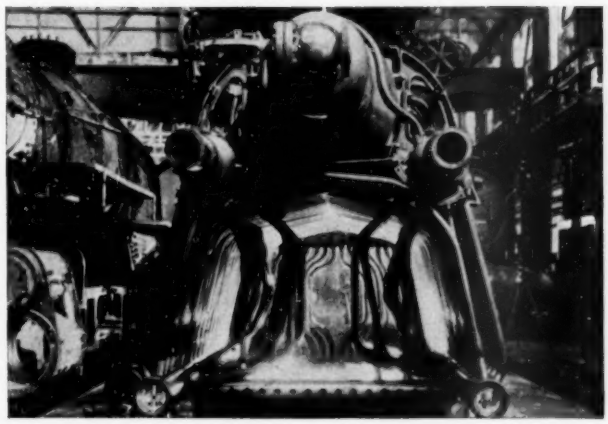


FIG. 6 CLOSED CIRCUIT AND HIGH-PRESSURE BOILER, LOOKING TOWARD THE BACK

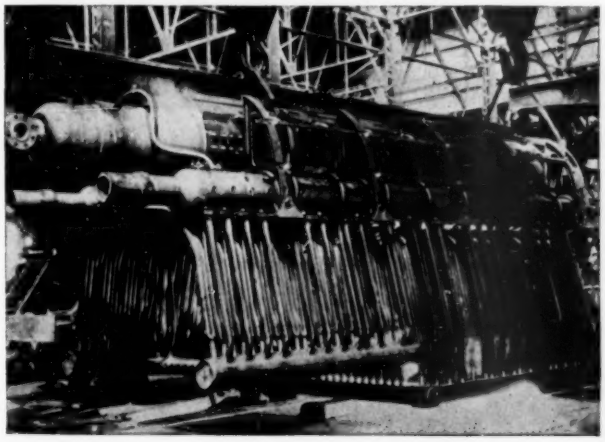


FIG. 7 CLOSED CIRCUIT AND HIGH-PRESSURE BOILER, LEFT SIDE FROM THE FRONT

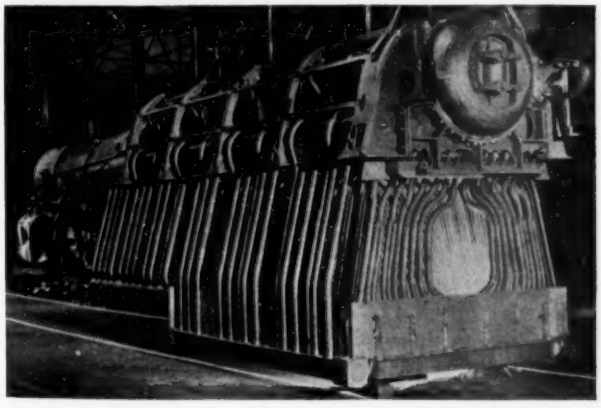


FIG. 8 CLOSED CIRCUIT AND HIGH-PRESSURE BOILER, LEFT SIDE FROM THE BACK

downcomers which convey the closed-circuit water as it leaves the heat-transfer coils, down to the firebox ring, or to the combustion-chamber lower drum, the latter being shown in the central foreground.

The yoke-shaped pipe leading over the front end of the high-pressure boiler is a closed-circuit cross-equalizing pipe connecting the two separator drums and maintains equality of steam pressure in these drums.

Fig. 8 is another quartering view of the closed-circuit and high-pressure boiler showing the back head and also the supports for carrying the back end of the separator drums and high-pressure boiler.

Fig. 9 shows the high-pressure boiler finished and ready to be placed in position. This is a seamless bottle-shaped forging with projections on the side with holes for the reception of the heat-transfer riser tubes which bring the steam from the separator drums to the heat-transfer coils located in the high-pressure boiler. These openings are shown as the upper row of holes on the side. Openings in the row below are for the connections leading the condensate from the heat-transfer coils down to the firebox ring or combustion-chamber lower drums. The openings on the top are for inspection and washout purposes.

Fig. 10 shows the heat-transfer coils, 16 in number, alternately emerging on the right- and left-hand side of the high-pressure boiler. Steam enters these coils through the tube connection, and flowing through the coils downward, becomes condensed and leaves through the lower openings to the downcomer tubes.

The low-pressure boiler is shown in Fig. 11. The smokebox end is in the background. The major portion of the boiler is of conventional shape. The rear end, however, is reduced in diameter and carries a heavy cast-steel member which receives the front ends of the high-pressure boiler and of the separator drums.

Fig. 12 shows the firebox tube sheet of the low-pressure boiler and the saddle casting previously referred to. Heavy caps are applied after the high-pressure boiler and separator drums are in position and securely bolted down, thus providing a firm and rigid attachment between the low-pressure section and the closed-circuit and high-pressure boiler.

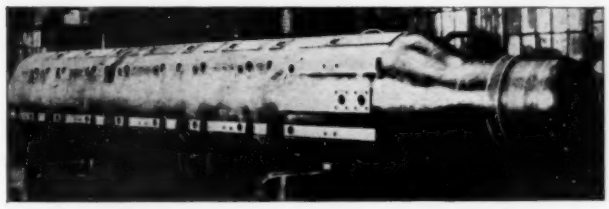


FIG. 9 HIGH-PRESSURE BOILER, RIGHT SIDE

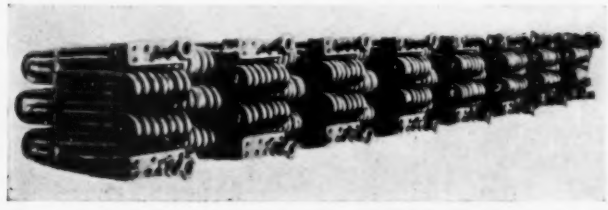


FIG. 10 HEAT-TRANSFER ELEMENTS

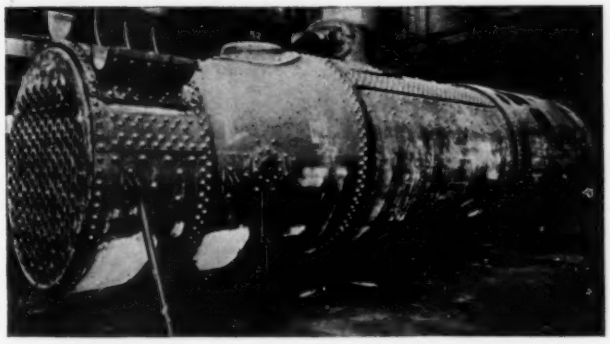


FIG. 11 LOW-PRESSURE BOILER, RIGHT SIDE

Fig. 13 shows the smokebox after the superheater, steam pipes, stack extension, and exhaust pipe have been applied. The vertical pipe just to the left of the center line is the high-pressure steam pipe from the throttle to the high-pressure chest. The high-pressure exhaust pipe is shown on both the right and left side, and carries steam upward and outward to the mixing chamber, where the steam enters the low-pressure steam pipe. The low-pressure steam pipes are shown in the conventional position leading from the superheater header on both the right and left side downward to the low-pressure steam chests.

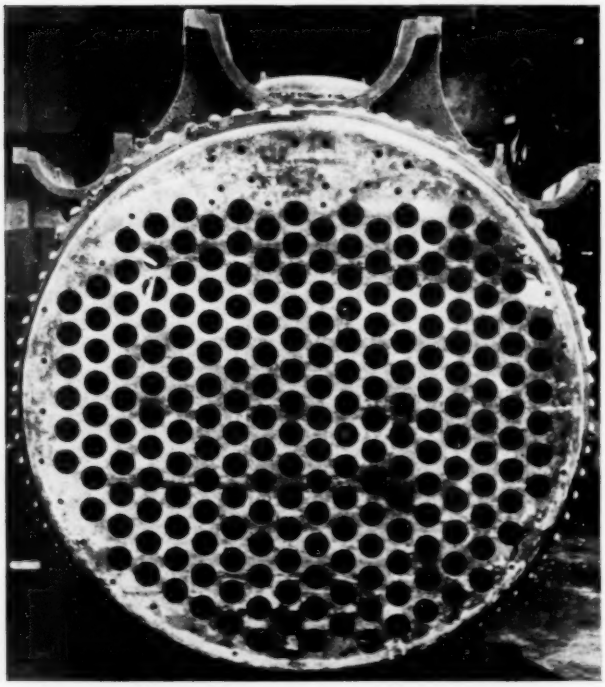


FIG. 12 FIREBOX TUBE SHEET OF LOW-PRESSURE BOILER

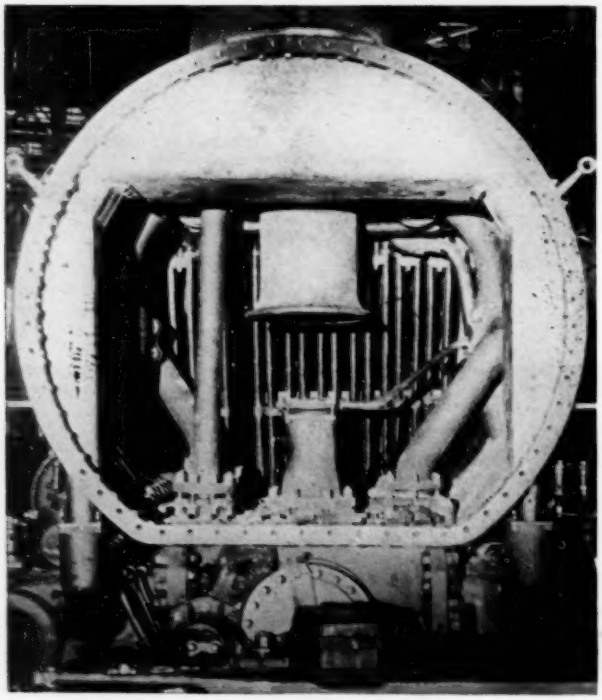


FIG. 13 SMOKEBOX

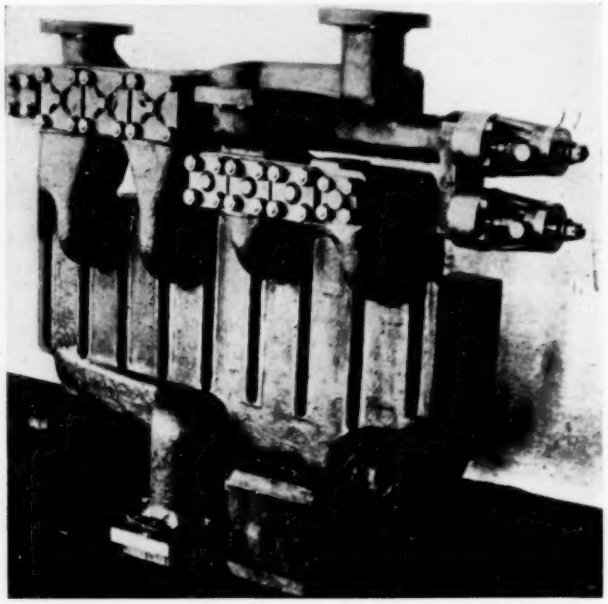


FIG. 14 COMBINED SUPERHEATER AND MULTIPLE-VALVE THROTTLE HEADER, TOP VIEW

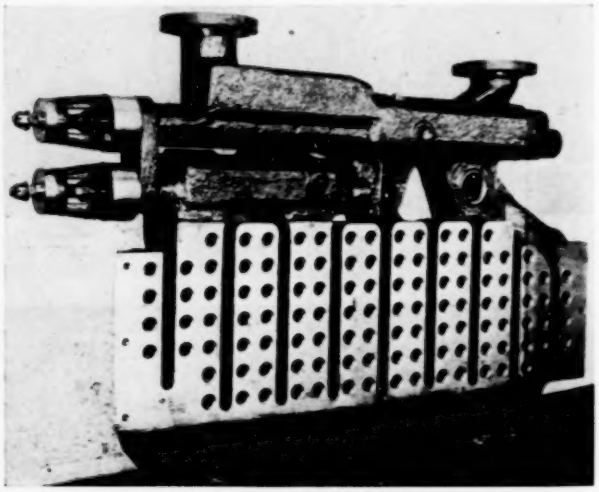


FIG. 15 COMBINED SUPERHEATER AND MULTIPLE-VALVE THROTTLE HEADER, BOTTOM VIEW

The superheater header is shown in Fig. 14. This is a top view with the low-pressure steam connection at the extreme bottom conveying steam into the low-pressure section, which is at the left as one views the picture. The high-pressure steam is led in through the opening at the lower right-hand part of the casting. The low-pressure throttle is at the upper left-hand corner and consists of a three-valve multiple throttle of conventional design. The high-pressure throttle is also a three-valve multiple design generally of the conventional form, but utilizing smaller valves on account of the greater density of the high-pressure steam.

Fig. 15 is a view of the lower side of this combined double superheater header and multiple-valve throttle casting.

Canadian Pacific Multi-Pressure Locomotive

Its Machinery, Locomotive Chassis, and Tender

By J. B. ENNIS,[1] NEW YORK, N. Y.

The remarkable thermal efficiency of the multi-pressure locomotive of the German State Railways has commanded the attention of the entire railroad world. The introduction of this type of locomotive to American practice involves far more than the mere reproduction of the German machine, for the difference between the problems of availability, maintenance, and proportionate output on the two continents is profound. The author gives the reasons for the adoption of the type of engine selected to introduce the multi-pressure principle to American service, and describes the leading features of the construction of the chassis and cylinders.

THE steam-generating system of the Canadian Pacific multi-pressure locomotive has been described. This paper will tell the manner in which the design of the machinery and of the locomotive as a whole has been worked out to utilize this system. It was the intention from the start to deviate as little as possible from conventional lines of locomotive design, and to use as many parts of the Canadian Pacific 2-10 4, two-cylinder locomotive, class T-1, as practicable. This locomotive, built by the Montreal Locomotive Works in 1929, is the most powerful freight locomotive of the Canadian Pacific Railway; its total weight is 370,000 lb, its weight on drivers is 312,800 lb, and its tractive effort is 77,200 lb without booster. The multi-pressure locomotive was supposed to be not less, and if possible more, powerful than locomotive T-1, without increasing the number of wheels. Therefore the requirement to adhere in general to the design of the T-1 locomotive was natural.

In two major respects, however, novel features had to be introduced: in the locomotive engine and in the structural connection of the boiler and locomotive chassis.

Regarding the locomotive engine, it was estimated that 63,000 lb of steam would be furnished, of which 56.0 to 47.6 per cent would be high-pressure steam of 850 lb per sq in. and the remainder of 250 lb pressure. Both high- and low-pressure steam were supposed to be superheated, and it was expected that the total temperature of the first would be between 600 and 800 F, while that of the latter would fluctuate between 550 and 725 F, depending upon the rate of firing.

In order to utilize the heat contained in the steam in the most efficient manner, it is necessary to use double expansion (compound) for the high-pressure steam; and the most practical combination of the two-stage expansion with one-stage expansion of the low-pressure steam, generated separately, is the one applied in the Schmidt-Henschel locomotive of the German Reichsbahn—namely, the mixing of the exhaust from the first stage of the high-pressure expansion with the low-pressure live and superheated steam, using this mixture for the second stage of the expansion. By doing so, the necessity of interstage heating for the high pressure is eliminated, resulting in simplification of the locomotive.

It is obvious that more than two cylinders are necessary for this scheme. The cross-compound system is not possible in view of the excessive size of the single low-pressure cylinder for a locomotive with 320,000 lb weight on drivers. Either three or four cylinders are necessary, and then several cylinder arrangements could be considered, such as:

(1) A tandem compound arrangement, having the advantage of the outside location of all four cylinders without the disadvantage of a crank axle, the drawback, however, being heavy reciprocating parts, resulting in poor balancing, the inaccessibility of piston-rod packings between cylinders, and the undesirable lengthening of the front part of the locomotive, with ensuing increase in weight.

(2) A four-cylinder compound arrangement, such as used in the high-pressure locomotive of the French Paris-Lyons-Mediterranean Railway. This would require a two-crank axle, which, in view of the power of the locomotive, it would be impossible to construct in the limited space between the frames.

(3) A three-cylinder arrangement with a single-crank axle, as used in the Schmidt-Henschel locomotive, with a middle high-pressure and two outside low-pressure cylinders.

(4) A three-cylinder arrangement, differing from the foregoing by making one inside low-pressure cylinder between frames and two outside high-pressure cylinders. This also would require too large a cylinder, which it would be impossible to place inside the frames.

Thus of all the four arrangements, the most practical seemed to be the third; and although this necessitated the use of a crank axle, it was thought that as the power on the middle pin is less than in the three-cylinder 4-12-2 Union Pacific locomotive, a satisfactory three-cylinder arrangement could be worked out.

Having decided on this, the next question was the distribution of power. The most natural thing to do would be to divide the power equally between cylinders. It was not so much the division of tractive effort at starting as the proper division of power under running conditions, for the over-all efficiency would be more affected by the latter than by the former. On the other hand, it was rather desired to get slightly less tractive power from the middle cylinder so as to protect the middle main pin.

The three-cylinder arrangement with two working pressures offers a very convenient means for equalizing power without exceeding stresses in the middle pin, by introducing a longer cut-off in the middle cylinder. By doing so, the size of the cylinder can be made smaller, and thus the piston thrust in the middle cylinder can be kept within limits. However, this would mean either introducing a complication in the valve gear or using an independent valve motion for the middle cylinder, whereas from the start it was agreed to use the Gresley valve motion, as in the majority of three-cylinder locomotives built in this country. In view of this, it was decided to make the cutoffs in all three cylinders equal by using the Gresley gear, and to compromise between division of power at starting and under running conditions in such a way as to keep the diameter of the middle cylinder as small as possible.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

A study of indicator cards for different cutoffs and various cylinder sizes was made, in connection with service conditions and boiler capacities, and it was found that the ratio of volume of the two low-pressure cylinders to that of the high-pressure cylinder should be 5.4. This determined the following dimensions for cylinders: $15^1/_2$ in. in diameter and 28 in. stroke for the middle high-pressure cylinder, and 24 in. in diameter and 30 in. stroke for the two outside low-pressure cylinders.

On the basis of 850-lb and 250-lb pressure in the two boilers, 63 in. diameter of driving wheels, 85 per cent coefficient in the tractive-effort formula, and making allowance for the areas of the high-pressure piston rod and piston-rod extension, the tractive effort was estimated to be 83,300 lb, corresponding to 3.84 adhesion factor. A study of tangential efforts in this three-cylinder locomotive proved that this factor should not cause slipping of the locomotive.

In view of some uncertainty in distribution of power under actual conditions which this experimental locomotive will have to

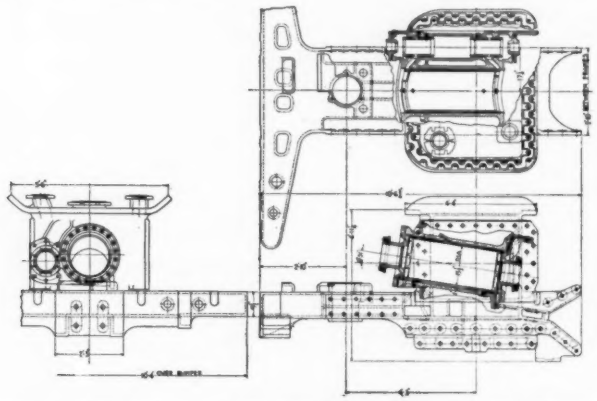


FIG. 1 HIGH-PRESSURE CYLINDER WITH BUMPER AND BUMPER BRACKET

meet, provision was made for the possibility of a slight change in the sizes of cylinders by making bushings in the cylinders of sufficient thickness. A 1-in. bushing was made for the high-pressure cylinder, which would permit of enlarging the cylinder diameter from $15^1/_2$ in. to 16 in., if this should be required in the future. The low-pressure cylinders were provided with $1^1/_4$-in. bushings, permitting an increase in diameter to $24^3/_4$ in.

CYLINDERS

The middle cylinder is shown in Fig. 1. It is of the same type as used on ordinary three-cylinder locomotives, with cylinder barrel, steam chest, and boiler saddle cast in one piece. In this case the front bumper and bumper bracket are cast integral with the cylinder.

Special precaution has been taken to make the bolting of the cylinder as rigid as possible. A great number of bolts and two $1^1/_4$-in. by 3-in. keys are provided for the connection of the cylinder casting to the frames, outside cylinders, and smoke box. In addition, keys 3 in. wide are used between the middle and outside cylinders for securing the cylinders in longitudinal direction.

The cylinder is made of cast steel to a special specification, with silicon content of 0.40 per cent and manganese of 0.82 per cent. The cylinder barrel is inclined at an angle of 8 deg 31 min to the horizontal in order to permit the middle main rod to clear the front driving axle, the second being the crank axle. The cylinder bushing is made of high-grade cast iron of a composition insuring good wearing qualities. The steam chest is horizontal, as is usually the case in three-cylinder locomotives with Gresley valve motion, and has two high-grade cast-iron bushings for the high-pressure 6-in. piston valve. Steam is admitted between the piston-valve heads (inside steam admission) through a $4^1/_2$-in. steam pipe. The exhaust is directed through two 5-in. openings, one on each side of the cylinder, into the two mixing chambers, where it meets the low-pressure superheated steam and whence it is further directed into the two low-pressure cylinders.

The exhausts from the outside cylinders have to pass through the middle cylinder into the exhaust pipe and nozzle. A rectangular $5^1/_2$-in. by 10-in. opening is provided on each side of the middle cylinder in the fitting surface. In order to insure tightness between the middle and outside cylinders, special copper-wire interwoven asbestos gaskets are used. Provision is made for passing the booster exhaust through a specially made passage in the saddle, and further into the exhaust pipe, in case of the future application of a booster. The openings for the booster-pipe connections are for the present blocked off by welded-in plates.

The outside cylinders are of regular design, with high-grade cast-iron bushings $1^1/_4$ in. thick, as already stated. The cylinders are made of manganese-silicon cast steel of a composition similar to that of the high-pressure cylinder.

CYLINDER HEADS, PISTONS, ETC.

High- and low-pressure cylinder heads and steam-chest covers are made of cast steel, except the front low-pressure steam-chest covers, which are of cast iron. The high-pressure cylinder heads and steam-chest covers are secured to the cylinder by chrome-nickel studs.

The high-pressure piston, in view of its comparatively small size, is of the solid-head type, as shown in Fig. 2. It is made of cast iron to special specification. The width of the piston is 8 in., and there are six snap packing rings of cast iron, $^1/_2$ in. wide and $^5/_8$ in. thick. Experience with German high-pressure loco-

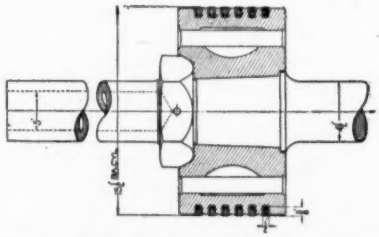


FIG. 2 HIGH-PRESSURE PISTON AND ROD

motives had shown that, for proper packing of high-pressure steam, a large number of narrow rings is necessary.

It is common practice in America to omit the extension piston rod, as this is not needed for proper guiding of the piston. While this omission would be possible also in the high-pressure cylinder, it was thought that, for better equalization of pressures on both sides of the piston, it would be more advisable to have the piston rod extended in front of the piston, with the same outside dimension of $4^1/_2$ in. as in the back portion. The front extension has a hollow boring of 3 in. diameter in order to save weight. The piston rod is forged of medium-carbon steel. It is comparatively short, having a length of $66^1/_2$ in., counting from the middle of the piston fit to the center of the crosshead wristpin.

The crosshead is of the common single-guide, three-bar type (Dean design), made of cast steel, with a long cast-steel shoe.

Pistons for the outside cylinders are of the regular Z-type, made of cast steel, have no extension rods, and are connected to alligator-type crossheads. Each piston has a cast-iron 6-in.-wide bull ring, with two narrow $^3/_8$-in. by $^3/_4$-in. cast-iron snap packing rings.

Rod Packings

The original high-pressure piston-rod packings were of special German design, as used in the Schmidt-Henschel locomotive of the Reichsbahn. In principle it is the standard German packing used on regular locomotives, differing from that only in the number of rings, six being used instead of three for each stuffing box. It was found necessary to replace this packing by that of the "King" type.

Rods

Out of a number of middle main rods tried on three-cylinder locomotives in America, a light and simple rod was chosen, as shown in Fig. 3. The body of the rod is integral with the lower portion of the strap, the upper being bolted to the rod by one $2^{3}/_{4}$-in. bolt in the front and three $1^{3}/_{4}$-in. bolts in the back. A floating bushing, made of bronze in three pieces, fits the 12-in. middle pin of the crank axle with small radial and side clearances between the crank-axle cheeks. The crosshead end of the middle main road has adjustable bronze bearings.

The outside main rods, connected to the third or main axle and all side rods are of the regular C.P.R. design, the low-pressure main rod having adjustable bronze bearings at the front end and floating bushings at the back end, revolving in a stationary solid-steel bushing. Side rods have floating bronze bushings in stationary cast-iron bushings on the main and intermediate pins and solid bronze bushings on the front and back pins. All main and side rods are of I-section and are made of nickel steel.

Valves and Valve Motion

The high-pressure steam valve is of the plug type, having two

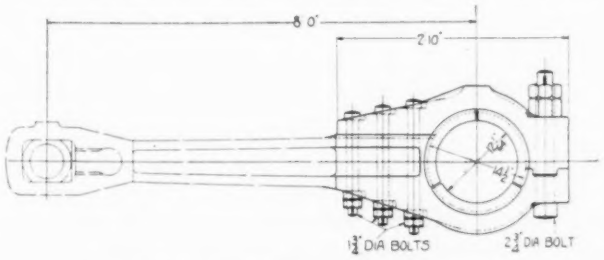


Fig. 3 High-Pressure Main Rod

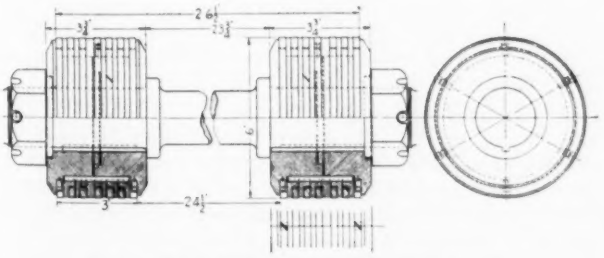


Fig. 4 High-Pressure Distribution Valve

heads, each made of two halves, with six cast-iron bull rings and seven narrow cast-iron snap packing rings, all assembled on a common forged-steel valve stem with an extension (Fig. 4). The low-pressure valve is of the common spool type, assembled on a valve stem with extension.

The valve-stem packings, both for the high- and low-pressure valves, are of the King type, which is standard on the C.P.R. The high-pressure valve-stem packing has no contact with high-pressure steam in view of the valve being of the inside-admission type.

The valve gear for the outside cylinders is of the Walschaert type and gives $7^{1}/_{2}$-in. valve travel. The valve motion for the inside cylinder is derived from the two outside gears by the combined Gresley valve motion frequently used on three-cylinder locomotives in America. The Gresley levers have ball bearings in the main fulcrum and in the connection between the two levers. The gear is shown in Fig. 5. As already stated, the travel and cutoffs are the same in all cylinders.

The reverse is of the screw type, which is standard on the C.P.R. No power reverse is used.

Firebox Framing

It already has been mentioned that the second novel feature introduced in the design is the connection between the boiler and

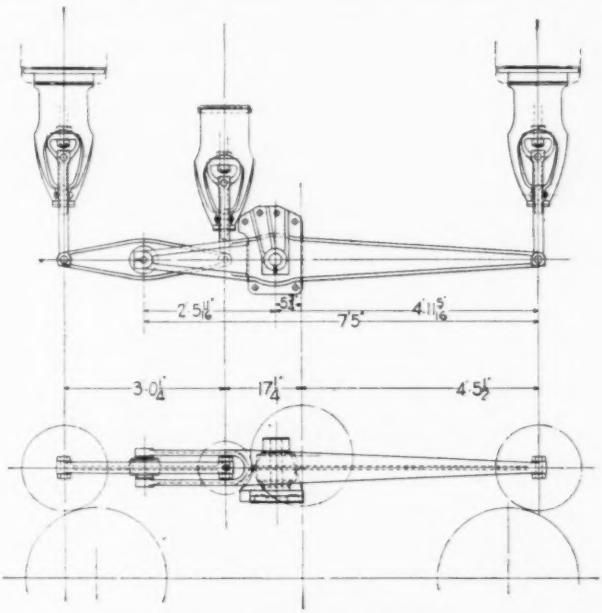


Fig. 5 Gresley Valve Gear

the locomotive frame. In the ordinary locomotive the boiler forms the backbone of the locomotive structure. The boiler barrel is rigidly bolted to the cylinders at the front of the locomotive, while the firebox, which is a part of the boiler, is connected with the rear of the frame. Flexibility is allowed in the conventional structure in the longitudinal direction, permitting free expansion of the boiler, while sidewise the boiler is rigidly held by the frames.

In the multi-pressure locomotive the low-pressure boiler is in principle of the same design as the conventional boiler. The high-pressure boiler in itself, however, does not offer sufficient rigidity to the locomotive structure. It was therefore necessary to enclose the firebox and the combustion chamber in a special framing which would form a rigid connection between the boiler and the locomotive frame.

Moreover, a water-tube firebox requires an airtight casing which will not permit outside air to enter the firebox, except the air that is admitted for proper combustion through the ashpan. These two aims have been attained in the following ways:

Support *A* (Fig. 6), with semi-circular brackets for the high-pressure boiler and separator drums, is riveted to the rear portion of the low-pressure boiler. The drums are kept in position by rigidly bolted caps *B* and *C*. The rear parts of the drums, with support *D* bolted to them, can freely slide on the upper part of the structure. Shoes *N* and rollers *O* are provided for this purpose.

On the under side of the low-pressure boiler, opposite support

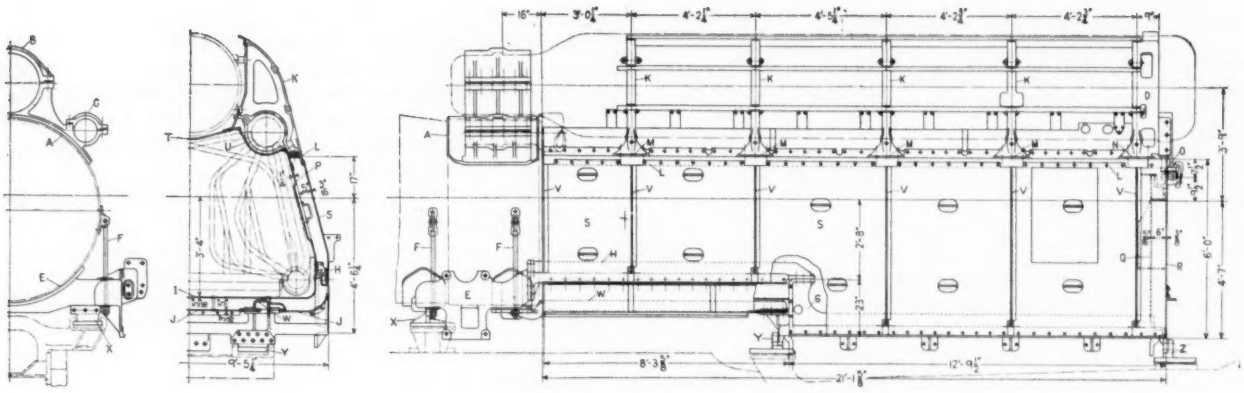


FIG. 6 FIREBOX FRAMING

A, a saddle *E* is bolted, which is in addition suspended by tierods *F*. Supports for the high-pressure pump and airbrake pump are built integral with the boiler saddle. The firebox ring is surrounded by a rectangular casting *G* called the foundation ring,

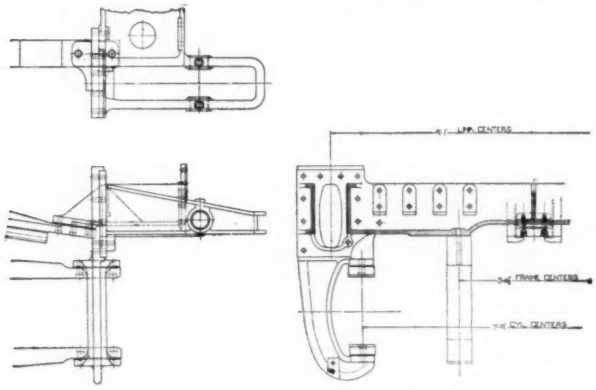


FIG. 7 GUIDE-YOKE AND LINK-SUPPORT CROSSTIE

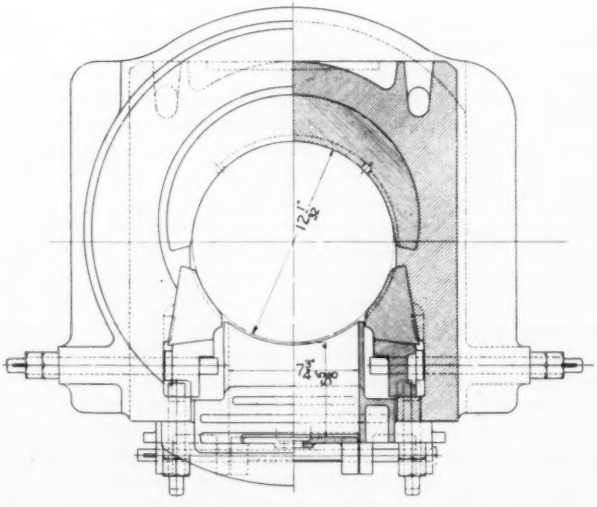


FIG. 8 DRIVING BOX, CRANK AXLE

which is connected with saddle *E* by two side-support members *H*, one on each side of the firebox. The rear end of the foundation ring has a built-up vertical wall for slides *N* and rollers *O* of support *D*.

Thus a skeleton of a substantially rectangular box has been built around the firebox and combustion chamber, rigidly bolted to the rear end of the low-pressure boiler. The whole box, the remaining parts of which are described farther on, is supported by the frame and can freely slide on it, permitting the boiler as a whole to expand longitudinally. The following parts complete the box:

Eight castings *K*, four on each side, are bolted to the high-pressure drum and embrace the separator drums. They can freely slide on two support members *L*, one on each side of the firebox, with slide plates *M* in between. Side supports *L* are further connected by vertical columns *V* to side members *H* and foundation ring *G*. Between these supports, on each side of the locomotive, a vertical wall is built of two rows of removable steel plates *P* and *S*, the inner plates *P* made of heat-resisting steel. The vertical wall for support *D*, already referred to, is also made of ordinary steel on the outside and heat-resisting steel plates on the inside.

In addition, two tiebars *W* are bolted to boiler saddle *E* and foundation ring *G*. Three expansion slides permit this structure to expand along the frame; one slide *X* is bolted to the boiler saddle *E;* another slide *Y* is bolted to the front lug, while the third slide *Z* is attached to the rear lug of foundation ring *G*.

The foregoing structure, needed for carrying the weight of the high-pressure boiler, for permitting its expansion, and for imparting sufficient rigidity to the boiler as a whole, is further completed by several plates in order to make it airtight. Plates *T* and *U* of heat-resisting steel are placed underneath the high-pressure and separator drums and tied by angles between themselves and plates *P*. Heat-resisting plates *I* and ordinary steel plates *J* bolted to members *H* and tiebars *W* finish the encasing of the firebox.

High-pressure and separator drums are insulated by high-temperature lagging, and approximately half of the space between heat-resisting plates *P* and *Q* and steel plates *S* and *R*, next to the heat-resisting plates, is filled with the same lagging. The low-pressure boiler is lagged with magnesia blocks and jacketed in the ordinary manner.

FRAME

The general design of the locomotive chassis, including frames, wheels, brake rigging, etc., is similar to that of the Canadian Pacific 2-10-4 locomotive, class T-1, except for the parts that were affected by the difference in the type of boiler and machinery. As a matter of fact, a great number of parts and the tender as a whole are practically identical.

The locomotive frame, however, for this experimental engine was not a locomotive bed, as in the T-1 class, but of the built-up type, consisting of two frames (right and left) $6^1/_2$ in. thick, made of nickel cast steel. The distance between frame centers is $40^1/_2$ in. crosswise. The frames are rigidly bolted to the sides of the

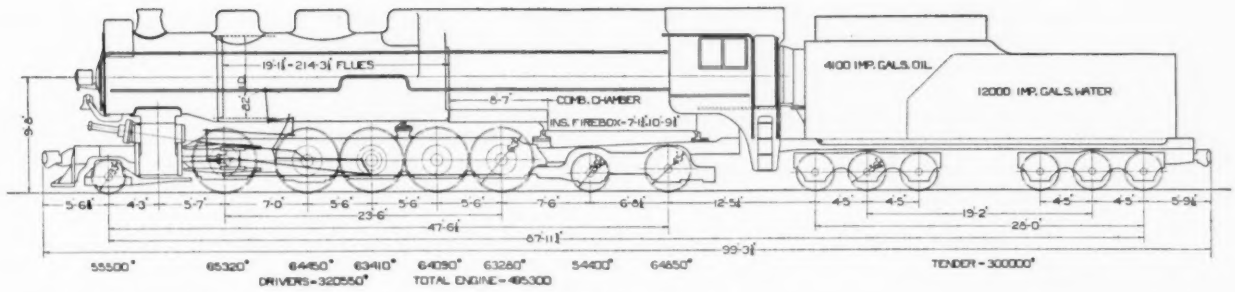


FIG. 10 WEIGHT DIAGRAM

middle-cylinder casting. The outside cylinders are secured to the frames by bolts, some of which pass through both cylinders and frame.

As has already been stated, the middle cylinder is integral with the front bumper and bracket. The frames behind the cylinders are connected by a number of crossties, as in all built-up locomotive frames. They are all made of cast steel.

The guide-yoke crosstie is of a special design, in view of the attachment of the middle crosshead guide. It is also made integral with the link support and is therefore called "link support crosstie." The guide yokes for the outside guides are bolted to this crosstie, as shown in Fig. 7.

The pedestal caps for the second driving axle, which is the crank axle, are of cast steel, while the other pedestal caps are made of forged steel.

All frame fillings for the spring rigging are made of cast steel. Shoes and wedges are made of cast iron.

TRUCKS AND BOXES

Driving boxes are made of cast steel and are equipped with cast-bronze wearing faces and bronze bearings. The crank-axle driving boxes (front intermediate) are of the Alco design, with supplementary adjustable brasses (Fig. 8). The boxes on the main (third driving axle) and on the back intermediate axle are of the Grisco type, with permanent additional brasses. The front and back driving boxes are of the regular design, with bronze bearings.

The locomotive is designed to negotiate 18-deg curves. For this purpose the front axle has the Alco lateral-motion device, with a play of 1 in. on each side. The front engine truck is of the Commonwealth two-wheel, cast-steel, outside-bearing type, with $7^1/_4$-in. lateral movement, while the trailing truck is of the four-wheel, cast-steel, Commonwealth type, with the $10^1/_4$-in. lateral displacement necessary for 18-deg curves. All truck boxes are of cast steel and are equipped with bearing brasses.

SPRINGS

The spring rigging is of the three-point suspension type, the front engine truck forming one group with the first three driving axles, while the right and left rear drivers, with the wheels of the trailing truck, form the second and third groups. All suspension springs are of the semi-elliptic design, made of steel with 0.65 to 0.75 per cent carbon content, with steel equalizers and case-hardened pins in case-hardened bushings. All spring shoes are made of cast steel, while spring saddles are of nickel cast steel.

WHEELS AND AXLES

All locomotive wheels have separate wheel centers made of cast steel and shrunk-on steel tires with retaining rings. All straight axles are made of carbon steel.

The middle pin and axle ends are made of nickel steel. The two disks, sometimes called cheeks, are forged of carbon steel. The middle pin has a diameter of 12 in. and is 7 in. long. The journals of the crank and other axles are as follows:

Crank axle, main axle, back intermediate, in.	12×14
Front and back, in.	$11^1/_2 \times 14$
Engine truck, in.	7×14
Trailing truck, front, in.	7×14
Trailing truck, back, in.	8×14

Following the example of the German Reichsbahn locomotive, the cranks were spaced at approximately 120 deg, this for the sake of torque uniformity. As was expected, this crank setting resulted in an uneven exhaust sound. The engine, being of the

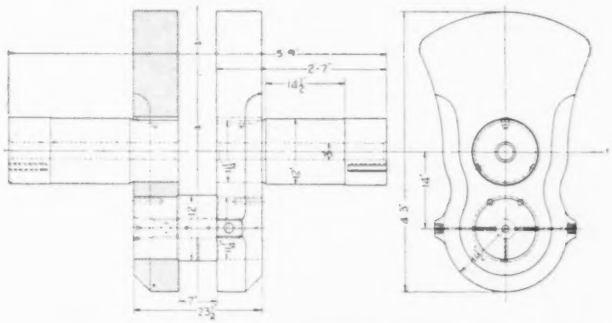


FIG. 9 CRANK AXLE

compound type and having only four exhausts per revolution, sounds "lame" at low speeds, but at speeds above 15 mph the un-uniformity of the exhaust vanishes.

The crank setting is of course corrected for the inclination of the middle cylinder and for the difference in strokes of the high- and low-pressure cylinders. For the middle crank it is 129 deg 12 min with the left-hand crank and 110 deg 48 min with the right-hand crank, the right being the leading one. The angle between the two outside cranks is exactly 120 deg. The crank axle is shown in Fig. 9.

All straight axles are hollow-bored. The crank-axle ends, middle pin, and all crankpins are also hollow-bored. All crankpins are made of carbon steel except main crankpins, which are of nickel steel. All journals and crankpins are ground.

The counterbalances for the reciprocating weights are distributed evenly between all driving wheels except the main. The total percentage of counterbalancing of the locomotive, figured by the static method, is 34.75 per cent. The counterbalance pockets on all wheels, following the C.P.R. practice, are filled with lead, on the main wheels to the limit of their volume capacity.

BRAKES

All driving and trailer-truck wheels of the locomotive have brakes operated by a Westinghouse automatic-brake equipment, with $8^1/_2$-in. cross-compound air compressor. The driving-wheel

brakes are divided in two sections, the front section operating the three front axles from two 14-in. by 10-in. cylinders, while the back portion operates the two rear pair of drivers from two 12-in. by 10-in. cylinders.

The trailer truck has a separate brake applied to all wheels, using two 8-in. by 12-in. cylinders.

Three main reservoirs are used on the locomotive, with a total capacity of approximately 80,000 cu in.

Cab and Auxiliaries

All steam auxiliaries are operated by low pressure, except the two high-pressure pumps, which are operated by high-pressure steam. There are two cab turrets—one for superheated and one for saturated steam. The first has connections to the high- and the low-pressure feed pump, the air pump, the blower, the oil-burner operating valves, the mechanical lubricator-heating coil, the headlight dynamo, and the whistle. The second turret has valves to the steam-heating line and to the injector.

The cab has sufficient inside room and does not differ much from an ordinary cab. The arrangement of apparatus in the cab is the same as usual, except only for those gages, pyrometers, handles, etc. which are necessary for the high-pressure boiler. The low-pressure water column and gage have connections which extend from the low-pressure boiler into the cab; the column and water gage in the cab are of the ordinary construction.

The top and bottom cocks of each of the two high-pressure water gages in the cab can be shut off simultaneously. The cocks of the closed-circuit water gage placed outside the cab can be likewise closed by a handle in the cab.

Drawbar

The drawbar between engine and tender is of the Unit safety type, both bars of laminated C.P.R. construction and made of nickel boiler steel. There is a radial buffer between engine and tender.

Tender

As previously stated, the tender is of the standard C.P.R. type, and in all respects is identical with that used for the C.P.R. 2-10-4, two-cylinder, class T-1 engine. The frame is of the cast-steel Commonwealth water-bottom integral type, with six-wheel Commonwealth trucks. The truck boxes are of malleable iron, while the wheels are of the solid rolled-steel type.

The tank is of the rectangular type, having a water capacity of 11,500 imp. gal and a maximum oil capacity of 4100 imp. gal. The tender tank has a recess in which the low-pressure feedwater-heater pump is located.

The tender brakes are of the conventional clasp type, operated from two brake cylinders 14 in. by 12 in., with Westinghouse distributing valve, feed valve, and other equipment. The brake is designed for 80 per cent of the sum of the light weight of the tender and 25 per cent of the load.

Lubrication

The following locomotive parts are lubricated by oil from a 14-feed mechanical lubricator: middle and outside crosshead shoes through the corresponding guides, the middle guide delivering oil also to the wristpin and front end of the middle main rod, high-pressure and low-pressure steam chests and cylinders, right-hand high-pressure and low-pressure feed pumps and air pump, and high-pressure piston-rod packings. The left-hand high-pressure feed pump is served by a hydrostatic lubricator.

All driving boxes, including those of the crank axle, have regular cellars with perforated plates for grease lubrication. Engine and tender truck boxes are lubricated by oil and waste. Outside main rods, middle main rod, and side rods have hard-grease lubrication from Alemite-type cups. The remainder of locomotive and tender parts, such as valve motion, driving and truck-box pedestals and hub faces, all engine and tender-truck center plates, engine center pin, lateral-motion device, radial buffer between locomotive and tender, etc. have soft-grease lubrication.

It is evident from the foregoing description that, aside from the high-pressure feature, the locomotive does not differ from the conventional engines. As stated at the beginning, the aim was to introduce no more novelties than absolutely necessary, to retain the general appearance of an ordinary locomotive, and to make possible the use of the methods of operation of the steam locomotive to which generations of railroad men have become accustomed.

Fig. 10 shows the weight diagram and the general outline of locomotive and tender. In Table 1 are given the principal dimensions of the locomotive, which bears road number 8000, and engines of class T-1.

TABLE 1

	C.P.R. multi-pressure locomotive 8000	C.P.R. locomotive class T-1
Wheel arrangement	2-10-4	2-10-4
Weight total, lb	495,300	452,500
Weight on drivers, lb	320,550	312,800
Weight of tender, loaded, lb	300,000	300,000
Cylinders, high-pressure, diameter and stroke, in.	(1) 15½ × 28	...
Cylinders, low-pressure, diameter and stroke, in.	(2) 24 × 30	(2) 25½ × 32
Diameter of wheels, driving, in.	63	63
Diameter of wheels, front truck, in.	33	33
Diameter of wheels, trailer, in.	36¼ × 45	36¼ × 45
Boiler pressure, closed circuit, lb per sq in.	1,350	...
Boiler pressure, high, lb per sq in.	850	...
Boiler pressure, low, lb per sq in.	250	275
Heating surface, closed circuit, sq ft	520	...
Heating surface, heat-transfer coils, high-pressure, sq ft	750	...
Heating surface, low-pressure, sq ft	3,746	4,928
Superheating surface, high-pressure, sq ft.	941	...
Superheating surface, low-pressure, sq ft.	1,102	2,112
Grate area, sq ft	77	93.5
Tender tank, fuel capacity, imp. gal	4,100	4,100
Tender tank, water capacity, imp. gal.	11,500	12,000
Tractive effort, lb, rated	83,300	77,200
Tractive effort, lb, actual, maximum	90,000	...

Canadian Pacific Railway Multi-Pressure Locomotive 8000

Its Construction, Operation, and Maintenance

By H. B. BOWEN,[1] MONTREAL, QUE., CANADA

Higher steam pressures and temperatures have marked the progress toward greater power at reduced cost. Locomotive development has not kept pace with stationary practice in the use of higher pressures. After studying all available information on new types of locomotive and after investigating improved types of locomotive abroad, the Canadian Pacific Railway Company decided to construct a heavy freight locomotive for service on its severe mountain divisions. Its 2-10-4 multi-pressure locomotive 8000, which is described in the paper, was the result.



INCREASING steam pressures and temperatures have to a great extent been the measure of the progress made in the development of greater power at reduced cost. While locomotive development has made marked advances, it has nevertheless lagged behind stationary practice in utilizing the manifest advantages of higher pressures. Conventional locomotive design, with a large-diameter fire-tube boiler construction, definitely restricts the maximum pressure that can be used, and the Canadian Pacific Railway Company has gone nearly as far as practical in the increasing of steam pressures with the conventional form of locomotive-boiler construction.

All obtainable information concerning new types of locomotives had been studied, and such designs as had been transformed into reality had been investigated thoroughly. In 1929, the author, during a trip to Europe, investigated carefully all of the available examples of new and improved types of locomotives. Conferences with the Superheater Company and its associate organizations, both in the United States and abroad, resulted in further investigations being made of locomotives in England and on the Continent which were constructed and which utilized the multi-pressure system of steam generation. Conferences with officials of the German State Railways, the London, Midland & Scottish in England, and the Paris, Lyons & Mediterranean Lines in France, had served to focus attention upon this type of steam-generating apparatus. Subsequently, further conferences with the American Locomotive Company and the Superheater Company resulted in the decision by the railroad to construct a heavy freight locomotive suitable for service on its severe mountain divisions; and its 2-10-4 type locomotive 8000, which has been described, is the result. This locomotive is termed the T-4-A class and is, so far as its running gear is concerned, a close duplicate of the T-1 class locomotives which are now the standard locomotives for the service referred to. While the T-4-A is slightly heavier as to total weights, it is nevertheless very closely the same as the T-1's in so far as weight on drivers is concerned.

The three-cylinder arrangement has permitted satisfactory operation with a somewhat lower factor of adhesion, and consequently has a somewhat larger tractive effort. High pressures and temperatures are not only attractive from the standpoint of improved thermodynamic performance, but in themselves bring about desirable changes in construction and performance. For example, higher pressures necessitate multi-cylinder construction in order to fully utilize the steam expansively. This immediately gives a more uniform torque, and permits lowering the adhesion factor between the driving wheels and the rails, which in turn provides for the better utilization of weight. Higher pressures give a better steam flow and reduce proportionate pressure drops. It is customary to allow a 15 per cent drop in boiler pressure in figuring the maximum tractive effort. This loss, however, is reduced as the pressure rises, giving greater proportionate tractive efforts. Higher pressure means smaller cylinders, removing some of the restrictions due to large cylinders as determined by customary boiler pressures. The higher density also provides for the use of smaller pipes for feeding both cylinders and auxiliaries, which is a most grateful relief to those who have struggled in trying to fit in the maze of complicated piping required by even the comparatively simpler conventional type of locomotive.

Multi-pressure locomotive 8000 is the joint production of the Superheater Company, the American Locomotive Company, and the Canadian Pacific Railway Company, which are respectively responsible for the high-pressure steam-generating system, the three-cylinder arrangement and valve motion, and the general locomotive proportions and design and construction. It was built at the Canadian Pacific Railway's Angus Shops, Montreal, and was turned out for operation in July, 1931. During the ensuing few months the engine was used in freight service between Montreal and Smith Falls, in order to determine its operating characteristics and particularly to develop the arrangement of oil burners, which it was anticipated would have to be somewhat different from the standard burner arrangement used on the Western Lines.

Construction was commenced in November, 1930. The machinery, together with the low-pressure boiler and firebox framework, was erected in a complete unit, while the closed system and high-pressure boiler were assembled on a jig especially constructed for this purpose in the boiler shop. Work on these separate units progressed simultaneously, and on April 10, 1931,

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[1] Chief of Motive Power and Rolling Stock, Canadian Pacific Railway Company. Henry Blane Bowen was born in Derbyshire, England, May 17, 1884. He was educated in the Manchester School of Technology and École de Commerce, Lausanne, Switzerland. He served a special apprenticeship at Crossley Bros., Ltd., Manchester, 1901–04; entered railway service, May 15, 1905, as an apprentice at Angus Shops of Canadian Pacific Railway Company, Montreal; transferred to Winnipeg as draftsman, December, 1906; foreman of wheel and tender shop, Winnipeg, August, 1908; shop engineer, Winnipeg, 1909; chief draftsman, mechanical department, Winnipeg, 1911; promoted to works manager, Weston Shops, Winnipeg, January, 1920; assistant superintendent of motive power, Lines West, July 1, 1928; and in his present position September 1, 1928.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

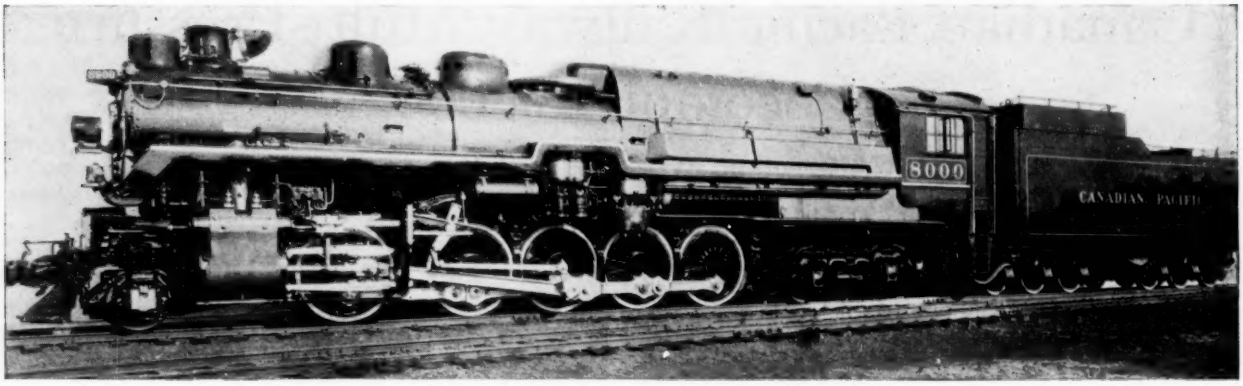


FIG. 1 MULTI-PRESSURE LOCOMOTIVE BUILT AT ANGUS SHOPS, LEFT-HAND VIEW

(Cylinders: High-pressure, one, 15½ in. by 28 in.; low-pressure, two, 24 in. by 30 in. Diameter of drivers, 63 in. Tractive effort, 90,000 lb. Boiler pressures: Closed circuit, 1350 lb; high-pressure, 850 lb; low-pressure, 250 lb. Weight on drivers, 320,550 lb. Weight of engine, 495,300 lb. Weight of engine and tender, 795,300 lb. Total length of engine and tender, 99 ft 3⅜ in. Water capacity, 12,000 imperial gal. Oil capacity, 4100 imperial gal.)

the closed system and high-pressure unit being completed, they were subjected to a hydrostatic test, each being found tight under the required test pressure. On April 13 the closed circuit and high-pressure unit were released from the jig and transferred to the machinery and low-pressure boiler unit. With the addition of cab, fixtures, fittings, and piping, the engine was ready for steam test. This test, involving the closed circuit and high-pressure and low-pressure boilers, was carried out in two stages—that is, it was thought desirable to work gradually to the maximum. Accordingly, the first test was discontinued when a pressure of 500 lb per sq in. was reached in the high-pressure boiler. Full pressure of 850 lb per sq in. was reached on final test.

Quite apart from the improved performance, the designing engineer probably receives more satisfaction from the surmounting of the many constructional difficulties always encountered in a new development. On account of its large size, many novel and ingenious features of construction had to be used for locomotive 8000, a few of which are of general interest.

The firebox ring serves as a header for the lower end of the high-pressure generating tubes. A complete casting with the likelihood of non-uniform walls and porosity was not favored, and finally this problem was solved by using four separate forgings drilled as in gun practice, with ends formed so that the circular extensions of one part slipped into lateral circular openings in the ends of the other two pieces. Corresponding wall holes gave the proper openings for circulation. The internal members were expanded into the outer members and then solidly welded in position, the openings in the ends of the headers being closed by solid plugs threaded and welded. This construction has proved most satisfactory and is a successful solution of a problem that initially gave considerable concern.

The closed steam-generating system with its evaporating tubes required most careful study to find a way of rolling in the tubes in the header in an absolutely uniform manner. These tubes are expanded into the header against small grooves in the header-tube opening. These grooves must be well filled and the rolling tight enough to withstand any possibility of leakage. A large number of tubes were expanded into test blocks, tested, and sawed apart. It was finally found that the tube was truly and thoroughly rolled in when it was expanded to the extent that the tube elongated 0.04 in. in a set distance above the tube opening. A small spring-plunger micrometer gage was then fixed by a quick-acting clamp to each tube, and the plunger was allowed to bear against the header. This method gave automatically the set distance, and the rolling was continued until the spring plunger extended sufficiently to indicate on the dial of the gage 0.04-in. growth in tube length. This method has proved thoroughly practical and has given most reliable results, ensuring uniformity of tube rolling. It might be interesting to mention here that when initial hydrostatic test was given the closed circuit, the pressure was raised at short intervals to a maximum of 2125 lb per sq in. At this pressure there was not the slightest indication of a leak.

The water gages in the closed generating system and high-pressure boiler were not successful with the rectangular glasses, which difficulty entirely disappeared with the use of sheet mica instead of glass, the mica resisting the heat, being flexible enough to withstand distortion, and being sufficiently transparent to give in the narrow opening a good indication of the water level.

Various features show excellent examples of how metallurgical developments have permitted constructions that would not have been possible with the materials of construction available only a few years ago. Stainless steels for valves and valve seats have surmounted the limitations of bronze and the corrosion defects of ordinary steels. Stainless-steel plates have provided a direct baffle for the oil flame to prevent the flame coming in direct contact with the drums. These plates, while resistant to high temperatures, have given some trouble on account of expansion which necessitated some changes in clearance. Low-carbon nickel steel for seamless-drum construction has permitted high factors of safety with reduced weight. Nickel-steel boiler plate also in the low-pressure boiler has given the requisite strength, with approximately 30 per cent reduction in weight. Nickel-steel forgings have provided toughness and resistance to abuse and impact, with minimum weight in various driving and motion parts.

It is customary on the Canadian Pacific Railway to operate locomotives by the pool system; that is, the engine is turned out for a train and the crew is assigned to it in the order in which they booked in from previous trips. This means that there are no steady engineers on any locomotive. Naturally it is desirable to make the operation of all locomotives as simple and as uniform as possible. This was kept in mind in all controls for locomotive 8000, and in spite of there being two superheaters and two throttles for the high- and low-pressure cylinders, the throttle mechanism was worked out so that only one throttle lever is used for opening both throttles. The operation is identical with that of an ordinary locomotive. The low-pressure boiler is fed with a standard "Elesco" CF-1 feed pump and is provided with a Hancock inspirator as an auxiliary. Incidentally, the CF-1 pump for the low-pressure boiler is located within a compartment provided in the tender tank—a location that had pre-

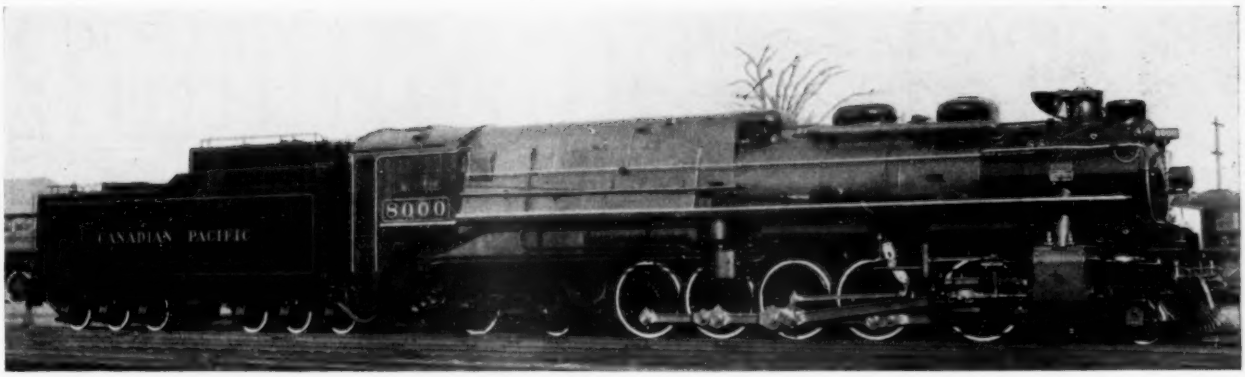


FIG. 2 MULTI-PRESSURE LOCOMOTIVE BUILT AT ANGUS SHOPS, RIGHT-HAND VIEW

viously been tried out with considerable success and advantage. The high-pressure boiler is fed with a specially designed CF-1 boiler-feed pump adapted to high-pressure service. A duplicate of this pump is also provided as a standby and auxiliary in connection with feeding the high-pressure boiler. The high-pressure water pumps are located one on either side of the boiler. The cut-off control in the valve motion is a straight duplicate of that of an ordinary three-cylinder locomotive. The oil-burner control is also identical. The only deviation is the two additional water gages on the high-pressure boiler for indicating the varying water levels and the two additional feed pumps for feeding this boiler. The crossover valve is an addition, and there are other features that require periodic checks by the engineer, but these have all been so simplified that it has not constituted any objection from the standpoint of simplicity of operation. In fact, after the initial period the locomotive was turned into the regular pool and is now handled by any engineer who may be assigned to it on any individual run. There has been keen interest on the part of the engineers and a desire to be assigned to locomotive 8000.

The most difficult operating problem on the Canadian Pacific Railway is the movement of both freight and passenger traffic over the Mountain Subdivision in British Columbia. Numerous heavy grades are encountered, the worst of which is 22.5 continuous miles of almost uniform 2.2 per cent grade, combined with numerous curves up to 12 deg.

In 1929, 20 new locomotives were put in service on this subdivision, of the 2-10-4 wheel arrangement, using 275 lb boiler pressure and two simple cylinders. These locomotives immediately introduced new standards of economy and performance, and by no means the easiest problem was selected when the Canadian Pacific Railway Company decided to construct for direct comparison a multi-pressure locomotive of substantially the same weight, wheel arrangement, and proportions as the T-1 class locomotives now handling traffic on this subdivision.

Locomotive 8000 burns oil as fuel, and the proportions of the flash pan and the size, number, and location of oil burners constituted a real problem, as there was practically no precedent on which to base the design. When the engine was originally turned out, several experiments were made with the location of a single burner and a double burner, and after a considerable amount of experimenting, proper locations for two burners were decided upon, one of which amply provides for ordinary demands, with the second one being cut in when required by abnormal conditions. It will be appreciated just how difficult this problem was when it is understood that three separate boilers—the closed system, the high-pressure boiler, and the low-pressure boiler—are all supported by the one firebox and combination of burners, and that a balance in the heat distribution must be maintained between these three heat-absorbing units throughout a wide range in steam demands of the locomotive. This has been partially offset by introducing a bypass valve from the high-pressure to the low-pressure boiler so that excess steam can be bypassed without losing it through the safety valves. Even as yet this bypass arrangement is used to some extent, but of course the ideal solution to be accomplished is the final adjustment and balance of component features of operation so as to obviate entirely the necessity of any bypassing. This can of course only be accomplished by close observation and adjustment of the locomotive in operation until the final proportions and adjustments are definitely determined. Originally the crossover valve gave trouble in that it caused severe disturbances of the water due to bypassing the high-pressure steam, which was overcome by a change in design of the valve outlet into the low-pressure boiler.

Proper draft adjustments and boiler conditions are somewhat more difficult to secure with locomotive 8000, as the exhausts are secured only at uneven intervals. This is due to the low-pressure cylinders having cranks at 120 deg instead of 90 deg, as in customary practice. There being no direct exhaust from the high-pressure cylinder, the uneven effects of the exhaust on the draft conditions of the engine are more noticeable at low speeds. This has been greatly improved with better combustion and increased superheat temperatures by installing baffle plates in the smokebox with apertures to more evenly diffuse the draft, a change in stack diameter, and changes in the exhaust nozzle. While great improvements have been effected, the problem is so different from an ordinary locomotive that further ones can be expected.

In operation the locomotive has thoroughly lived up to expectations in the matter of uniform torque, marked advantages in hauling heavy loads at low speeds due to more even torque, smooth running, development of maximum tractive effort, and slipping far less than the T-1-A engines under adverse rail conditions.

At the present time there is no doubt as to the increased maintenance cost of the multi-pressure engine over that of the conventional simple engines operating over the same division. This is generally true of engines radically different in design and construction, as the shop men lack experience on this type of engine and must become acquainted with the various details which are different from those on the engines they have been maintaining. Many experimental changes are also continually in progress at present which are apt to be mistaken for regular maintenance. Boiler men that are not acquainted with the new boilers require more time to wash out the two systems than they will when thoroughly acquainted with this work.

The safety valves presented a difficult problem in that their dimensions were strictly limited and that they must operate

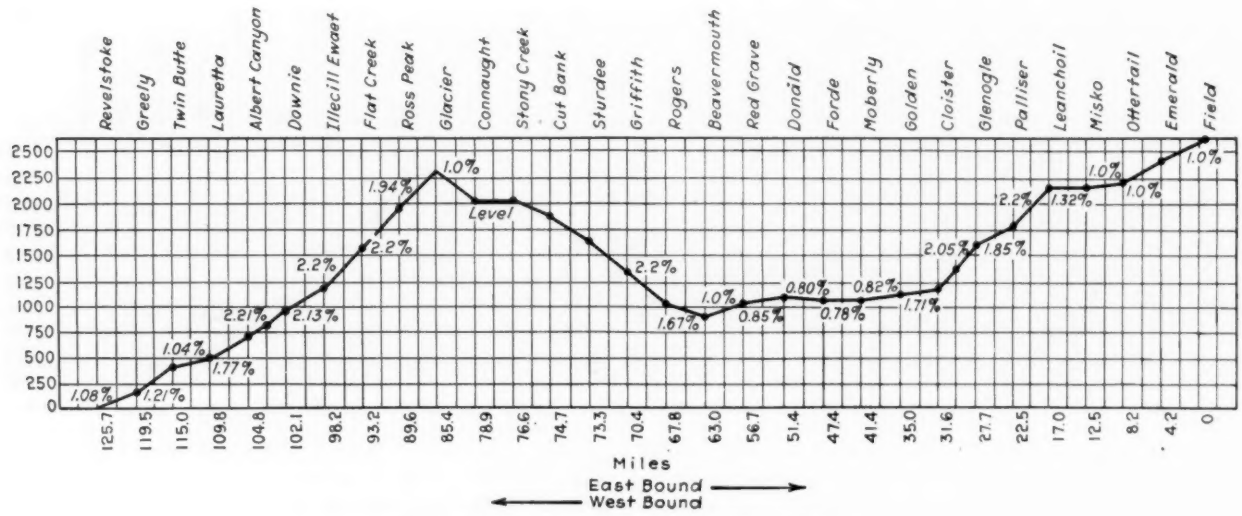


FIG. 3 PROFILE OF ROAD, REVELSTOKE TO FIELD

TABLE 1 PERFORMANCE OF ENGINE 8000 AS COMPARED WITH ENGINES OF THE 5900 CLASS

(Record of T-4 and T-1 class, Beavermouth to Glacier, westbound, 22.5 miles)

	T4-a	T4-a	T4-a	T4-a average	Improvement T4-a over T1	T1-a average	T1-a	T1-a
Engine no.	8000	8000	8000	.....	....	.....	5907	5905
Date	4-23-32	4-15-32	6-22-32	.....	....	.....	4-26-32	4-28-32
Number of cars, total	37	44	16	.....	....	.....	20	17
Weight of train, equivalent gross tons	1,029	1,060	1,103	1,064	....	1,069.5	1,046	1,093
Equivalent gross ton miles	23,152	23,850	24,818	23,940	....	24,063	23,535	24,592
Lb of oil, total	3,374.8	4,399.8	4,689.9	4,154.7	....	4,878.4	5,144.4	4,612.5
Lb of oil used per 1000 e.g.t. miles	146.0	184.8	188.9	173.2	14.9%	203.5	218.8	188.2
Lb of oil used per locomotive mile	149.9	195.5	208.4	184.6	....	216.8	228.6	205.0
Lb of water, total	47,850	47,130	49,150	48,040	....	49,885	47,840	51,930
Lb of water used per lb of oil	14.2	10.7	10.47	10.79	5.3%	10.25	9.3	11.2
Total time on road, hr and min	1:16	1:45	1:47	1:36	....	1:24	1:18	1:31
Number of stops	.....	.....	.....	.....	....	.....	.....	.....
Total time delayed, hr and min	.....	.....	.....	.....	....	.....	.....	.....
Ruling grade (per cent) and curve (deg)	2.2-10	2.2-10	2.2-10	.....	....	.....	2.2-10	2.2-10

(Record of T-4 and T-1 class, Albert Canyon to Glacier, eastbound, 19.4 miles)

	T4-a	T4-a	T4-a	T4-a average	Improvement T4-a over T1	T1-a average	T1-a	T1-a	T1-a
Engine no.	8000	8000	8000	.....	....	.....	5907	5905	5905
Date	4-30-32	6-15-32	6-23-32	.....	....	.....	4-25-32	4-27-32	4-29-32
Number of cars, total	49	37	41	.....	....	.....	44	45	44
Weight of train, equivalent gross tons	1,088	1,076	1,108	1,090.6	....	1,053	1,058	1,051	1,052
Equivalent gross ton miles	21,107	20,874	21,495	21,158.6	....	20,441	20,525	20,389	20,408
Lb of oil, total	4,583.5	4,960.7	4,951.0	4,831.7	....	5,467.0	5,424.8	5,753.6	5,192.7
Lb of oil used per 1000 e.g.t. miles	217.1	237.6	230.3	228.3	14.7%	267.0	264.6	283.4	254.0
Lb of oil used per locomotive mile	236.2	255.7	295.2	249.0	....	281.2	279.5	296.5	267.6
Lb of water, total	54,460	49,150	54,070	52,560	....	54,040	50,570	53,270	58,270
Lb of water used per lb of oil	11.88	9.9	10.92	10.9	10.1%	9.9	9.3	9.2	11.2
Total time on road, hr and min	1:52	1:19	1:48	1:40	....	1:33	1:23	1:51	1:26
Number of stops	2	.....	1	1	....	2	2	1	3
Total time delayed, hr and min	0:29	.....	0:20	0:16	....	0:2	0:2	0:1	0:2
Ruling grade (per cent) and curve (deg)	2.2-10	2.2-10	2.2-10	.....	....	.....	2.2-10	2.2-10	2.2-10

under conditions of vibration and exposure not met with in stationary practice. The valves originally applied were not altogether successful. The valves on the closed-circuit system gave trouble largely on account of the actual details of construction, which has been overcome by rebuilding the valves in the railway shops by the use of a new form of seat and valve. The high-pressure safety valves also required a change in form.

The check valves in the water-delivery line to the high-pressure boiler gave trouble on account of the very high concentrated load on the valve, it being almost impossible to maintain it in a tight condition for any length of time due to the pounding it received. The solution was a twin check valve, with the proper capacity provided by two valves side by side in the same body, so that the total load on each valve of reduced diameter was little greater than the load on the single valve used in conventional practice.

Although over one year has elapsed since the engine was first put into service, the breaking-in period being on the Eastern Lines, its transfer in knocked-down condition and re-assembly on the Western Lines, and the various adjustments required while the engine was in regular operation on its assigned location on the Mountain Subdivision, it has not been possible to give a complete summary of the operating performance of the locomotive over the entire period since the locomotive was built. The engine performance has been closely followed by observers, and results show that a fuel saving of 14.8 per cent has been effected on this division under regular service conditions at slow speeds on 2.2 per cent grades between Albert Canyon and Glacier eastbound and Beavermouth and Glacier westbound. Tests conducted on the Eastern Lines over the level Winchester Subdivision, between Montreal and Smith Falls, showed fuel economies of 2.5 per cent under higher speed and heavy tonnage.

(*Erratum: In the next to the last line on this page, 2.5 should read 25*)

DEPOSITORIES FOR A.S.M.E. TRANSACTIONS IN THE UNITED STATES

Alabama

Auburn.....Engineering Library, Alabama Polytechnic Institute
Birmingham.....Public Library

Arkansas

Fayetteville.....Engineering Library, Arkansas University

California

Berkeley.....Library, University of California
Long Beach.....Public Library
Los Angeles.....Public Library
University of Southern California
Oakland.....Oakland City Library
Teachers' Professional Library
Pasadena.....Library, California Institute of Technology
Santa Clara.....Library, University of Santa Clara
San Diego.....Public Library
San Francisco....Public Library (Civic Center)
Engineers Club of San Francisco
Mechanics Institute
Stanford University.....Library, Stanford University

Colorado

Boulder.....Library, University of Colorado
Denver.....Public Library
Fort Collins.....Colorado State Agricultural College

Connecticut

Bridgeport.....Public Library
Hartford.....Public Library
New Haven.....Public Library
Library, Yale University
Waterbury.....Silas Bronson Library

Delaware

Newark.....University of Delaware
Wilmington.....Wilmington Free Institute

District of Columbia

Washington.....Scientific Library, United States Patent Office
Library of Congress
Bureau of Standards Library
George Washington University
Catholic University

Florida

Gainesville.....University of Florida
Jacksonville.....Free Public Library
Miami.....Public Library
Tampa.....Public Library

Georgia

Atlanta.....Carnegie Public Library
Georgia School of Technology
Savannah.....Public Library

Idaho

Moscow.....University of Idaho

Illinois

Chicago.....John Crerar Library
Western Society of Engineers
Library, Armour Institute of Technology
Public Library of Chicago
Moline.....Public Library
Urbana.....University of Illinois

Indiana

Evansville.....Public Library
Fort Wayne.....Public Library
Indianapolis.....Public Library
Indiana State Library
Notre Dame.....Library, University of Notre Dame
Terre Haute.....Rose Polytechnic Institute
West Lafayette...Library, Purdue University

Iowa

Ames.....Iowa State College
Des Moines.....Public Library
Iowa City.....State University of Iowa

Kansas

Kansas City.....Public Library, Huron Park
Lawrence.....Library, University of Kansas
Manhattan.....Kansas State Agricultural College
Wichita.....Wichita City Library

Kentucky

Lexington.....University of Kentucky
Louisville.....Speed Scientific School
University of Louisville

Louisiana

Baton Rouge.....Louisiana State University
New Orleans.....Howard Memorial Library
Louisiana Engineering Society
Public Library

Maine

Orono.....University of Maine

Maryland

Annapolis.....United States Naval Academy
Baltimore.....Johns Hopkins University
Engineers Club of Baltimore
Public Library

Massachusetts

Boston.....Northeastern University
Boston Public Library
Cambridge.....Harvard University (Engineering Library)
Massachusetts Institute of Technology
Fall River.....Public Library
Lowell.....Lowell Textile Institute
Lynn.....Free Public Library
New Bedford.....Free Public Library
Springfield.....Springfield City Library
Tufts College.....Tufts College
Worcester.....Worcester Polytechnic Institute
Free Public Library

Michigan

Ann Arbor.....University of Michigan
Detroit.....Public Library
Cass Technical High School
Highland Park Public Library
University of Detroit
East Lansing.....Michigan State College
Flint.....Public Library
Grand Rapids....Public Library
Houghton.....Michigan College of Mining & Technology
Jackson.....Public Library

Minnesota

Duluth.....Public Library
Minneapolis.....University of Minnesota
Minneapolis Public Library (Engineering and Circulating Libraries)
St. Paul.....James Jerome Hill Reference Library

Mississippi

Columbus.....Mississippi Agri. & Mech. College

Missouri

Columbia.....University of Missouri
Kansas City.....Public Library
Rolla.....Missouri School of Mines and Metallurgy
St. Louis.....Engineers Club of St. Louis
Public Library
Washington University
Mercantile Library

Montana

Bozeman.....Montana State College

Nebraska

Lincoln.....University of Nebraska
Omaha.....Public Library

Nevada

Reno.....University of Nevada Library

New Hampshire

Durham.....University of New Hampshire

New Jersey

Bayonne.....Free Public Library
Camden.....Free Public Library
Elizabeth.....Free Public Library
Hoboken.....Stevens Institute of Technology
Jersey City.....Free Public Library
Newark.....Newark College of Engineering
Free Public Library
New Brunswick...Rutgers University
Paterson.....Free Public Library
Princeton.....Princeton University
Trenton.....Free Public Library

New York

Albany.....New York State Library
Brooklyn.....Polytechnic Institute
Pratt Institute
Brooklyn Public Library
Buffalo.....The Grosvenor Library
Engineering Society of Buffalo
Buffalo Public Library
Ithaca.....Cornell University
Jamaica, L. I.....Queens Borough Public Library
New York.....Engineering Societies Library
Public Library
College of the City of New York
Cooper Union
Columbia University
N. Y. University Library
Potsdam.....Clarkson College of Technology
Rochester.....Rochester Engineering Society
Schenectady.....Union College
Syracuse.....Syracuse University
Public Library
Troy.....Rensselaer Polytechnic Institute
Utica.....Public Library
Yonkers.....Public Library

North Carolina

Chapel Hill.....University of North Carolina (Engineering Library)
Raleigh.....North Carolina State College

North Dakota

Fargo.....North Dakota State College
Grand Forks.....University of North Dakota

Ohio

Ada.....Ohio Northern University
Akron.....Public Library
University of Akron
Canton.....Public Library
Cincinnati.....University of Cincinnati
Public Library
Engineers Club of Cincinnati
Cleveland.....Public Library
Case School of Applied Science
Cleveland Engineering Society
Columbus.....State of Ohio Library
Public Library
Ohio State University
Dayton.....Engineers Club of Dayton
Toledo.....Public Library
University of Toledo
Youngstown.....Public Library

Oklahoma

Norman.....Oklahoma University
Oklahoma City...Public Library
Stillwater.....Oklahoma Agricultural and Mechanical College
Tulsa.....Public Library

Oregon

Corvallis.....Oregon State Agricultural College
Portland.....Portland Library Association

Pennsylvania

Allentown.....Free Library
Bethlehem.....Lehigh University
Easton.....Public Library
Lafayette College
Erie.....Public Library
Lewisburg.....Bucknell University
Philadelphia.....Engineers Club
Drexel Institute
University of Pennsylvania
Franklin Institute
Pittsburgh.....University of Pittsburgh
Engineers' Society of Western Pennsylvania
Carnegie Institute of Technology
Carnegie Library (Schenley Park)
Carnegie Free Library of Allegheny
Reading.....Public Library
Scranton.....Public Library
State College.....Pennsylvania State College
Swarthmore.....Swarthmore College
Villanova.....Villanova College
Wilkes-Barre.....Public Library

Rhode Island

Kingston.....Rhode Island State College
Providence.....Brown University
Providence Engineering Society
Public Library

South Carolina

Clemson College..Library, Clemson College

Tennessee

Kingsport.....Public Library
Knoxville.....University of Tennessee
Memphis.....Goodwin Institute
Nashville.....Vanderbilt University

Texas

Austin.....University of Texas
College Station...Agricultural & Mechanical College of Texas
Dallas.....Public Library
El Paso.....Public Library
Fort Worth.....Carnegie Public Library
Houston.....Rice Institute
Public Library
Lubbock.....Texas Technological College (School of Engineering)
San Antonio.....Carnegie Library

Utah

Salt Lake City....University of Utah
Public Library

Vermont

Burlington.....University of Vermont

Virginia

Blacksburg.....Virginia Polytechnic Institute
Charlottesville....University of Virginia
Norfolk.....Carnegie Public Library
Richmond.....Virginia State Library

Washington

Pullman.....State College of Washington
Seattle.....Public Library
Engineers Club
University of Washington
Spokane.....Public Library
Tacoma.....Public Library

West Virginia

Morgantown.....West Virginia University

Wisconsin

Madison.....Library, University of Wisconsin
Milwaukee.....Public Library
Board of Industrial Education, Vocational School Library
Marquette University

Wyoming

Laramie.....Wyoming University

Progress in Textiles

NOTWITHSTANDING that the textile industry in practically all branches has been operating under a materially reduced schedule, all of its plant executives have been searching for means to eliminate waste and to develop improved machinery and methods, so that even in 1932 some of the mills and textile plants have been operating successfully. A depression may therefore really develop into an important industrial revival.

During the last few years textile-machinery builders have been greatly improving and standardizing their products and developing new mechanisms and processes that have materially benefited the industry. In fact, this has been carried on to an extent not before witnessed in 25 years.

Much progress has been made in scrapping obsolete mills and mechanisms, as well as the processes themselves, and much more remains to be accomplished.

Such buildings as have been erected have been mostly small additions to existing textile mills, and little of new design or construction has been noticed, but the new machinery and processes that have been developed require very little actual floor space as compared with the old methods, and this argues well for the development in the years that are to come.

An important trend that apparently will terminate in a most satisfactory condition is the seeming willingness now of the actual leaders of the textile industry to meet and discuss their several problems, whereas in the past this has been difficult to accomplish. And it is earnestly hoped that during the coming year the engineering executives connected in any way with the textile industry may make it possible to bring into a still more closely knit group the textile executives of every branch of the industry so that the basic problems of machinery, development, policies, and processes may be more thoroughly discussed and worked out. For to the farseeing engineer there is every evidence that most of these problems now so jealously guarded by the several branches of the industry are in reality only problems common to the industry as a whole when viewed by the impartial technically trained engineer.

Dyeing and Finishing

An outstanding engineering achievement in the textile field is the development of the two-motor all-electric dye jig. This mechanism was developed by an English engineer, and the American rights were acquired by a textile-machinery manufacturing firm that has greatly improved the original machine. This all-electric jig gives constant cloth speed, constant tension controllable at will from no tension to any tension required, and also automatic reversing and stopping after a predetermined number of ends, making it a valuable addition to the textile-producing mechanisms.

A simple mechanical fabric guide is operated by a selvage finger moving with the fabric, its claimed advantages being immunity to moisture or lint and adaptability to either wet or dry fabrics and the successful handling of delicate fabrics.

A light-running dye jig adapted for light-silk, rayon, cotton-and-silk, rayon-and-silk, and fine cotton fabrics is an individually driven, self-contained unit and, because of the absence of pulleys and belting, it is claimed that the operator can move freely on all sides of the jig. The mechanism has been designed to secure silent operation and low upkeep costs. It is possible that this mechanism is quite similar to another developed during the last three years. Both appear to be a long step in the right direction in the development of these operations.

In an improved piece-dyeing machine used principally for silk fabrics, the goods instead of being pulled through the liquor in rope form are said to practically float through it in semi-open form. The mechanism is designed to reduce the pull to a minimum, thereby eliminating marks and insuring good penetration.

A machine for shrinking cloth without the application of tension accomplishes this through automatic festooning of the goods. It is asserted that with these feeding and looping devices the materials can be fed in the flat, open condition, without the danger of wrinkles or marks. Another claim is natural drying without heat or air circulation, although heat or air may be used if desired.

Another conditioning machine for restoring the moisture content has been offered to the finishing industry and is claimed to be equally effective on worsteds, woolens, cottons, silks, and rayons. Close control and the handling of a large volume of goods in minimum space and time are claimed and also that the mechanism can be operated in range in connection with other machinery.

In the Palmer Sanforizing unit brought out last year, recent developments that are in fact decided improvements over the older methods accomplish the shrinking of heavy fabrics such as denims, moleskin, and auto-top materials through an ingeniously designed and arranged feeding device, sprays, and steamers, together with individual motor drives and feeding machines and the clutches.

A metal-clad piece-dye tub introduced about the first of the year will apparently replace the old-time wood tub on account of accessibility in cleaning when changing colors.

Dyeing of a yarn package either from the inside out or the outside in is a feature of a new ultra-package-dyeing machine. The advantages claimed are in the ready flow of the dye liquor, the distribution of the dye liquors over the material, the special carriers, the tank design, and the dyestuff agitator, and also the low steam consumption and minimum space requirements.

A textile drier developed abroad after several years of effort is based upon a novel method of air circulation and the manner of conveying the material. Negotiations for the manufacture of the drier in this country are now under way.

A highly mechanically perfected mechanism for the continuous dyeing and slashing of cotton and rayon warps has been patented, and it is believed that the model recently turned out is the first really practical one that has been produced. It has many novel features and is of beautiful construction. It will be necessary, however, to withhold comment until the machine has been in use for a few months so that it may be observed under actual operation.

Cotton

A new jig has been designed to give a better penetration of the dyestuffs in the cotton-finishing industry. This jig will apparently replace the older type of jigs and padders on the cotton and possibly on the silk fabric. The more satisfactory penetration should result in better color. In this apparatus

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Presented at the session of the Textile Division, Thursday morning, December 8, 1932, during the Annual Meeting, New York, N. Y., December 5 to 9, 1932, of The American Society of Mechanical Engineers.

Compiled by the Progress Report Committee, W. L. Conrad, Chairman.

Personnel of the Executive Committee of the Textile Division for 1932: Henry M. Burke, Chairman; Paul A. Merriam, Vice-Chairman; M. A. Golrick, Jr., Secretary; H. V. W. Scott, William L. Conrad, Clifford H. Ramsey, Associates: Reynolds Longfield, Harry D. Learnard, A. W. Benoit, Albert Palmer.

the squeeze rolls are inside the dye tank, and may therefore be partially or completely submerged in the dye liquor, overcoming the objection that in some operations the squeezing on the selvage and body of the fabric has not been uniform, resulting in uneven color.

A lubricator for bobbin winders has been developed in Europe. This mechanism, combined with a special stretch device, insures thorough and uniform lubrication, with friction eliminated.

What has been described as an entirely new machine for the twisting of wool, worsteds, rayon, cotton, and other yarns has been announced. This mechanism is claimed to give increased uniformity of twist, together with a materially increased production; in fact, it is claimed that more product is possible than is obtainable in existing frames. It is also claimed that this mechanism combines both the principles present in today's methods of twisting, and a further claim is made that the manipulation of this machine is very simple and the frame is said to be suitable for twisting crepes and all yarns where medium or hard twist is required. It is particularly advantageous for the twisting of cotton crepe yarn, wool crepe, single yarn twist, and knit silk crepe yarn.

A novelty twister for producing a wide range of fancy yarns has been introduced during the past year. The heads of this mechanism are so arranged that two or three lines of independently driven under-rolls may be used with each line of rolls having more than one speed a head, and these lines of rolls may be timed to run at the same speed or at different speeds and are centrally controlled. This mechanism has many other advantages that make it of special interest to the cotton industry.

A novelty tester has been produced for portraying graphically the inherent evenness characteristics of various slivers and coarse rovings, and the manufacturer has outlined a method of evaluating the graph prepared by this instrument.

The general tendency of the manufacture of looms for cotton and worsted goods has been to standardize in the manufacture of this machine, and these parts are now interchangeable, as they are on other modern equipment. These new weaving machines have been developed with the idea of showing a return on the investment when they are purchased for the replacement of old machinery.

The single-process picker has become almost the standard for picker rooms. Two- and three-process pickers are today considered obsolete. In the card room the long-draft system has been applied to roving frames, and it is understood is being used in a few mills at the present time. The long-draft spinning system is now being used extensively as replacement becomes necessary or additions are made, and it is claimed that it may be now considered as the standard type of spinning.

Spinning frames are being redesigned and are now being made to produce large packages suitable for use direct in the warper creels, thus doing away with spooling. These spinning frames are equipped with knee brakes, moving thread boards, and improved features in many parts.

Old-style spooling and warping have been largely eliminated and has been replaced with either automatic spoolers and high-speed warpers or with high-speed winders and warpers. The elimination of the spooler or winder from this process is forecast.

For mills that comb their cotton, a new and improved comber is available that gives a much greater production than heretofore.

Process Development

One of the important developments during the year in the process field is the commercial production of cuprammonium rayon by a particular process. This radical change in rayon production methods has been described in detail in some of the leading textile journals, and its adoption would indicate a revolutionary feature in the rayon industry.

There is also reported the possibility of the production of acetate yarn by another organization, but up to the present time few actual details of this process have been made available, although it is understood that it is an improvement and simplification of older methods.

Silk

There has been developed this year the 2 × 1 shuttle-changing automatic silk loom, and this improvement has also been applied to the so-called super-silk loom. This is an electro-mechanical mechanism and permits both shuttles of this loom to be replaced with new shuttles without stopping the loom, the necessity for changing shuttles being indicated by an electrical detector. This type of detector, however, cannot be used except under certain conditions, but an internal detector, which is a part of the shuttle, has been developed that will fill the needs of all general conditions. This internal detector shows great promise, especially with very fine rayon yarn. The 2 × ' shuttle-changing silk loom is particularly adapted to fine rayon goods where bobbin-changing looms have been found to be impracticable.

An improved coning machine equipped for silk throwsters has been placed on the market and is said to produce a desirable package economically and at an exceptionally high spindle speeds, and the arrangement of the spindle units is declared to promote flexibility and in the allotment of spindle per operator to meet changing mill requirements.

Wool

An interesting aging machine of conventional design has been introduced into the wool industry and would seem to be well adapted for the purpose. This mechanism is provided with heat-retaining insulation as well as vapor-tight bushings and other devices for retaining the necessary heating requirements.

A single-apron stock drier of improved design has now been developed and, inasmuch as this machine is built in sections and can be furnished in any length and in a number of widths, it would appear that this mechanism would offer many advantages. The conveyor for carrying the stock to this machine is declared to be of new and improved construction, and the machine is adapted for fine stock, with the conveyor drive equipped for variable-speed control. Other improvements and attachments combine to make this mechanism very interesting.

The woolen industry has apparently accepted and installed tape condensers in greater quantity than ever before, thus indicating that this machine is an improvement in the industry.

Spinning frames are now in use in the place of mules, and it is understood that more are being installed as tradition gives way to experience.

It is understood that a new line of French drawing machinery is in process of manufacture, and the announcement of the introduction of this mechanism may be looked for in the near future.

Printing

A multi-color warp-printing machine is announced. In this machine yarns in warp form can be printed in from one to four colors by means of the new multi-color mechanism. Direct basic or acid colors may be handled in this machine. For vat dyeing a steam or aging chest, along with a two-compartment wash box, is provided. These machines are designed to be mounted directly behind the slasher so as to print the warp as it leaves the section beam in its full width.

Knit Goods

Several new features and a number of improvements are embodied in recent models of a full-fashioned hosiery machine that has been offered to the trade. Many of the new devices of this machine are really improvements over a similar machine offered a short time ago. For single-operation work the single-narrowing machine is now replaced by a full-narrowing machine, and it is now possible to make the diamond toe on single-operation hosiery.

Another full-fashioned hosiery machine has been developed and is what might be styled a jacquard machine that will produce an openwork pattern over the entire width of the stocking, and it is reported to produce this at an unusually high speed. The machine is designed to make single-unit stockings that do not have to be transferred, and it can also be used either as a legger or footer to make stockings in the usual manner with French foot or English instep. And the speed of this machine, depending on the gage and the yarn employed, is 70 to 75 courses per minute.

Still another hosiery machine recently developed is a new wrap machine of individual-needle type and makes possible the using of a very large number of different colors. This machine will turn out hose with high splice, double sole, and plated body. Any type of warp yarn may be used.

It is understood that some years ago a patent was taken out covering the manufacture of non-run hosiery, but up to quite recently little has been heard of this improvement. It is now expected that this device will be rapidly perfected and that very shortly the product will be placed on the market. This will fill a very important position in the knitting industry, and it is hoped that it will eliminate the many objections that have been raised to the products of the present-day knitting machines, notably the run feature of knit goods.

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Sanforizing Process

By SANFORD L. CLUETT,[1] TROY, N. Y.

The paper describes a machine that will treat a fabric so that its dimensions in warp and filling will be what is desired after the fabric is subjected to laundry washing or other cleaning process. The causes of the shrinkage of a fabric when subjected to full laundry treatment being mostly mechanical, an effective treatment to prevent shrinking may be found in some process of mechanically rearranging the fibers of the fabric (including changing the count of warp and filling) to the same extent that the fibers would rearrange themselves if subjected to a full laundry washing. Data are given of characteristic runs of different fabrics in the Sanforizing process.



THE Sanforizing process and the mechanism for it were designed primarily to treat a fabric so that its dimensions will remain substantially unchanged when the fabric is subjected to a laundry washing or other cleaning process. It is common experience that finished textile fabrics change in length or width when laundered; this change is generally a shrinkage. The principal reasons that shrinkage occurs are as follows:

(*a*) Practically all textiles are woven under tension, generally in both warp and filling. For obvious reasons textile machinery is designed to operate this way.

(*b*) In the bleaching and finishing of textiles, from the moment that the webs are sewed together for putting through the rope or open-bleach processes until they are finally bleached and finished, they are stretched every time they are transported from one station to another. This pulling tends to stretch and straighten out the warps and thus narrow the goods. Narrowness is counteracted at one or more stations during the finishing process by pulling the goods out in width through the use of expanders or tenters, or both. As a rule when the material is pulled out or held out in width the warps are still held under tension; thus the pulling out in width also puts tension on the warps as well as on the filling. The result is that most finished woven fabrics are elongated during the finishing process.

(*c*) When textiles are manufactured into garments, the material may be subjected to more or less stretching in length or in width.

(*d*) As a result, such fabrics are only awaiting a favorable opportunity to change their dimensions. This opportunity occurs if the finished fabrics are dampened with or immersed in water. The water acts as a lubricant and allows the fibers to readjust themselves. The fibers also swell; and as the yarns are twisted this swelling causes a shortening of the yarns. The combination of swelling and shortening of the yarns, owing to the twist, further causes a shrinkage of the fabric because of a rearrangement of the position of the yarns. The most general cause of garment shrinkage is the laundry wash wheel or other mechanical manipulator of wet garments. During laundry washing a garment is tumbled about in hot soapy water, generally with a heavy charge of goods, and the yarns are not only further allowed to contract, but they are forced and pounded together by the action of the water and of the other garments in the wheel; there is a fulling effect somewhat similar to that which takes place when wool is washed. Also caustic and bleaching solutions may be present in the wash wheel and have a further shrinkage effect on the material. It has been observed that woven fabrics shrunk by water alone will, when subsequently subjected to a full laundry wash, shrink an additional amount varying from $^1/_2$ in. to the yard to as much as 2 in. to the yard. In fabrics in which the yarns are only partially or altogether unbleached, the fibers are generally water repellent. These fabrics as a rule not only have a high shrinkage factor on washing, but continue to shrink in subsequent laundry treatments until the waxes and gums are entirely eliminated.

Methods of Preshrinking

In order to minimize the laundry shrinkage of fabrics as far as possible, several methods of preshrinking have been in use for many years. Among these may be enumerated:

(*a*) Wetting or soaking the fabric and drying it with as little strain as possible on the warp and filling.

(*b*) Chemical shrinking.

(*c*) Washing the fabric.

These three hold important places in the shrinking art. However, the process to be described has been built on the principle that inasmuch as the causes of the shrinking of fabrics when they are subjected to a full laundry treatment are mostly mechanical, the most effective treatment to prevent shrinking may be found in some process of mechanically rearranging the fibers of the fabric (including changing the count of warp and filling) to the same extent that the fibers would arrange themselves if subjected to a full washing in a laundry.

Process for Anticipating Shrinkage

The Sanforizing process is, broadly stated, one for predetermining the superficial dimensions of a fabric and at the same time conserving, and usually improving, its finish. It may be so applied as to shrink or to elongate either the warp or the filling, or both, in any desired combination of dimension change. For example, any one of the following dimensional effects may be attained:

(1) Warp shrunk, filling shrunk
(2) Warp shrunk, filling unaffected
(3) Warp shrunk, filling elongated
(4) Warp unaffected, filling shrunk
(5) Warp unaffected, filling elongated

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

(6) Warp unaffected, filling unaffected
(7) Warp elongated, filling shrunk
(8) Warp elongated, filling unaffected
(9) Warp elongated, filling elongated

In the great majority of cases preshrinkage is an end in view; and discussion will therefore be confined to that aspect of the process. The procedure consists of the following:

(*a*) Precisely determining the changes that take place in the longitudinal and transverse dimensions of the fabric when it is subjected to a full laundry wash.

(*b*) Applying the process to bring about these dimensional changes.

(*c*) Giving to the fabric by the operation of the process an enhanced commercial finish.

The standard method of determining the changes that take place in the dimensions of a fabric when subjected to a full laundry wash is as follows:

A 20-in. test sample is cut from a web of the cloth to be shrunk, and 18-in. in both warp and filling is marked off accurately on it in indelible ink. The sample is subjected to normal laundry washing treatment: a 5-min. rinse in either cold or warm water, preferably in water at 140 deg. fahr.; the water is drawn off; hot water and sufficient soap are added to give a good running suds, which is run in the wheel for 30 min., and then drawn off; a 10-min. rinse is given in clear water at 140 deg. fahr., followed by two 5-min. rinses, each time drawing off the previous rinses before adding fresh water; a final 5-min. rinse with water containing about one-eighth of 1 per cent of 56 per cent acetic acid is given. This is done to neutralize thoroughly any small traces of alkali remaining in the goods. After washing, the sample is squeezed by hand without stretching and placed on a screen in a dry room. After drying, the sample is dampened slightly with a spray of water and pressed with a hot iron. During the pressing, care is taken not to rub the cloth, as is customary in hand ironing, but simply to press out and smooth the cloth without stretching.

The changes that take place in the marked 18-in. sample are noted and are recorded on a card for both warp and filling in terms of inches and decimals of an inch per yard of fabric. These changes are called the "PWS" or "potential wash shrinkage" of this particular fabric, elongation being marked with the plus sign and shrinkage with the minus sign. This record serves for setting the Sanforizing machine so that it will change the warp and filling dimensions of the material to be treated by the amount that has been indicated by the sample test.

A similar test piece is cut from the finally treated fabric to determine whether the process has been accurately carried out. It is, as a rule, not difficult to attain stability of a fabric to within $^1/_4$ in. per yard in both warp and filling.

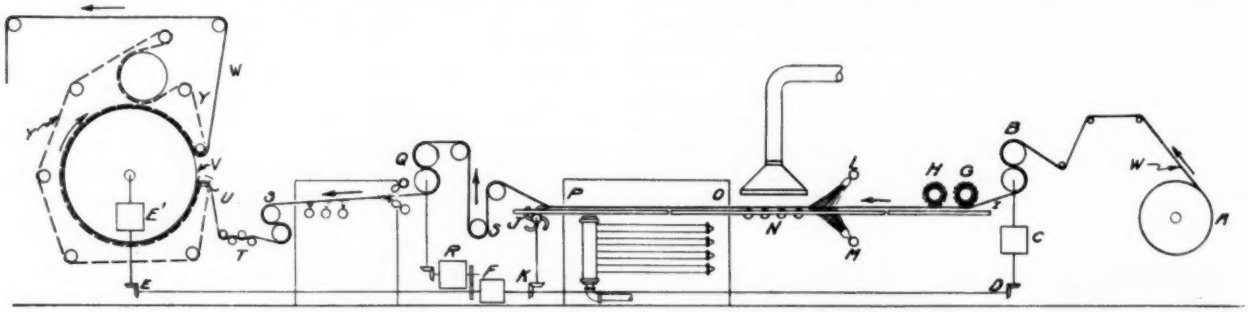


FIG. 1 DIAGRAM OF SANFORIZING MACHINE AND ITS OPERATION

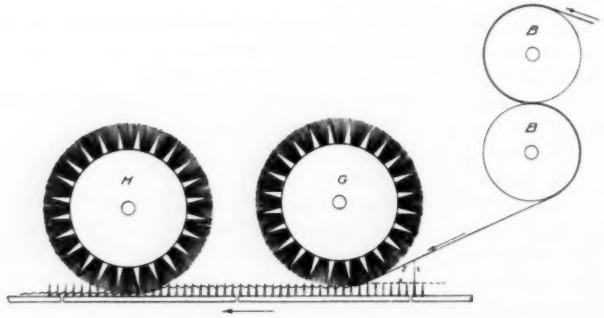


FIG. 2 BRUSHES FOR PRESSING THE FABRIC UPON HOLDING PINS

SANFORIZING MACHINES

The process, in its essentials, can be carried out with various species and types of apparatus. The machinery comprises a combination of types which has, after extensive tests, demonstrated efficient adaptability and adjustability economically to satisfy the requirements and solve the problems presented by a wide range of fabric conditions. The machine and its operation (see Fig. 1) may be briefly described as follows:

The fabric to be treated (marked *W*) is brought to the machine at *A* on a roll or loose on a truck. The material is passed over one or more bars to open it and eliminate wrinkles and is seized at *B* by two grasping (or feeding) rolls that are driven through a variable-speed gearing *C* from a main-line shaft *D-E*, which shaft is driven by a motor *F*. The motor, driving through one shaft, ensures synchronization throughout the machine. The material passes from the feeding rolls *B* to a tenter *I-J*, and is fed onto pins or into other gripping devices at *I*, which are mounted on the chain of the tenter, the tenter chain being driven from the main shaft at K. If it is desired to shrink the goods in the warp, the variable-speed gear *C* is adjusted to feed the goods to the tenter at a faster rate than that of the tenter chain, with the result that the material, seized by the tenter pins or clamps, is slack and undulated in the direction of the warp. The algebraic differential between the velocity of feed at the feed rolls and the velocity of the tenter chain is the measure of warpwise dimensional alteration to be attained in the tentering machine.

Fig. 2 shows the manner in which material is applied for positive warpwise preshrinkage. *G* is a vertically floating revolvable brush. In starting, the goods are passed under this brush and on the pins close to the selvage edge of the material; the brush of its own weight drops until the goods are stretched taut between the brush *G* and the feed rolls *B*. As the machine is started and the pins move forward in the direction of the arrow, the brush *G* drops to a point where in the small triangle *xyz*, the side *y* is to the side *x* as the velocity of the surface of the feed roll *B* is to the velocity of the tenter chain. As the chain with the material on the pins passes from under the feed brush *G*, the excess

material takes the form of ripples somewhere between the points and the bases of the pins. A second vertically fixed brush H, rotated by the passing pins, presses the goods down on the pins.

Referring to Fig. 1, the material passes on the tenter past the brush H and is moistened from above or from beneath (sometimes both) by water sprays L and M, which may be operated to deliver any desirable quantity of moisture, and at about this point the tenter chains diverge so that a slight stretch is given to the filling threads. As the material progresses through the tenter the tension thus imposed on the filling threads tends to straighten

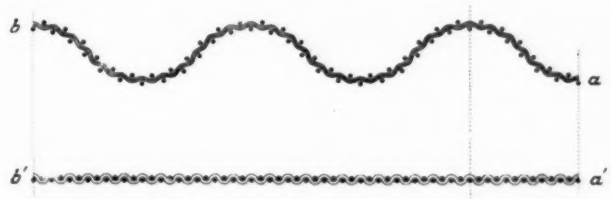


FIG. 3 METHOD OF SHORTENING THE FABRIC

them, and to cause a largely increased accentuation in the weave undulations of the warp threads. This action shortens the fabric, as illustrated in Fig. 3, in which ab shows a cross-section along one of the warp threads near the middle of the fabric before the filling threads are stretched, for a distance of, say, 2 in. in a straight line. As the filling threads are pulled they tend to become straight, causing the warp threads to assume an accentuated crinkle, as shown between a' and b'. All of the wave in the fabric between a and b has now been converted into increased undulations or crinkles in the warp threads. There are the same number of filling threads between a' and b' as there were originally between a and b, but the fabric itself is shorter by an amount equal to the excess feed.

The assumption of this new relation by the warp and filling threads is greatly facilitated by the moisture from the spray at L and M, and may be still further assisted by steamers at N, through which just sufficient steam may be supplied to properly diffuse the moisture through the fabric.

The tenter chains carrying the cloth holders (pins in this case) diverge just sufficiently to cause the fabric to become smooth, and completely take up the excess material fed to the machine at B. Forward movement of the chains now carries the fabric into a drying chamber O-P, where the moisture is driven out, so that dry fabric emerges from the chamber at P. This drying is very rapid as the moisture added by the sprays L and M and the steamers N is only about 10 per cent of the weight of the fabric.

As the fabric emerges from the drying chamber it is doffed from the pins by a high-speed revolving starfish-shaped wheel at J. At this point the material has been preshrunk in the warp and stretched in the filling.

The material is carried away from the tenter at J by a second pair of grasping rolls Q, which are operated through a variable-speed drive at R, which drive is direct from the main shaft D-E. The compensating roll S controls the variable-speed drive R so that the surface speed of the grasping rolls Q just corresponds to the amount of material delivered from the tenter at J. No undue tension is imposed on the material between Q and J.

The material now passes from the grasping or feed rolls Q to a modified Palmer drier equipped with a rather thick woolen belt and with other important new adjuncts to be described. Immediately after leaving the feed rolls Q a spray is applied, generally to the under side of the fabric, and sometimes steam also with the spray, to soften it and the adhesions that may have occurred in it from the use of sizing. Only sufficient moisture is added to render the material pliable and soft. This softening of the material as a whole causes a decrease in the width of the fabric. If the fabric has been tightly stretched in the tenter, the width shrinkage under the influence of the moisture at Q is considerable. The material is drawn away from the feed rolls Q at U. The speed at which the material is fed into the modified Palmer at U is in excess (say, about 6 per cent) of the speed of delivery of the material at Q. This causes a slight stretching of the material between Q and U so that it is desirable to place at S two rollers mounted on ball bearings which are revolved by the fabric and which tend to keep the filling straight. During the passage of the fabric from Q to S, the moisture and steam, together with the tension, exert a narrowing effect on the material. To prevent the fabric becoming too narrow, a series of adjustable scrimp bars or expanders is inserted at T, which may be so operated as to hold the goods out so that they enter the modified Palmer feed the exact desired width.

At this point U, however, the warps have become slightly straightened from their condition of high crinkle when the web was delivered to the point U, and it is now not only necessary to dry and finish the material, but also desirable in the case under consideration to put back the preshrinkage in the warp. This is accomplished by that part of the machine shown in Fig. 4, as follows:

Referring to Fig. 1, the differential between the surface velocity of the large heated drum of the modified Palmer and the rate of input of fabric at B (to the tentering machine) is the measure of

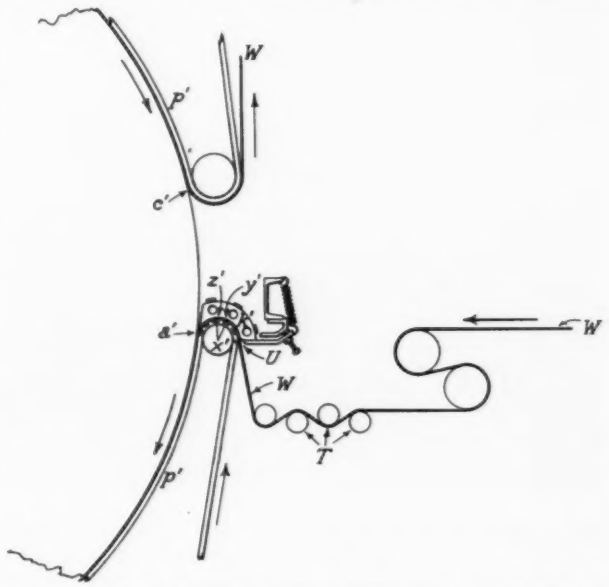


FIG. 4 PART OF MACHINE WHERE PRESHRINKAGE IN THE WARP IS RETURNED

the final, or net, warpwise preshrinkage attained. A variable-speed drive from the main shaft is provided at E', for adjustment of this differential. If the preshrinkage (warpwise) imposed in and by the tentering machine is to be exactly restored in the modified Palmer, the surface velocity of the modified Palmer will be adjusted to equality with the velocity of the tenter chain. If, however, it is intended to impose, in the modified Palmer, a larger percentage of warpwise preshrinkage than that produced in the tentering machine, the velocity of the drum will be by so much the less. The large drum (see Fig. 4) drives the belt p', with the result that the surface of the belt in contact with the large drum assumes the same speed as that of the drum. The material W, having passed through the adjustable scrimp bars T,

runs onto the belt at U, at which point the belt is flexed around a relatively small revolving cylindrical roll x', thereby stretching the surface of the belt at y' between the points U, y', and a'.

It may be assumed for purposes of this exposition that the median longitudinal plane Z' of the belt travels at constant velocity equal (or very nearly) to the surface velocity of the large drum. At y', where the belt is abruptly flexed, the speed of the outside surface of the belt is to the mid-section speed of the belt as the radius y'-x' is to the radius z'-x'.

The material W is caused to adhere frictionally to the surface

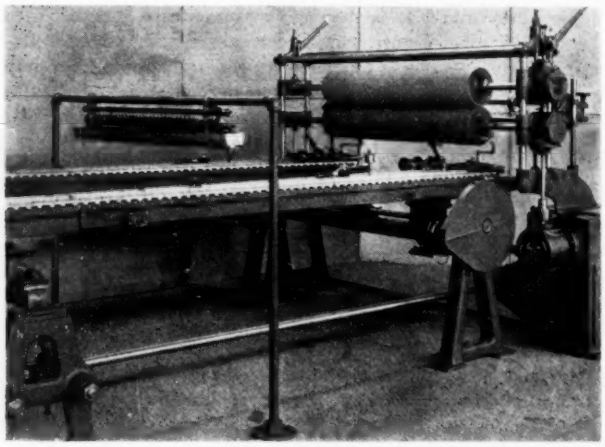


FIG. 5 INTAKE END OF SANFORIZING MACHINE

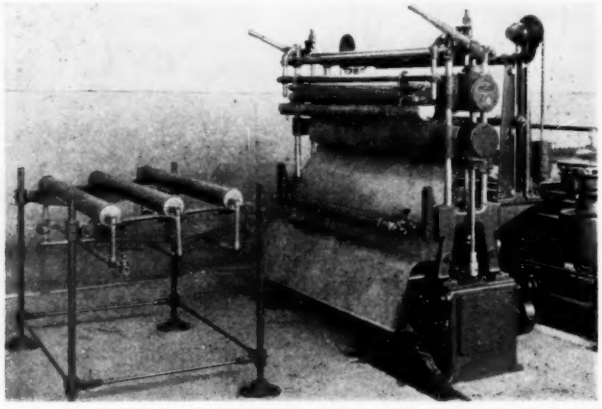


FIG. 6 TAKE-OFF ROLLS AT END OF TENTER, WITH THE ATOMIZER AND STEAMER

of the belt between U, y', and a' by the light pressure of an electrically heated iron k' which rests on the fabric mostly by its own weight, and which extends across the full width of the machine in short sections of 4 in. or more. The material W on coming to the point a' is fed at the higher velocity, as stated, to the surface of the hot drum, which is moving at the slightly lower velocity, approximately equal to that of the mid-section Z' of the belt, and the material would at this point wrinkle were it not for provision for maintaining fixity of the web to the belt surface up to, at, and through the line where the curvature of the belt reverses; these being, in actual apparatus, the maintained pressure of the irons k' and the location of the nip-line between roller X' (which is hung so as to press toward the large drum) and the large drum. When the belt reverses its flexure, its supporting surface shortens longitudinally at a', and inasmuch as the material W is held practically fixed to the belt surface, it also shortens with the belt surface and travels along between the hot drum and the belt until it emerges at point c'.

But inasmuch as the velocity of the surface of the drum and of the belt at c' is less than the velocity of the fabric feed at B (Fig. 1) by an amount equal to the shortening that is desired in the fabric, the fabric at c' is shortened by exactly this amount; the heat from the hot drum has dried the fabric so that it emerges at c' (Fig. 4), containing only a small amount of moisture, generally in the form of steam vapor.

The fabric during these operations has become shortened in length and decreased in width. The weaving undulations in the yarn have become accentuated, resulting in a "bead" that gives increased diffusion of reflected light. The drying of the fabric against a polished hot drum has moreover given to the fabric a smoother appearance and better "hand" than it had originally. It will further be noted that this process disturbs to only a small extent the fibers of the yarns, in consequence of which and be-

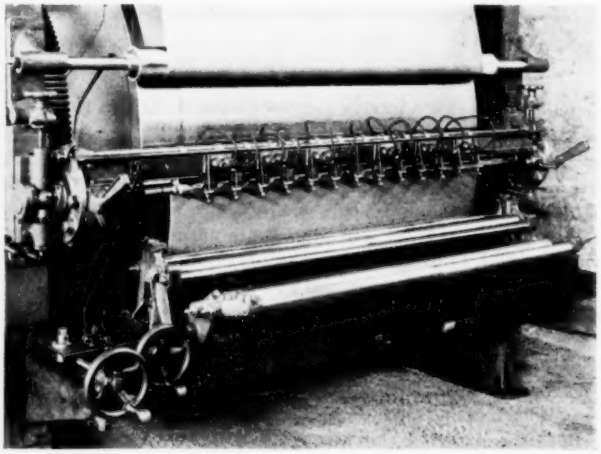


FIG. 7 MODIFIED PALMER FEEDING MECHANISM

cause of the increase in yarn crinkle a diffused luster is obtained. This is a natural luster which is ordinarily enhanced by the subsequent dampening and ironing that occur during the manufacture of a garment.

There may be or there may develop, of course, situations in which the modified Palmer and its adjuncts will be sufficient to perform the shrinkage operation without the use of the Sanforizing tenter frame. In the Palmer machine, modified and equipped as described, if a belt of increase thickness is employed, an increase in the rate of feed of fabric at U will be obtained, while the rate of delivery of the fabric from the modified Palmer will remain unchanged; but whatever the thickness of the Palmer belt and the resulting ratio of input fabric length to output fabric length, that ratio is fixed, and will be appropriate to only one fabric, and will serve only one predetermined purpose, for the inflexibility of this portion of the complete apparatus is contrasted with the inevitable non-uniformity in most carefully woven fabric. However, it has been found, in practice, that for the purpose of handling a large variety of fabrics previously finished in a large variety of ways, the combination of the whole machine, as shown in Fig. 1, has a much greater range, capacity, and adjustability than the section between the second feed rolls Q to and including the modified Palmer. The tenter prepares the material for the first operation in connection with the modified Palmer, straightening the filling threads (if previously curved), bringing all of the fabric to the most convenient and constant width, and at the same time shrinking warpwise

to almost any degree desired (up to 3½ in. per yard in the present machinery).

Referring to Fig. 4, the ratio of the outside surface speed of the belt at y' is to the mean speed of the belt as the distance y'-x' is to the distance x'-z'. For example, if the cylindrical roll x' has a radius of 1.5 in., and an extra heavy belt 0.34 in. thick be used, the distance x'-z' then becomes 1.5 in., plus 0.34/2 in., which equals 1.67 in. The distance y'-x' becomes 1.5 in., plus 0.34 in., which equals 1.84 in. The shrinkage or the shortening of the fabric on passing to the drum, expressed in terms of percentage, will be

$$\frac{1.84 - 1.67}{1.84} \times 100 = 9.24 \text{ per cent.}$$

This, multiplied by 36, gives a shrinkage of 3½ in. per yard of fabric. If the large drum is made relatively smaller, the surface of the belt against the drum becomes more compressed, resulting in a further shortening of the fabric held between the belt and the drum.

To operate individually the apparatus comprising only that from the second feed rolls Q to and including the modified Palmer, so that fabrics may be shrunk in warp and filling to a predetermined extent, it is only necessary that the belt of the machine, as illustrated, be sufficiently thick in comparison with the cylindrical roller x' to take care of the maximum shrinkage that it may be desired to impose on a fabric. Smaller degrees of shrinkage or shortening of the fabric are obtained merely by the adjustment of the variable-speed drive R controlling the speed of the feed rolls Q, so that the surface speed of the feed rolls Q is in excess of the surface speed of the hot drum by an amount equal to the desired shrinkage. If the speeds were exactly equal, there would be no shortening of the fabric; the latter would simply be stretched between Q and U an amount equal to the amount that it is shortened between U, a', and c'.

TABLE 1 CHARACTERISTIC RUNS ON SEVERAL SAMPLES

Material			Before Sanforizing			After Sanforizing		
				Shrinkage in in. per yd.			Shrinkage in in. per yd.	
Name	Count in gray	Color	Width, in.	Warp	Filling	Width, in.	Warp	Filling
Broadcloth	164 × 84	White	36	2.10	0.25[a]	36½	0.00	0.10[a]
Broadcloth	152 × 80	White	35¾	1.95	0.15	36	0.25[a]	0.15
Broadcloth	128 × 68	White	36	2.10	0.25[a]	36½	0.00	0.10[a]
Broadcloth	108 × 52	White	35¾	1.35	0.50[a]	36½	0.18[a]	0.00
Broadcloth	136 × 60	Blue	34½	1.80	0.00	35¼	0.12[a]	0.00
Broadcloth	136 × 60	Tan	35	1.65	0.35	35¼	0.15[a]	0.10[a]
Broadcloth	136 × 60	Green	35½	1.80	0.25	35¼	0.12[a]	0.05[a]
Oxford	88 × 40	White	33¾	0.70	0.20	35½	0.20[a]	0.06
Madras	80 × 80	Blue	36	1.65	2.20	35⅛	0.15[a]	0.30
Madras	80 × 80	Tan	36	1.90	2.00	35¼	0.10[a]	0.50
Madras	80 × 80	Green	36	1.87	1.95	35¼	0.25[a]	0.30
Flannel	.....	Blue	36	1.65	0.95	35¾	0.25[a]	0.15
Flannel	.....	Tan	36	1.95	0.90	35¾	0.00	0.05
Flannel	.....	Gray	36¾	2.15	0.50	35¾	0.33	0.00
Army poplin	112 × 60	Olive	35½	0.85	0.65	35¼	0.35[a]	0.00
Chambray	80 × 72	Blue	36½	1.40	1.75	35¼	0.18[a]	0.30[a]
Basket weave	76 × 76	White	33¼	0.90	0.70[a]	34½	0.35[a]	0.40
Basket weave	76 × 76	White	33½	1.00	0.90[a]	34¾	0.45[a]	0.30
E. & E. madras	80 × 80	Blue	35½	1.60	1.50	35	0.12[a]	0.00
E. & E. madras	80 × 80	Tan	35½	1.55	1.75	35¼	0.00	0.10
E. & E. madras	80 × 80	Green	35½	1.60	1.65	35½	0.05[a]	0.15
Fancy broad.	.....	Blue	35½	0.95	2.15	35⅛	0.40[a]	0.75
Fancy broad.	.....	Tan	35½	0.95	2.00	35⅛	0.00	0.65
Fancy broad.	.....	Green	35½	1.00	2.15	35⅛	0.00	0.75
Woven madras	80 × 80	Blue	35½	2.05	1.55	35	0.05[a]	0.12
Woven madras	80 × 80	Tan	35½	1.85	1.30	35	0.15[a]	0.25
Fancy madras	80 × 80	Blue	35½	2.55	1.05	35¼	0.15	0.05
Fancy madras	80 × 80	Tan	35½	2.60	0.85	35¼	0.00	0.12
Fancy madras	80 × 80	Green	35½	2.55	1.00	35¼	0.10[a]	0.12

[a] Denotes a gain.

Table 1 gives the data of characteristic runs of different fabrics in which it will be noted that while the warps are, in all the examples, shortened during the Sanforizing process, the material in the filling dimension has in some cases been lengthened.

As hitherto ascertained, rates of production, varying with the class of goods, on the machine equipment described are as follows:

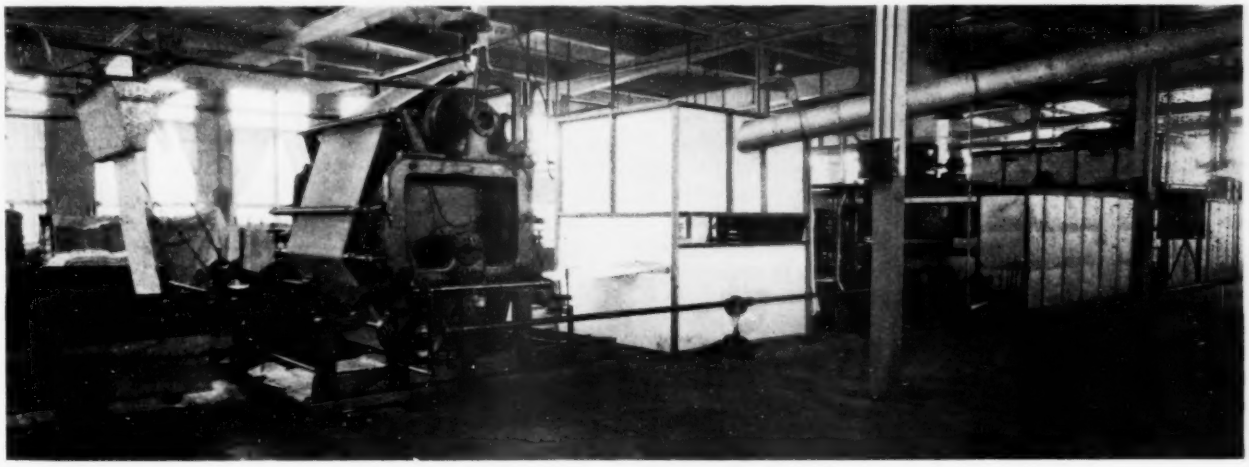


FIG. 8 SANFORIZING MACHINE IN OPERATION

Print cloth, 60 to 90 yd. per min.
Broadcloth shirtings, 60 to 80 yd. per min.
Duck, 8-oz., 30 yd. per min.

Three men are required to operate the equipment. Electrical controls at either end of the equipment are provided for the main drive and the moisture sprays.

Discussion

G. W. PAULSON.[2] The impression may have been gained that this process is a simple one and that the machinery might be installed and put in the hands of a few handy men around the mill to operate.

It so happens that the writer is the man who has to function as the "go between" to see that machinery installed is operating satisfactorily and that proper results are obtained, and he, there-

[2] Cluett, Peabody & Co., Troy, N. Y.

fore, is interested in the type of men selected to operate these machines.

The operating technique can be readily acquired by any one of average intelligence. On the other hand, there are hundreds of types of fabrics to be considered, and it is essential that, previous to Sanforizing, information be obtained as to how these fabrics had been finished and how much it is necessary to shrink them so that they will not change in dimensions when submitted to a laundry treatment. Men should be chosen to operate the machines who are above the average in intelligence.

TEX-55-3

Cause and Effect of Recent Advances in Textile Machinery

By EDWIN D. FOWLE,[1] NEW YORK, N. Y.

There is an important general advance in the improvement of textile machinery. The cause is mainly in the necessity which the machinery manufacturers face of developing equipment for replacement rather than for expansion. A slight falling off in the rate of development began this year, due largely to the great expense which the work entails, but the rate is still far above normal. If the industry will refrain from employing high-production machines to increase total output, that is to say, if it will use such equipment strictly to decrease operating costs, it will have a splendid opportunity to eliminate or reduce the overproduction which now hampers it. This is because continual renovation of the better mills will close the less efficient ones, and because reduced costs, followed by reduced selling prices, will widen the market.

A REPRESENTATIVE of a prominent textile-machinery firm early in 1931 declared "We have brought out more improvements the past year than in any like period in our history." The same statement might truthfully have been made by a number of other equipment builders; it could certainly have been made for the textile-machinery industry as a whole.

The year referred to, 1930, may receive special attention from students of industrial history as marking the apex of an astonishing advance in the improvement of textile machinery. For decades previous to 1929, developments appearing on the market were, as market reporters say, spotty. There were a few outstanding ones, it is true, but they appeared sporadically at comparatively long intervals and affected a relatively small number of processes. There was no general movement toward improving the technique of textile manufacturing in all departments of the mill, such as became evident in 1929 and 1930.

Several of the developments which took place before that period were of tremendous importance and probably were as vital to the industry as any of the multitude which followed them. Among the more recent of these earlier inventions were the first generally acceptable, and accepted, long-draft cotton spinning, single-process picking, high-speed warping, woolen ring spinning, and wrap-stripe circular-hosiery knitting. No doubt these innovations helped to awaken the majority of textile engineers to the possibility of improving many other processes of the mill. It would not be surprising if, in those comparatively cool and collected days, the average importance of the developments which were brought out exceeded the average of those which were the product of the subsequent more hectic, mad-scramble age, in which builders vied with each other to make the most startling announcements of new products. But, although this may be true on a basis of averages, it is probable that a number of the later improvements are individually the equal of the best which the earlier period produced.

While it would be difficult to reach any definite conclusions regarding the relative merits of improvements made at different periods, it is easily possible to compare their numbers. Since many changes in equipment were too insignificant to afford any appreciable advance in the art, it seems wise to take as a basis for comparison only those which the textile press considered of sufficient importance to mention in its columns. The years 1926, 1927, and 1928 were typical of those which preceded the upward swing, and over that period the number of developments so mentioned averaged 64 per year. In 1929 the number rose to 107, and in 1930, to 203. In 1931 it has fallen off somewhat, and it is estimated at this writing that it will be somewhere around 165. The reasons for this decline will be discussed later. Although these figures indicate that the era of development first showed results in 1929, it must be understood that the work of research and experimentation, which is the true basis of it, actually began several years earlier.

To mention all of these developments here would be to recite a rather wearisome catalog, and those who wish to obtain a detailed picture of the progress which has taken place are referred to the annual summaries which have been made by the Textile Division of The American Society of Mechanical Engineers, and by *Textile World*.

It would be of doubtful service at this time to endeavor to pick out the most vital of these stepping stones of progress, because the brief period which is now of principal concern, and in which the industry is still involved, began too recently to permit a mature judgment to be passed on the exact value of the inventions it produced. Certain developments, however, which are of sufficiently new departure to appeal particularly to the imagination include the single-unit full-fashioned hosiery machine, the sanforizer, the continuous kier, the "geyser" skein-dyeing machine, and the garnett-card. Who can say, however, that these innovations will prove of more importance to the industry than certain machine improvements which adhere more closely to established principles, such as the enormously increased capacity of the rulling mill, the adoption of the variable-speed-motor drive for the full-fashioned-hosiery machine, the introduction of the flexible top comb and the auxiliary top nipper knife to the cotton comber, the application of more blades and a cloth-turning device to the woolen and worsted shear, the redesign of cotton, worsted, and silk looms for greater speed, and so on almost, it seems, indefinitely.

It should be pointed out that included in the developments counted for comparison and otherwise referred to in this paper are only those which apply strictly to textile machinery, as compared with general industrial equipment applicable to a number of industries. Important advances have been made in the latter

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

class, but any detailed discussion of them here would not be pertinent to the subject.

Causes of Advance

Having given some idea as to the importance of the advance which has recently been made, it is logical to consider the causes which were responsible for it. It appears to be a fact, in certain respects unfortunate, that the mills themselves are responsible for comparatively little mechanical progress. The reasons for this are not difficult to understand. It is not easy for them to make basic changes in their machinery, and they are not equipped to build and experiment extensively. Their organizations are not especially trained in basic mechanical principles and they often do not have free access to other mills from which to cull ideas. Rather than devote time and attention to the fundamental development of machinery, they are more apt to make the most of their equipment as it stands. Even those mills which are the exception to this rule, and there are several, cannot carry on a complete program of development totally unaided.

Thus it is that responsibility for mechanical progress in the textile industry lies at the door of the machinery builder, and to him one must turn for an explanation of any irregularities which occur in such progress. When the war boom collapsed in 1921, the momentum of the expansion which the mills had conducted carried the machinery builders fairly comfortably for two years. Few mill men realized that the poor condition of business would be more than temporary. Surely the expansion should be continued while the industry awaited the return and higher upswing of the business pendulum. The machinery builders had also expanded. For some time their activity was sustained by hope. Gradually they began to trim their sails by consolidating their plants, closing certain departments, and economizing in other ways.

At the same time they found it advisable to take another step. Since lack of expansion had killed the demand for the old lines of machines, their main hope lay in developing something which mills would wish to employ in order to improve the quality of their work or to reduce their operating costs. In this way mills would be induced to scrap their present equipment and make further purchases of machinery.

No doubt careful study was made of those factors which render a new machine popular. Concisely stated, these are improved production, quality of product, flexibility, durability, and dependability, together with decreased first cost, upkeep cost, labor cost, power consumption, floor space, and danger. Consequently there began to appear upon the market ingenious new machines having one or more of these qualifications.

Managers of machinery plants who believed that only temporary measures were necessary, as well as some philanthropic managers, entered a program of research largely to supply work to idle hands. Others, always eager to undertake such work, found, in slack business, a golden opportunity to devote time and attention to an activity dear to their hearts. Still others were driven to adopt such programs in order to meet the competition from firms which had already done so.

Just as in the textile industry proper there has lately been introduced a younger and more alert type of direction, so in the machinery industry has conservative, tradition-adhering management largely given place to a less restrained, more progressive type. This in itself is responsible for some of the advance. At the same time, and partly as a result of the newer type of management, more attention has been given to merchandising problems by both textile and machinery men, with the result that manufacturers both have come into closer contact with the users of their products, and have developed a better understanding of their needs. More thorough knowledge of the factors involved in style changes, for example, has influenced builders supplying equipment to certain branches of the industry (such as, notably, the knitting branch) to develop machinery capable of meeting more accurately the changing tastes and demands of the public. Recent trends toward novelties in textile products have also made their demands upon the inventing genius of the machinery builder.

Another and later, though much less important, stimulus was possibly the three textile-machinery shows of 1930, held respectively at Boston, Philadelphia, and Greenville. Naturally every exhibitor at such affairs strives to make his booth the center of interest and the main topic of discussion among the visitors.

The 1931 Recession

The slowing down in rate of development which has taken place this year is not particularly marked, and the rate is still far above normal. The causes which created and fostered the advance are still active, but an important opposing influence has taken effect. This is the failure of the machinery companies, thus far, to obtain full return on the investments they have made in development work. Naturally their resources are not unlimited, and it is necessary that a reasonable return be obtained before much further progress can be made.

Many mill men seem to entertain the notion that a new machine is the immediate result of a brilliant idea suddenly conceived. Such is not the tempo of modern achievement. New equipment of originality is today the result of long and expensive research and experiment. Large engineering staffs are required. One machinery firm, for example, has in recent years expanded this department to eight times its former size. Much extra drafting work is required. Considerable material and effort are wasted in following through fallacious ideas or delusive lines of reasoning. Many apparent accomplishments are found to be failures from a practical standpoint, as was the case with some of the innovations exhibited at last year's shows. To go into production on new lines often calls for expensive purchases of new machine tools on the part of the equipment manufacturer. A further expense is the many new parts which the builder must carry in stock for any new machine.

Some mill men might look on the selling prices of new machines in a more tolerant mood if they understood the actual cost of the development behind them. One of the more active machinery firms has spent approximately $600,000 in improving and originating machinery during the last four years. This expenditure was distributed mainly over 34 new products, making the average cost per machine around $17,500. Another, considerably smaller concern, spends on work of this character, many times over $10,000 a year. On five new machines recently developed by still another company the total expense for engineering, drafting, and experimental work alone amounted to nearly $36,000, exclusive of new machine tools, overhead, and selling expense. The latter item is of considerable importance in connection with new machinery.

The following comments from a machinery builder are pertinent: "On one of our new departures from the stereotyped machines of the past, we have spent eight years in bringing it to perfection. This involved practically the entire shop at one time or another. The amount of research work, both mechanical and chemical, preliminary design, new patterns, and subsequent changes in these patterns, not to mention complete scrapping of original machines to replace something else that has been discovered during the manufacture of the first machine, all added together form but a small total of the amount of expense in money and labor that is involved in such work."

The mill men are not prepared mentally or financially to absorb this cost at once. They can do so only gradually over a long period of time. The builder must, accordingly, price his machines

attractively. One builder, for example, has sold the first 11 machines of a new model at a loss of around 25 per cent. It is no wonder then that progressive builders find themselves without sufficient funds to maintain the pace of development work for an unlimited period.

Other factors are to a less extent responsible for the recession. The furore of development in 1929 and 1930 left many rough spots which must be ironed out before further new departures in the same direction can be made. A number of innovations have still to prove their worth in practice. In some cases adjustments in preceding and following processes must be made before a specific development can find a permanent status. It appears also that the human mind is capable of only a limited progress in a given time and that inventive genius, particularly when under such pressure as has lately been the case, requires periods of rest. Furthermore, failure in attempted accomplishments, although stimulating to some minds, are more likely to be discouraging and to foster inactivity. Finally, the mills are confused and embarrassed by too radical a shake-up in established ideas and methods and must be given time to accept the new developments.

Effects of Advance

The probable effects of this general advance in machine improvement are interesting to contemplate. Purchases of machinery will undoubtedly be stimulated, but whether the increase will be sufficient to repay the builder for his effort for some time to come is debatable. At least the majority of firms are certain to survive long enough to reap benefits from it.

The resulting activity will be of some immediate benefit to general business. Probably 70 per cent of the money spent by textile mills for new machinery is absorbed by the employees and, under normal conditions, the stockholders of the machinery companies, while the remainder goes to the industries which supply the iron, steel, wood, leather, rubber, brass, copper, coal, oil, paint, and other raw materials. All workers in those industries are purchasers of textiles, and the money they spend for other things reaches other textile consumers.

The most general tendency in new machinery is toward increased production. Often an operation is now carried out at more than double the old rate of speed, thanks in some cases to the application of new mechanical principles and in many others to such simple improvements as the adoption of anti-friction bearings. The fear has been expressed that the textile industry, already overproducing, will find any improvement in the efficiency of its individual units more disastrous than helpful.

Any manager who makes the mistake of believing that in these times the benefits to be derived from high-production machinery lie in increasing output is steering toward disaster. It is exceedingly essential that high-production machinery be used to give the same, or less, total mill output. Of course this output will be attained at reduced cost. The possibility of cost reduction lies mainly in minimizing the requirements for labor, upkeep, and power.

All improved machinery does not offer increased production. Some affords mainly an improvement in quality of product or in the ease by which changes may be made in the type or character of product. No danger is seen in the installation of such machines, unless the improvements they offer are not sufficient to make the investment returnable within a reasonable period of time.

It has even been suggested that the machinery producer is forcing the obsolescence of present equipment to his own benefit, and to the detriment of the industry. Already short of funds, it is argued, the mills are suddenly called upon to make new and unexpected expenditures, which they cannot afford. There are two answers to this criticism, both exonerating the builder from exerting any detrimental influence, although not denying that he naturally has his own interests at heart.

In the first place, the purchase of improved machinery is not in the long run an expense to the mill. If it is worth installing, it pays its own way, returning to the mill all which was spent on it, and adding more. Often it permits a reduction in selling price on the mill product, with the result that the market is widened and overproduction is, in effect, reduced or eliminated.

In the second place, it is only the mills in at least a fair financial position which can afford to purchase the new equipment. Continual renovation of these plants will widen the gap between them and those competitors which are less efficient or which are faced by insurmountable obstacles, such as unfortunate location, for example. The result of this will be that one of the most frequently proposed solutions to the textile tangle will be accelerated—the mills which are not capable of operating profitably against the competition of their stronger neighbors will be forced more promptly out of the picture. Again, overproduction will be reduced or eliminated and the major cause of the disastrous price cutting from which the industry has long suffered will be removed. The mills which remain, therefore, will be able to sell what they produce at a fair profit.

Therefore, in the possibility of the mills failing to recognize the importance of reducing costs without increasing total output lies the only danger. The mills may feel that an up-to-date plant can afford to increase its production in order that it may assume some of the business of the departing inefficient plant. It must be remembered, however, that shaking off the shackles of ruinous competition is, like the working of that law of evolution known as the survival of the fittest, a gradual process. Any appreciable attempt to take advantage of it too early will obliterate all the potential gain which it creates. Instead of eliminating only the inefficient mills, it will become necessary to eliminate many of the better ones as well.

Furthermore, it is unfortunately true that the elimination of a mill does not necessarily remove from competition all the machinery which it contains. In this connection it is interesting to note that, of the machinery in the many knitting mills of Brooklyn, New York, which have gone out of business in recent years, practically all of it is in operation today, although in different hands. Far too little scrapping of machinery is undertaken by the textile industry, not only in the case of mills which close down, but also in that of mills which replace obsolete machinery with new. It takes a long-distance view into the future for a manufacturer to appreciate the ultimate importance to himself of reducing his replaced machinery to scrap. Its resale value for active production is naturally tempting in view of the fact that the income so received reduces in effect the actual cost of the new equipment. Yet more farsightedness in this respect is needed, and one must live in hope that before very long this fact will receive the full recognition it deserves.

It is therefore unsafe at this time to draw up a workable estimate of any decrease in production which the current improvement in machinery may cause. Instead, it is safe to assume only that, under any circumstances, there is likely to be an overproduction for some time to come, and that consequently an attempt to increase the output of plants must not be made. It will be far better to err on the side of creating underproduction than to maintain present conditions.

Another effect of the advance is an intensifying of the problem of allowing properly for obsolescence. Recent study of the methods employed by mills for anticipating and deciding upon replacement of machinery has disclosed an appalling carelessness and even lack of understanding regarding this matter. Depreciation reserves appearing on income-tax returns do not actually exist. No systematic methods for keeping track of the condition

of various units of the mill machinery are adhered to. Many mills make no special effort to investigate and study new equipment. Very many lack the knowledge of costs needed to calculate when a new machine is capable of paying for itself. The current advance in machinery improvement is bringing home, as nothing else could, the vital necessity of paying attention to these things.

Due to the fact that even the stronger mills are in a none too healthy condition financially, one more result of the advance is likely to be the general adoption of a policy of buying equipment on long-term credit. Since the machinery builders lack the resources to supply an unlimited amount of credit, the assistance of credit companies must be solicited.

Discussion

ALBERT PALMER.[2] In the weaving-machinery business the period of activity described by the author began in 1924. In that year, for the first time in 20 years, the number of weaving patents issued exceeded 300. The number increased annually to over 400 in 1926 and then dropped somewhat during the succeeding years. The average annual number issued, however, from 1924 to date is almost twice the average annual number issued during the 10 years prior to 1924.

Coincident with this increase in the number of new inventions was the activity of redesigning existing equipment. In the essentials of construction, looms were no different in 1924 than they were 40 years earlier; in fact, some of the parts used on looms built in 1924 were made from patterns that were about 40 years old.

As the author says, the incentive and the opportunity to produce new looms came with the recession which started in the industry during the latter part of 1926. Realizing that there are thousands of worn-out weaving machines in operation that should be replaced, the builders undertook to construct machines that would produce at a low cost the highest quality of goods and that at the same time would give great flexibility as to the type of fabric woven.

To redesign completely a machine like a loom costs between $75,000 and $100,000. That such a large amount of money is required can be appreciated when consideration is given to the work that must be done.

In a relatively simple machine such as a cone-harness non-automatic silk loom there are about 770 pieces, of which 33 per cent are castings, 14 per cent are shafting and forgings, 6 per cent are wooden parts, and 47 per cent are miscellaneous items such as studs, collars, pins, etc. Each of these pieces must be considered from the standpoint of cost, quality, and suitability to the purpose for which it is used. In practically every instance changes are necessary, with the result that money must be expended, not only for the original study of the part, but also for drawings, patterns, jigs and fixtures, and often for machine tools by which to make the new parts.

The mills in general do not realize that the selling prices of new machinery at the beginning do not include any profit or even a share of the development expense. In fact, until quite a substantial number of units have been built, the selling price does not even cover the factory cost, mainly because the plant has not been converted to the new line of manufacture.

The chief problem of selling new machinery is to obtain a sympathetic hearing from the textile industry. While many mill executives are anxious to improve their methods, there are others who actually resist the efforts of machinery builders to introduce new ideas. Instead of welcoming the opportunity to cooperate, they place the burden of proof on the builder and force him to work under the most unfair odds. The result is that the builder is put to unreasonable expense, and sometimes, because of the opposition of mill employees, is obliged to withdraw from the project entirely.

The author's statements with regard to the necessity for preventing an increase in the productive capacity of the industry are correct. So are those with reference to the matter of obsolescence and the handling of depreciation reserves. That more knowledge concerning these matters of cost is needed is demonstrated by contact with the industry.

EUGENE SZEPESI.[3] The progress made in the construction of textile machinery within the past few years is of vital importance, not only because it represents a technical advancement of the textile industry, but also because these events bring to the foreground the fact that the economic ills of the industry have been consistently ascribed to consequences, instead of to the originating causes.

The builders of machinery and equipment have been accused for the past quarter of a century of backwardness. This was unquestionably true, because if we survey the technical progress of the American industries, we find with minor exceptions the textile industry decidedly behind the other branches of industrial arts. But this is already an effect and not a cause. The originating cause is due to the attitude of the producing factors of the industry, because no progressive development can be expected in any industrial branch where a 20- or even a 30-year-old equipment is considered just as good as new. This fallacious conception is mainly responsible for the past backwardness of technical developments in the industry as well as for its chronic economic plight. Behind this is hidden much of the overproduction of unprofitable merchandise, because the unflexible equipment cannot respond quick enough to the changes of the demand. There is also the horizontal overexpansion of the industry during the past decade instead of a vertical replacement, which helped to bring about the present situation. And last, but not least, the universal transgression of the economic laws, expressed, as an over-depreciated equipment, with the funds expended on everything but its original purpose, has acted as the keystone to the present situation.

With all these facts in mind, the notable advance made by the makers of textile machinery within the past few years, no matter how gratifying, comes unfortunately too late to serve as the chief medium for the economic rehabilitation of the industry. No question about it—it will be contributory, but on the whole it can play only a minor role.

To begin with from a financial point of view, the industry was never less prepared than it is today for a radical replacement of its basic equipment. It needs no argument, the equipment made today is far superior to the cross-section still in use by the industry at large, but as it is today, the difference of the cost of production on new equipment and of the equipment in use is not sufficient to warrant an immediate discard of most of the equipment. This means that the replacement procedure must be gradual, and a decade is by no means a too conservative estimate for a visible effect of such new equipment on the industry. Even this will be possible only if the machine manufacturers succeed through coordinated and intelligent effort in eliminating the present detrimental practices and misconceptions concerning depreciation which is so deeply rooted in the industry.

To make a visible progress in the replacement of old machinery, the rates of depreciation must be increased by the individual manufacturers and the accumulated funds systematically applied for its only purpose, the replacement of equipment.

Without such intermediate steps, the machine manufacturers

[2] Research Assistant, Crompton & Knowles, Worcester, Mass. Jun. A.S.M.E.

[3] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

cannot continue their costly experiments and the investments essential for the production of improved equipment. Without sufficient sales in sight for new equipment a reaction is bound to follow.

There we have a wheel within a wheel with an undeserved disappointment in store for the makers of textile machinery, unless they can work out the economic phases of their distribution problem through a collective action, which as far as is known, has not yet been undertaken.

Now for the other phases of the economic possibilities of the textile industry As has been stated, the advance made in the construction of the basic textile machinery represents only a part of the probable improvements of production and the reduction of operating costs.

The handling of the product between the basic conversion operations is hopelessly behind times if compared with other industries, and it is amazing that machine manufacturers have so far failed to recognize the golden opportunity of business that would follow the development of up-to-date and scientifically constructed secondary equipment for the textile industry.

The equipment used in our mills for the handling of laps, sliver, roving, yarn, and cloth has not been improved for generations. It is not an exaggerated estimate that 50 per cent of waste and imperfect products is due to negligence and improper handling of the product between the operations. Similarly, production losses during the spinning conversion and weaving operations are to a notable degree due to the same cause. Improvements in this direction would bring about a notable advantage in quality of product, increased production, and lowered cost of production, and at a decidedly lesser investment of additional capital than would be required by the replacement of the basic equipment—hence, more in conformity with the present economic ability of the industry.

Finally, there is another direction toward better and more production, particularly of the spinning operations, which is today only superficially exploited by the industry. This is the functionalization of the roving and spinning operations. If this opportunity would be exploited to its fullest possible extent, the results would in many instances mean as much as the replacement of the old equipment by new ones, but with the continuation of the present methods of operation.

The principle of this possibility is the functionalization of the service to the machinery in such a manner that the complex duties of the spinning operators are separated in such a way that a worker looks only after one particular service. Using this principle the workers are not assigned to a definite number of machines, but move from machine to machine continuously in the form of an endless chain.

For instance, according to this inverted conveyor principle (because instead of the machines moving to the worker, the worker moves to the machines) the "piecer-on" devotes her time exclusively to the mending of broken thread, followed at a proper distance by another "piecer-on." The distance between them depends on the size and the grade of the yarn, and is determined by the minimum allowable time a spindle should be permitted to remain idle—hence, maximum production and minimum waste are the consequence. Should the roving run on occasions below the standard, this fact is immediately known, because the distance between workers will increase, and a reserve operator can be immediately inserted into the chain to take care of the deficiency, and thus reestablish standard production, standard quality, and standard waste immediately.

The same principle is used for all of the other functions of the spinning operations, such as the replacement of the roving, the cleaning of rails, clearer rolls, etc., and the cleaning of the machine itself.

This is only a sketchy illustration of the possibilities of functionalization of many of the operations in the textile industry, and it is up to the members of the engineering profession engaged in the textile industry to bend their efforts toward such constructive directions as functionalization, the improvement of the secondary equipment, and the development of proper plans for the replacement of the prime equipment. This would be their share of contribution toward the economic rehabilitation of the textile industry, not only individually, but also collectively. The engineers should bend every effort to cooperate intelligently with the machine makers as well as with the producers to bring about a well-balanced progress in the future of the textile industry.

Progress in Wood Industries

Wood-Utilization Research at Forest Products Laboratory

Arthur Koehler.[1] A new surface treatment was developed for nails by which their resistance to withdrawal was made from two to three times that of plain nails. This treatment is from 30 to 50 per cent better than the cement coating now commonly used.

It was found that by employing the proper technique, glued laminated beams could be made with horizontal joints that would not fail by shear in the joints. This type of beam construction offers the possible use of high-grade material along the top and bottom, with inner layers of lower grade.

In the search for chemical treatments to control the shrinking and swelling of wood several were found that possessed high effectiveness, but all of these made the wood attract moisture at high humidities and become too damp for practical use. The search is being continued, with the object either of discovering effective chemicals that are not too hygroscopic or of discovering satisfactory ways of reducing the hygroscopic properties of the wood after treatment.

Compounds with the basic radical ammonium, aluminum, lithium, magnesium, tin, or zinc and the acid radical arsenate, arsenite, borate, molybdate, or phosphate were found to be particularly effective in reducing the hazards of wood from the standpoint of ignition and spread of fire.

A detailed discussion of the strength and serviceability of wooden crates was prepared for publication as a bulletin of the Department of Agriculture.

The effect of temperature on the seasoning degrade of beech, birch, and maple small-dimension stock was determined to the end of establishing the maximum temperatures which may be safely used in drying this class of material.

A new wood hygrostat employing the swelling and shrinking of a thin strip of wood for the control of the humidity condition in a dry kiln was perfected.

In the study of lumber storage and handling problems, it was found that the effect on changes in average moisture content of lumber was negligible between unheated open and unheated closed sheds, provided both types of shed afford adequate protection against rain. Some heating was found necessary under most conditions of lumber storage in order to maintain a moisture content in the wood as low as 7 per cent.

Lumber Manufacturing

Frank P. Cartwright.[2] There is a strong trend toward greater refinement or more complete fabrication of lumber at the mill. Examples of this are the guide-line framing and four-square lumber available, the former being marked in feet and inches for convenience of measurement and the latter being squared to exact length.

Wilson Compton.[3] U. S. Census reports on footage of lumber cut in 1931 are not yet available. Estimate has been made by the Federal Reserve Board of production of 16,400,000 ft. It is estimated that replacement growth is from 7 to 10 billion feet a year.

The Timber Conservation Board estimates that consumption of lumber, including domestic and exports in 1931, was approximately 18,710,000,000 ft. Imports totaled about 700,000,000 ft. Some of the consumption in 1931 was of lumber taken from stocks which decreased appreciably during the year. Stocks at the mills decreased from 10 to 20 per cent; at retail yards probably about 10 to 12 per cent, during the year.

The lumber industry in 1932 was at its lowest point in many years in volume of production, prices at the mills, and consumption. This was largely due to the decline in building and especially in residential construction. Building contracts were only about 40 per cent of what they were in 1931, which in turn was a low building year, as compared with 1928 and 1929. The railroads, which customarily use directly about 15 per cent of the lumber cut, purchased only about one-third as much lumber and tie material in 1932 as in 1929 and ordered only a small percentage of new freight cars as compared with that year. The farmer and small-town resident used less lumber than normally, due to low level of farm prices and the farmer's financial inability to erect new buildings or to keep up adequate repairs.

After building, including sash, doors, millwork, interior trim, flooring, etc., more lumber is normally used for boxes and crating than in any other industry. Boxes and crates shared in the general decline in 1932, but less severely than many industries, since its volume depends largely upon the movement of merchandise, of fruits, vegetables, etc.

Pursuant to the recommendations of the U. S. Timber Conservation Board, lumber production has been restricted in line with demand and stocks have been reduced during the year, thus placing the industry in a stronger statistical position than a year ago.

Wood Utilization

Axel H. Oxholm.[4] The most important development is the work on metal connectors for wood construction, the culmination of 14 years of constant research and labor. This is a cooperative undertaking with the Forest Products Laboratory. It is certain that these metal connectors can be worked out to advantage in connection with wooden parts of machinery, agricultural implements, boxes and crates, and perhaps also for furniture. So far we have concentrated on the building construction aspect.

Another matter of considerable importance is the study of nonutilized wood. We have just finished up the New York survey this year, and have previously covered the States of Virginia, North Carolina, and Maryland. We can show a number of excellent examples of how these reports have stimulated the utilization of wood waste.

The prefabrication of houses has received an important impetus, following similar developments in the steel industry. A dozen concerns in this line are now developing plans for standardized prefabricated homes.

Fireproofed Wood

Ernest Hartman.[5] The Forest Products Laboratory is continuing its studies of chemicals useful in the fireproofing of

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[1] Forest Products Laboratory, Madison, Wis.

[2] American Forest Products Industries, Inc., Washington, D. C. Assoc-Mem. A.S.M.E.

[3] American Forest Products Industries, Inc., Washington, D. C.

Presented at the National Wood Industries Meeting, Jamestown, N. Y., November 15 and 16, 1932, of The American Society of Mechanical Engineers.

The Wood Industries Division's Progress Report for 1932 was compiled under the supervision of Arthur D. Smith, Jr.

Personnel of the Executive Committee of the Wood Industries Division for 1932: Chester L. Babcock, Chairman; Arthur D. Smith, Jr., Secretary; A. S. Kurkjian, R. K. Merrill, H. B. Carpenter. Associates: T. D. Perry, G. R. Petrie.

[4] Director National Committee on Wood Utilization, Department of Commerce, Washington, D. C.

[5] Protexol Corporation, Kenilworth, N. J.

wood. The results are recorded annually in the proceedings of the American Wood Preservers Association. Further progress has been reported by Committee C-5 of the A.S.T.M. in testing the fire retardance of structural materials. Work being carried on under the direction of Prof. A. H. Beyer, of Columbia University, aims to measure not only the fire retardance of doors of fireproofed wood, but also to measure the heat transmitted, as well as the volume and character of the smoke passed by doors during test. A gas-fired furnace for testing wood-door assemblies has been constructed during the past year in the metropolitan area, still further attesting progress in the field of retarding the ignition and combustion of wood. A special committee of the National Fire Protection Association during 1932 brought in a report in which it is recognized that the fire hazard is reduced as a result of using fireproofed wood. No new processes or materials have been presented during the past year.

Dry-Kiln Progress

Kenneth Redman.[6] The year has been marked by a more widespread use of forced recirculation of the air by means of internal fans—i.e., fans placed inside of the kiln. The kiln manufacturers are advocating air movements far in excess of the 25 ft per min speed which has been suggested by the Forest Products Laboratory as the maximum speed required in drying hardwoods. The slight extra speed in drying is accompanied by large increase in mechanical horsepower to drive the fans, so that it would seem that the laboratory recommendations still remain quite sound practice. Anything that increases the cost per thousand of drying can hardly be classed as an improvement.

On the other hand, forced circulation to insure the 25 ft per min rate is possible at very slight cost. This may be put in for roughly $6 per lineal kiln foot with an electric load of $1^1/_4$ hp per 100 lineal kiln feet. Such an investment is quickly amortized by increased production, whereas a power load of 21 hp per 100 kiln feet as is advocated in certain instances is likely to be a dubious investment.

Between these extremes is a recent effort to introduce a forced circulation into the center of each truck load with a motor-driven unit blowing into a plenum chamber. After a year's trial we will know more about it, but several questionable mechanical features, such as keeping the motor cool, are at once apparent.

In addition to more air movement, the control of such air movement is being given more careful attention, and far better results are being secured.

As regards kiln practice in general, it is safe to state that higher temperatures are being used in most hardwood drying, toward the end of the kiln run. This is not only safe, but invariably results in stock with less stress than where low final temperatures were used.

Furniture Manufacturing

G. Rollo Petrie.[7] It would seem that the most significant progress to be credited to the furniture industry during 1932 was the beginning of the realization that the unit costs of the product have less influence on the net profits than had been previously taken for granted.

During the year, more actual consumer-demand analysis investigations have been instigated than for probably any similar period, and while practically all of these attempts toward research-controlled merchandising have been initiated by advertising agencies or publications and by large retail organizations, it is nevertheless significant that they have been started, as they are certain to be continued and enlarged by their adoption by the industry in general, probably through the trade associations.

Better lumber handling, better machining and gluing practice, and better finishing methods are not only lowering production costs, but are giving the furniture consumer better utilization value in his purchases and more satisfaction with them.

The use to which new furniture can be put, with the pleasure which can be derived from its ownership, is beginning to be made the basis of appeals in the endeavor to influence the spending of more consumer dollars for furniture than for other articles of intimate use and possibly more actual utility convenience.

It has been seen that the proper course to pursue is that of increasing the popular desire for furniture and not that of each individual manufacturer's simply striving to secure a greater portion of the existing demand by price cutting or the sacrifice of quality, or both.

Designs seem to have been toned down somewhat on the average, and a distinctly American motif is more apparent than for some time, both with respect to the popular Early American or provincial pieces and to the increasing popularity of a modified modern type in which the appeal depends on simplicity and a clean-cut balance of each component part.

There is also the appearance of a faint recognition of the fact that because the home across the street has a certain suite made by a certain manufacturer is no reason why an exact duplicate should not be purchased. This trend is belatedly following that of the automobile mass-production idea which showed long ago that the advantages of the production of duplicate items are generally appreciated and the fact that many others own identical units is not a deterrent to pride of ownership.

It would seem as though this realization would tend to bring back the advantages of the large- and medium-sized factories as contrasted with the recent almost actual price control of the small and oftentimes unreliable plants which could afford the rapid and bewildering style changes better than their larger and more stable competitors.

The increased success of efforts toward less frequent market periods is another step toward more efficient marketing and manufacturing methods (through more stable styles), as is also the gaining popularity of the possible advantages of admitting consumers to these central market exhibitions at times which would not conflict with the scheduled dealer buying periods and for the purpose of creating consumer desire by their viewing of the latest "models."

Millwork and Doors

Sern Madsen.[8] An amazing number of patents were granted on new window developments. Probably 99 per cent of them will never be developed further than obtaining the patent. The efforts have been devoted mostly to obtaining narrow center mullions for multiple windows, improved weatherstripping means, balancing means, and means for washing the outside of the window from the inside. These features apply to the double-hung type of window.

There have also been some developments on casement sash. Recently a new type of casement window was brought out, which is provided with a metal-framed pane of glass which is inserted to swing with the window proper and serves as a storm sash or a double-glass arrangement. It can be removed easily for window washing. A metal-framed screen is used inside for keeping out insects. In this window at least is shown a tendency to combine the use of metal and wood to a greater extent. This window comes completely weatherstripped. Dividing bars are aluminum.

An outstanding contribution of the millwork industries has been a new window. This window is very effectively weatherstripped, cannot stick in the frame, will not rattle, and has elimi-

[6] Redman Lumber Seasoning Service, High Point, N. C.

[7] St. Johns Table Company, Cadillac, Mich. Mem. A.S.M.E.

[8] Curtis Companies, Inc., Clinton, Iowa, Mem. A.S.M.E.

nated the use of weights, cords, and pulleys. This window makes possible the use of very narrow trim and narrow mullions. It also has other valuable features, such as improved sealing of frame to wall joint.

Another outstanding development is a "miter-tite" window and door trim.

During the last year there has also been some development in the way of overhead-garage-door hardware. This development really started about two years ago, and now many companies have developed hardware or some sort or other for making one-piece garage doors lift upward and overhead inside the garage.

It is quite apparent that, with reduced volume in the millwork business, millwork manufacturers have devoted more attention to development of new products.

Plywood and Glues

Thos D. Perry.[9] There has been a marked change of attitude on the part of plywood manufacturers in the United States toward the hot-press, dry-glue method of assembling plywood which has been successfully practiced for some years in Europe. This change has found expression in extensive research work along lines of wider markets and simpler processing, and has led to several initial installations of hot-pressing plywood equipment. The industry, as a whole, is only on the threshold of this far-reaching change.

It becomes more and more apparent that the old method of cold-press wet glue was anything but logical, since it required the meticulous drying of all wood parts to approximately normal 5 per cent moisture content preparatory to applying the fluid glue that would carry this moisture content up to 15 per cent and sometimes 20 per cent during the pressing operation. The internal stress and strain introduced during the water-acquisition, wood-expansion step had to be later neutralized or eliminated as far as possible by a very careful redrying, which in turn might set up different internal stresses and strains.

Compared with this illogical process is the new and simpler way with its pre-drying of material to be glued to known moisture content and its application of sheet or powdered glue. This results in a complete and final glue joint in the hot press, giving a sheen or luster to the face veneer, and requiring no subsequent redrying or attempts to neutralize interior stresses or strains. The product comes out much flatter, and the heat element permits a completely waterproof bond.

The glues used in this hot-press, dry-glue procedure may be of the phenolic-resin type, which comes in sheet or powdered form, or animal glue, or a mixture of casein and latex. These latter two are prespread on the cross-band or other glue-receiving member and predried before pressing.

These plywood developments are presenting a wide range of opportunities never before available for plywood products, and it is the belief of pioneers in this new line that the plywood industry will gain immeasurably by these decidedly evolutionary, if not revolutionary, changes in factory processes.

Woodworking Machinery

Chester L. Babcock.[10] The development of woodworking machinery during the last year has been retarded by the lack of demand for equipment due to the economic depression, which has caused the liquidation of several woodworking plants, making available much modern equipment at reduced prices.

The small short-coupled 4-in. and 6-in. electric molders and automatic shapers and multiple carving machines of from 10 to 24 spindles have been in greatest demand, and these types of machines will continue to be improved in an effort to enable woodworkers to lower costs and realize a profit, and yet meet the incessant demand for lower-priced goods.

Blower systems with circuit mains using flight conveyors within the airtight main pipes are gradually, due to economy of operation and adaptability, replacing the systems which depend entirely on air suction, and several have been installed in large and small plants during the past year.

Abrasives

E. C. Schacht.[11] Developments in the coated abrasive industry have been almost wholly concerned with product improvement. Of major importance to the woodworking industry was the general introduction of heat-treated garnet, which has met with universal favor on account of the considerable increase in sanding production which results from the hardening and toughening of the garnet mineral.

Manufacturing processes have been carefully studied with a view to perfecting coatings and controlling the mineral-particle relationships so as to increase cutting efficiency. Factory testing against known standards indicates a much greater coating uniformity, which must be apparent in field applications. The variation from run to run is negligible compared to the tolerance of ten years ago. Another feature which must be readily noticeable and appreciated by the sandpaper user is the improved uniformity of flexibility. This is obtained by securing the optimum moisture content in the coated goods at the time of the flexing operation. Not only is the same flexibility maintained from roll to roll and from run to run, but the correct degree of flexibility is produced for the various classes of goods according to the field application—for example, very flexible spindle paper, finishing cloth, and sheet material for hand sanding and less flexible for flat-belt and drum-sander operations.

The rate of substitution of aluminum oxide for garnet in the woodworking industry did not continue in 1932 as in the immediately preceding years. As a matter of fact, the sales relation of the two products remained almost static. On account of abnormal operating conditions in the woodworking industry, no significance is attached to this relation. It will probably take several years before the ultimate correct field applications are generally made throughout the industry for both minerals.

Dust Collecting

H. A. Colvin.[12] Progress in this field has vitally affected the woodworking industry this year, by greater use of the mechanical self-clearing dust-collecting main, which allows mobility in manufacturing operations. This mobility permits taking advantage of worth-while improvements in methods and in productive machines, allows new machine layouts to meet variations in design and kind of product, avoids excessive expense on the services of supply, and makes possible any desired combinations of machines for less-than-maximum schedules, without increasing the production cost per unit of output.

Education

Thos. D. Perry.[9] A campaign for the annual crop of engineers trained to serve the woodworking industry is proceeding vigorously, with some converts at every turn. It is not an easy matter, however, to overturn or substantially modify the existing engineering curricula in a short time, especially in eras of closely prepared budgets, when every new study must be most carefully investigated.

There are three principal issues that seem to stand out as problems in the minds of the educators:

[9] United Plywood Corp., New Albany, Ind. Mem. A.S.M.E.
[10] Babcock Machinery Corp., New York, N. Y. Assoc-Mem. A.S.M.E.
[11] Behr-Manning Corporation, Troy, N. Y.
[12] Allington & Curtis Mfg. Co., Saginaw, Mich.

(1) Is a specialized woodworking-engineering education desirable or is a general fundamental engineering course better with only a few specialized subjects? In the former case, the expense of instruction is materially increased and the field for graduates correspondingly lessened. In the latter case, considerable adjustment and experience are required before full activity in the industry can be accomplished.

(2) Should there not be greater recognition of wood as a raw material? In most of the fundamental engineering courses wood and its various products have been almost ignored; yet all of the textbooks and courses of study since the establishment of the technical schools seem to have been such as to encourage metal-mindedness and not to recognize the fact that in the industrial world the value of wood used ranks close to that of iron and steel.

(3) Shall courses in woodworking-engineering education be best adapted to so-called forestry schools or so-called engineering schools? There are arguments on both sides of this question.

The chairman of the committee has been working on this subject through the faculties and deans of Purdue University, University of Michigan, Massachusetts Institute of Technology, and University of Iowa.

Investigation of Mechanical Properties of Certain Tropical Timbers

W. Kynoch.[13] All wood material for the work was collected in the respective countries of origin by persons believed to be competent to recognize the woods in the forms of standing trees. The School of Forestry and Conservation has had no control over collection.

A sample from every bolt received has been submitted to Prof. S. J. Record, of Yale University, for examination as to identity. In some cases an element of doubt as to specific identity exists, but in the majority of cases identity is believed to be well established.

All timber reached New York in the form of logs and was there sawed into test-piece blanks, under our direction, in accordance with standard procedure. All material, therefore, was received at Ann Arbor in the form of these blanks, with the exception that cativo, the first wood tested, was received as a log, and a quantity of mahogany for special hardness tests, originating in different districts in Central and South America, was received in the form of planks.

The timber in hand at Ann Arbor for testing at the present time represents 48 wood species, originating in Brazil, British Guiana, Mexico, Nicaragua, Panama, Peru, Philippine Islands, San Domingo, Venezuela, and West Africa.

The methods of testing employed are those specified in "Standard Methods of Testing Small Clear Specimens of Timber," serial designation D-143-27, of the A.S.T.M.

The work in progress on the material in hand falls under the following heads:

(1) Machining of test-piece blanks and manufacture of standard test pieces for the different kinds of tests and determinations.

(2) Mechanical tests and physical determinations as called for in D-143-27.

(3) Mathematical computations.

(4) Miscellaneous items, including monthly and (later) weekly weighings of blanks for air-dry tests and maintenance of the necessary temperature and relative-humidity conditions in the later stages of conditioning air-dry test pieces for testing.

(5) Preparation of reports.

Work on material representing five wood species has been completed. Four of these are Venezuelan woods and one is from Panama. Work on 17 additional species is well advanced, and in this group the testing of the green material of eight species has been finished. Work on several more species has been begun. During the year ending October 31, 1932, a total of a little more than 8000 tests and determinations was made.

A report on the (completed) work on cativo, the species from Panama, was prepared by W. Kynoch in April, 1930, and was transmitted at that time to the United Fruit Company, which supplied the timber representing this wood.

A progress report covering work to May 31, 1932, on the six species from Brazil was prepared by W. Kynoch last spring for transmission to the Brazilian Government through the Tropical Plant Research Foundation, Washington, D. C.

Wood Finishes

Paul S. Kennedy.[14] Development of improved and more economical wood finishes was particularly fruitful during 1932. Probably the most salient features include:

Baking Finishes on Wood. Temperatures of 150 to 160 F convert varnishes (new synthetic-resin type in particular) to harder and more durable films. Possible limitations for general employment are the type of wood unit to be finished, close control of application, and evident necessity of curtailed baking periods. A possible important extension is the use of synthetic-resin enamels on general cabinet work. Many of these bake rather quickly to a vitreous-like surface and have been adopted with success on small baked wood parts, such as brush handles etc., replacing oil- and varnish-type enamels in previous use.

Stains and Fillers. Quick drying, non-grain-raising, fast-to-light stains have made extended replacement of old-type acid and water stains, with important savings and new color effects. Fast-to-light combination stain filler came into use; it reduces cost in many instances, but is overpriced for some production. A filler was finally developed to give perfect adhesion with lacquer, but is at present too costly for most production.

Clears and Flats. Synthetic-resin-base varnishes have been greatly improved, particularly from the standpoint of chemica resistance and case hardening. A new silica pigment "flatting agent" succeeds in eliminating most of the former objections to transparent lacquer and varnish flat finishes, but is rather costly in varnish vehicles.

Outstanding Specialties. Imitation of metal finish has been most successful, particularly with wooden caskets, by the use of so-called opalescent lacquers. Imitation of ebony finish, used principally on pianos, is secured by a unique treatment of pure carbon-black pigment, which permits a black lacquer to dry naturally to a high gloss, and eliminates use of anilines, which have been so troublesome in the past.

[13] School of Forestry and Conservation, University of Michigan, Ann Arbor, Mich.

[14] Vice-President, Murphy Varnish Company, Newark, N. J. Mem. A.S.M.E.

Modern Door Construction—Manufacture and Use

By SERN MADSEN,[1] CLINTON, IOWA

Doors, along with windows, constitute the moving parts of a building, and their manufacture must provide for all contingencies that might affect their fit and wear and durability. The many kinds of doors are described, and the author details the methods of construction and the machines that are used. Advice is given for the proper handling of the door from the preparation of the lumber for the mill processes, through all the details of construction and finishing, the shipping of the completed product, and its installation in the building for which it is intended.



NO PART of a building is used more than its doors. With the windows they constitute the only moving parts. For this reason a door must be constructed most carefully. Upon it are imposed all conditions of use and abuse and the effects of dry and wet weather. Up to the middle of the last century most doors were hand made. Woodworking machines with revolving cutter heads, such as planers, molders, and tenoners, were not developed extensively until that time. These early machines as used in door manufacture still left much hand work to be done. Sanding and trimming were largely hand operations.

The doors were built with mortise-and-tenon construction and were generally held together with wooden crosspins driven through the tenons or by wedges driven into the ends of the tenons. Such construction was somewhat slow and the joints were likely to open and allow the doors to sag. Such construction also required an extra amount of lumber for the tenons. Use of glue on tenon joints was of little value because the shrink and swell of wide tenons in the deep mortises would tend to shear the glued joint and destroy its holding power.

Since mortising and tenoning were comparatively expensive, doors were constructed with as few of this type of joint as practical. This no doubt accounts for the prevailing use of the four-panel door during this period.

In these earlier doors the panels were made of solid wood and had raised surfaces in the centers. Naturally the size of panels across grain had to be limited to reduce the bad effects of shrinking or swelling. This led to the use of dividing bars or vertical mullions.

In the early nineties the modern roll-feed drum sander was developed, and it influenced the kind of door to be made. Previous to this doors were finished largely by hand. A type of door planer had been developed that had a head set at a 45-deg angle, and the door was passed under it. Such an arrangement would surface the door with little tendency to tear the grain on the cross-rails. About all such a machine could do was to level the surface. To finish the surface large sander disks were mounted on vertical shafts. The disks were covered with sand paper on the under face, and the doors were sanded by passing them under the disk, which was rotated at a suitable speed. With such methods of finishing and with the mortise-and-tenon joint, the highly hand-decorated doors of the period were a logical result.

The drum sander at once made the plainer types of door more economical to make and influenced its use where the cheaper doors were wanted.

About 1900 the doweled door construction was introduced, and with it came newer designs. The cost of making rail-to-stile joints was reduced, and the five-cross-panel door was a logical result. In this door all rails could be cut and tenoned identically, and only two widths were required. The panels could all be made identical in size and shape, and the assembly of the door was simple.

Use of plywood for panels came at about the same time and made practical the use of larger panels. This resulted in the one-panel and two-panel doors. Extended use of veneers and plywood led to development of slab or smooth-faced doors.

Thus the development of new machines and methods and the employment of new materials changed the whole business of door manufacturing.

The influence of the architect has probably had less to do with changes in the trend toward modern door manufacture than have machinery, methods, and materials.

Modern Door Details

Modern doors can of course be made of almost any desired height and width. The usual height for residences is 6 to 7 ft and the width from 2 to 3 ft. Garage doors average 8 ft wide by 7 1/2 to 8 ft in height. The thickness of present-day doors depends upon the use. Light closet doors, storm doors, etc. may be of 1 1/8 in. thick stiles and rails. Ordinary residence interior doors are 1 3/8 in. thick. Larger residences, offices, public buildings, etc., are fitted with 1 3/4 in. thick doors. Greater thicknesses are often found in public buildings, while as little as 3/4 in. is used for cupboards, cabinets, and the like.

The details of sticking or molding of doors is varied. The common types are shown in Fig. 1, the most common being O. G., ovolo, and rule joint.

Aside from the paneled types there are some newer designs that may be considered among the modern doors. The French door is little else than a swinging sash with the stiles and rails made heavier. The glass is usually held by means of wooden stops nailed in. The slab doors with smooth faces are used extensively in hotels, hospitals, etc. A plain-faced slab door

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Presented at the meeting of the Wood Industries Division of The American Society of Mechanical Engineers, Jamestown, N. Y., November 15 and 16, 1932.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

ordinarily has a thin face veneer, and is often ornamented by inlays. Some have a thicker face veneer and vertical V-shaped grooves to give the effect of heavy plank construction. Openings of various shapes and arrangements are employed for design effects. Plain or leaded glass is stopped into such openings. Another practice is to trim the tops in curved or Gothic designs.

Much of the effect or beauty of these modern doors depends upon the entrance frame. In developing pleasing combinations of door and frame the modern architect has made his contribution.

By different arrangements of panels, rails, etc., modern door-construction methods permit quantity production of doors classified by architects as French, English, Colonial, Spanish, etc. There is at present probably more thought given to the use of best modern construction and manufacturing methods combined with attractive design than at any previous period.

Woods Used in Door Construction

The kinds of wood used for door construction vary about as much as the sources of supply. In the United States the one preferred above all others is the Northern white pine, but the available supply began to dwindle about 1900. It has been replaced largely by California sugar pine and Western yellow

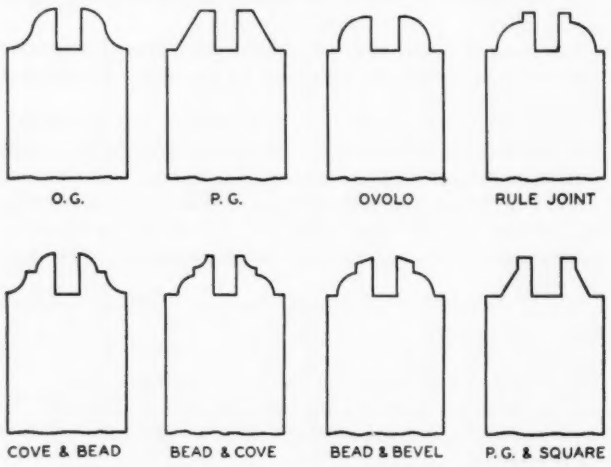


Fig. 1 Common Patterns of Door Sticking

pine (*Pinus ponderosa*). Douglas fir is used to a large extent for fir doors. A wood often used for garage doors is a species of hemlock. The Southern yellow pines and cypress are used to a limited extent. Yellow pine is more subject to "working"—shrinking and swelling and checking.

Other woods such as the oaks, birch, walnut, and mahogany, are either too expensive to use solid or are subject to "working," and their use is limited to veneers applied over built-up cores of the softer pines and firs.

The ideal wood for door construction has these qualities:

(1) Fairly straight grained and machines easily
(2) Least subject to swelling, shrinking, and warping
(3) Suited for taking stains and finishes
(4) Available in quantity at reasonable cost.

The shrinking and the swelling of wood are important factors in door manufacture. A report of the research department of the Western Pine Manufacturers Association gives information on shrinking for Ponderosa pine, most commonly used for door construction.

A moisture increase or decrease of 1 per cent will change a 1 in. width of lumber a maximum of 0.0030 in., an average of 0.0022 in., and a minimum of 0.0013 in. Ordinarily a door in a residence will have a moisture content between 6 and 16 per cent. This might cause a variation in size of $^{5}/_{16}$ in. Such a door would require $^{1}/_{8}$ to $^{5}/_{32}$ in. clearance when hung and when at about 10 per cent moisture content. The average clearance is about $^{1}/_{16}$ in. at each side.

The moisture content of lumber at the time of manufacture into doors should be about that of the door as installed. For doors of pine the general rule is 8 to 12 per cent of moisture, based on oven-dry weight. Lighter woods may contain a higher percentage of moisture than the denser woods. This is important where different woods are used in a door, such as in veneering with denser woods over softer cores or where hardwood dowels are used in pine or fir doors.

Control of moisture content of materials used in door construction is important. Quite generally, pine lumber to be used for doors is dried at the sawmills to less than 8 per cent, and it picks up moisture in transit and arrives at the door factories ready for use, with moisture content of 8 to 14 per cent. Storage in sheds before use will tend to equalize this to between 8 and 10 per cent in summer or 10 and 12 per cent in winter. These values refer to the upper Mississippi Valley.

Such lumber retains its moisture content through the factory processes in the summer months. In the winter it will dry out somewhat in the shop with artificial heat. The exhaust fans cause several changes of air per hour, the outside air entering the shop being heated and dry. Care must be taken that door stock in its machining processes is not exposed long enough to make it too dry. In a general way this dryness of the shop in winter may be used to correct the increased moisture then present in lumber. An atmosphere with as low as 20 per cent relative humidity is not uncommon in a heated woodworking shop in the winter season. Very few shops are equipped to add moisture to the air.

In an unventilated and unheated building the moisture content of pine lumber may rise as high as 16 to 17 per cent. It is therefore important that unheated storage sheds should be well ventilated in the winter. Excess moisture can be corrected by a drying out in the shop.

After doors have been built up, they should be kept in a warehouse where the temperature never goes below the dew point. In winter about 40 F is a safe low limit. If frost forms on a door in the warehouse, it will upon melting be soaked up by the doors. Repeated conditions of this nature will raise the moisture content to excess. The remedy is adequate ventilation and a small amount of heat. Sheds for storing cut stock should be treated in a like manner.

The Northern white pine of the Great Lakes region was originally the preferred wood for door and sash construction, but its growing scarcity increased the cost so that the more plentiful supplies of pines from the Pacific Coast, even with higher freight rates, were more economical. Regardless of cost, it is now the most practical wood available. Increased cost naturally led to a search for cheaper materials or methods. This caused the development of veneered stock for stiles and rails and of plywood for panels, and also largely of slab doors.

Production of Cut Stock

Cut-stock production from lumber is an intricate procedure. Enough 100 per cent clear lumber to be cut into door parts of exact size without waste is not available, and cut stock is therefore developed from various grades of lumber. A board is graded according to the quality and number of clear stiles and rails that can be cut from it by dodging knots, pitch pockets, cracks, stain, and other defects. "Standard Grading Rules" are published by the Western Pine Association.

Most door factories produce their own cut stock from lumber. There are plants devoted exclusively to its production and sale.

The dimensions of cut stock vary greatly. In thickness the stock generally allows for sanding 1/32 in. off each side. In width not less than 1/8 in. is allowed on each edge for dressing or molding. If all stock were straight this might be reduced. Some allowance is also made for removal of feed-roll marks. In length from 1/2 to 1 in. should be allowed for trimming, tenoning, etc.

Dowel stock is usually accumulated from hardwood rippings. Core blocks are developed in the course of producing cut stock. Many use lengths as short as 5 in. The value and the use of core depend largely upon the cost of lumber. As lumber prices go down, a condition is reached where the cost of producing and gluing up core and veneering it is greater than the cost of solid stock.[2]

Adhesives in Door Manufacture

Glues as used in door manufacture are limited to a few general classes, though they may vary widely within each class.[3] Animal glue, made of hides, bones, etc., is used in door construction chiefly for the doweled joints. Casein, made from curdled milk, is used in assembling cores and for veneering or plywood construction. Casein glues are probably the best that have been in use in door construction for 15 years. Vegetable glues have also been used extensively in veneering and core work. Blood-albumin glue requires a hot-plate press and is not used much in door construction. A recent development is a powder that is sprinkled on the wood parts, and when heat and pressure are applied a strong joint is produced. Still another development is a bakelite sheet that is inserted between parts to be glued. A hot-plate press in a few minutes produces a waterproof joint. Much seems to be expected from these later glue developments.

Machining Processes

The Fig. 2 shows the names and relative locations of the parts of a paneled door. This will assist in following the machining processes. The author will attempt to show only one of each type of modern machine.

Assuming that there are in the cut stock the sizes needed for a certain run of doors, the first step is to have the proper quantity counted out and started to the mill.

It is best practice to dress all cut stock such as stiles, rails, and muntins just previous to their going to the machining department. A modern type of direct-motor-driven single surfacer with flat-belt feed is ideal for this purpose. The door cut stock entering the mill divides into three routes:

1 The stiles are to be laid out, bored, and stuck
2 The rails are to be tenoned, bored, and doweled
3 The panels are to be sanded, squared, and raised.

Following this the door is assembled and "squeezed up" in a clamp. The last operations are sanding, inspection, hand work, if any, and trucking to the warehouse.

Each part of the door will be followed individually. In their trips from one operation to another the parts are usually transported on "cars," or platform trucks, about 18 in. high, 2 ft wide, and 4 ft long. Two wheels on an axle, cart fashion, at the center carry the main load. Swiveled casters at each end keep the cars from tipping up. Stakes at the four corners aid in keeping loads in place.

[2] See paper, "Modern Sash and Door Manufacturing," by Sern Madsen, presented at A.S.M.E. Wood Industries Division meeting, Chicago, Ill., Nov. 26, 1926, and printed in *Mechanical Engineering*, December, 1926, pp. 1453–1457.

[3] See "Symposium on Glues for Wood Products," A.S.M.E. Trans., 1932, papers WDI-54-1 to 7.

Stiles

The stiles enter the shop surfaced on the two sides, and the two edges are as they come from the saw. About 1/4 to 3/8 in. over width is allowed for dressing and sticking, and the length is about 1 in. more than the door it is to be used in. The stiles are practically the only parts that are laid out, and the only layout necessary is a marking of the dowel holes for the kind of door to be made. All other door parts are made to specified over-all widths and lengths.

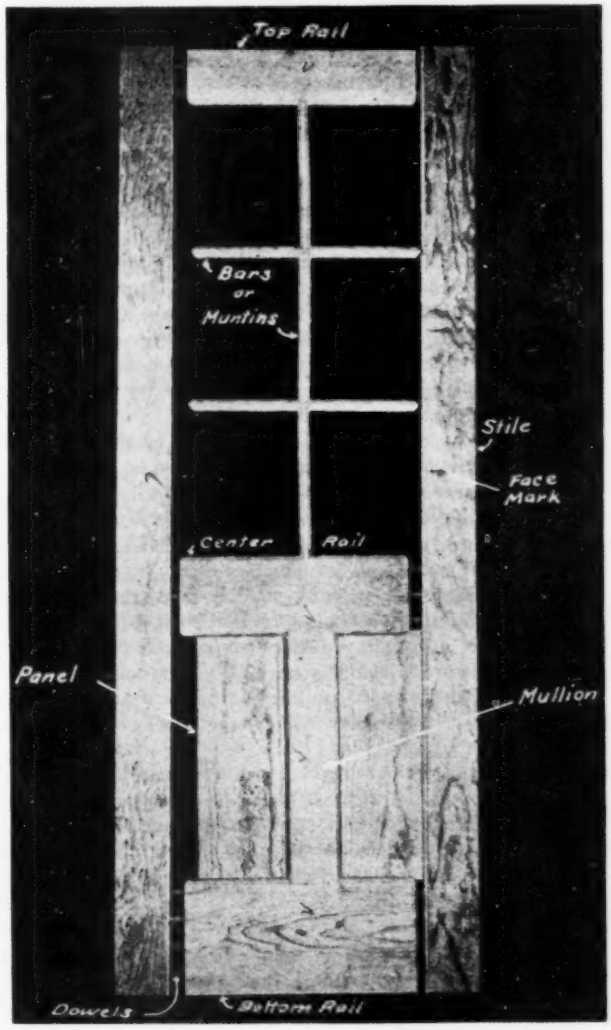


Fig. 2 Location and Names of Door Parts

The spacing of the dowel holes depends upon the number used and the panel sizes and arrangements. Even this layout may be eliminated by using rods marked off for setting up the boring machines.

There are two types of boring machines in general use. For stock doors gang boring machines with 16 or 20 bits in a row are standard equipment. For special doors or small runs it is more economical to set up and bore the holes on a four-spindle machine, where one can selectively bore with one to four bits. These bits are all permanently spaced to standard center distances such as 2 1/4 or 2 1/2 in. On the gang borers the bits are yoked in pairs. Adjoining pairs are spaced by bosses on the yokes so that spacing of all holes is facilitated.

The drills may be vertical or horizontal. The most common types bore from the under side. The drills are of the twist variety, the same as used in metal drilling. It is quite important that the lands of the drill be ground for clearance up close to the leading edge, otherwise the drills will be apt to heat excessively or break off. High-speed steel is preferred to reduce resharpening.

It is necessary to clamp the stiles rigidly between straight and powerful jaws during the boring to correct any bowing or warping. Unless this is done all the holes will not be equidistant from the face.

In doweling the doors all boring should be accurately spaced from the side called the face. Both stiles and rails are therefore face-marked as they pass through the first machines, by a hand crayon or some automatic means.

In the case of the stiles, because of this facing, one must bore rights and lefts, by having "right" or "left" holding jaws or by making one of them stationary while the other moves.

From the boring operation the stiles go through the sticker. This is really a two-head molder, which dresses one edge square and also generally chamfers the corners and molds the opposite edge. The molded edge is the one that was previously bored. The mold is usually symmetrical for paneled doors, the groove at the center being adapted to receive the panel. Glazed doors usually are "stuck," with one side of the mold cut away inwardly to the groove. A mold is later nailed in as needed. For certain doors the rabbeting or cutting away of the mold may be done more expeditiously on a shaper.

The types of door stickers vary greatly. The older machines have roll feeds and have top and bottom dresser heads. The greatest weakness in these machines is that the feeding mechanism is apt to slip and dig into the wood, if for any reason the force required to push the stock through the machine is too great. Crooked or warped stock is not adequately "ironed out," so that the sticking is not always accurately faced throughout the length of the piece. Another fault is that stock bowed edgewise is generally still bowed after sticking instead of being straightened. Besides this there is a tendency, especially on crooked stock, for the piece to kick up as the trailing end passes the feed rolls or chip breaker, with a resultant scalloping or uneven cutting at the ends.

The improved machines make use of a continuous chain feed through the length of the machine. Such machines produce exceptionally straight stock. A piece once started passes entirely through the machine without needing another to push it out. This aids greatly in setting up the machine.

To stick rights and lefts it is customary to run half the bored pieces "other end first." To do the right kind of sticking it is important to have the best types of cutter heads. Before the days of high-speed steel almost any type of head was used. The old flat-ground bit used on a square head has given way to the milled or formed bit that can be sharpened by simply grinding one flat face.

With motors direct-coupled to the cutter-head arbors, less vibration of the machine permits higher cutter-head speeds and faster feeds. With higher speeds and less vibration, the jointing of the bits has become best practice. It is always difficult to properly joint complicated contours on bits. This has been simplified by making the center or grooving bits retractable, so that they may be drawn inwardly out of the way during jointing and subsequently set outward in place again.

Self-centering sleeves or a pair of centering cones help to set up the heads accurately on the arbor. Proper grinding and fitting machinery is necessary to do a good job of setting up the heads. The ideal method would be to set up a head completely sharpened and jointed, then place it on the machine for immediate operation. Unfortunately this idea is difficult of achievement. There are many small difficulties yet to be corrected. Unless jointed on the machine modern heads are generally limited to about 80 ft per min feed at about 4500 rpm. Higher speeds have been and are being used.

Rails

The rail cut stock passes directly to the tenoning machines. Single-end machines are used for small runs and special set-up work. Double-end machines are used for stock runs.

In these machines the stock is sawed to exact lengths and coped to fit the molding or sticking on the stiles. These cuts are also made square with a reference edge, which later is the controlling or facing edge in the sticker. The sticking will be parallel with this edge and therefore square with the ends.

As with the stiles, it is also necessary to mark one side of the rails as a face side. These face marks on stiles and rails must be on the same face on the door when it is assembled. If it were not for this precaution, it would be difficult to make a door that would be flat and level.

To describe a tenoning machine is a long story and has already been done very well in a paper by Mr. Mansfield.[4]

The function of the tenoner in door construction is to shape or form the ends of the rails to fit the sticking and the groove on the stiles. Heads for this purpose may have bits milled to exact pattern, but more often they are ground from high-speed steel of rectangular section. The exact patterns are best determined and made by use of accurate steel templets. Almost without exception a set-up piece is run and tried on the molding of the stile. It is highly important that a tight joint must be provided.

All door-sticking patterns in common use have a flat shoulder or quirk at the outside about $1/8$ to $3/16$ in. wide. This shoulder is important in getting a perfect face fit, since it is at right angles to the surface. A surface joint of this nature if found at an angle other than 90 deg is very difficult to fit up tightly. The tenoned ends of the rail bear slightly harder against these shoulders at each side than at any place between them. This is to insure a tight "squeeze up" at the surface joint when the door is assembled.

The rails now pass to the sticker, which may be the same machine that sticks the stiles. The rigid side of the run for the stock is called the fence side. It is usually the inside or frame side. The rail must be run through the machine with its face-marked side against the fence. Also the edge that was engaged by the lugs on the tenoner chains should run upon the rigid bed surface. In this way a parallel cut is removed, and the edge of the rail remains square with the tenoned ends. Any irregularities are cut out by the opposing or top head.

If the cutter head is correct to profile, then the important part of the set-up is to see that the cut is centered or faced up correctly from the face and that the stock is being cut to correct over-all width. Accurate width is more important on the rails than on the stiles, since they must assemble in the door with proper allowance for panels.

After sticking, the rails are bored for dowels. Small runs or special lots are bored on the four-bit machine used for stiles.

Stock runs are put through a rather complicated special machine known as a "three in one" machine. This machine is provided with a hopper for the rails and a pair of reciprocating bars or chains to advance the stock one piece at a time successively past three stations. At each station the stock stops and is clamped down securely while being operated upon by the machine simultaneously at each end of the three rails.

At the first station properly spaced drill bits advance inwardly and bore the holes for the dowels. From three to five

[4] "Double-End Tenoners and Their Use," by J. H. Mansfield. A.S.M.E. Trans., paper WDI-54-11.

bits are provided, the number of bits used for boring depending upon the width of the rail. The face side of the rail must be down against the solid bed of the machine.

At the second station glue is squirted into the bored holes by two sets of nozzles which enter the holes while a pump forces glue through them.

At the third station a suitable mechanism drives as many dowels as needed outwardly from the bottom of two dowel hoppers.

Thus there are six operations in progress at once, and each in turn is multiplied by the number of dowels. When there are four dowels being driven, there are 24 operations in progress at one time. About 20 rails are doweled complete per minute.

Instead of the machine mentioned, one might use one or more of the improved single-end boring, gluing, and doweling machines.

The rails in passing through the last three machines are held down flat with all twist or warp squeezed out so that the operations will face up correctly. If this were not done, the parts would not assemble in a level surface.

Panels

The panels may be obtained either from plywood or from solid stock. Plywood needs no other work done upon it than having the sides sanded so that they are smooth and the thickness is proper to fit the grooves, and having the sides and edges trimmed and squared to proper size.

The usual practice is to run the plywood panels through a double-end saw for trimming. A better way is to run them through a machine that also chamfers the edges so that the panels will enter the grooves of the stiles and rails easily.

Solid panel stock is first dressed on a planer and then sanded both sides on a drum sander. An endless-bed sander is ideal for this purpose.

If a single-end panel raiser is used, the panel stock must be accurately squared and sized first. One edge at a time is raised on this machine. Raising a panel consists of actually cutting away stock on the surfaces around the edges so as to leave a raised surface effect in the center.

From the step-off at the border of the raise the panel tapers down to near the edge, where the sides again become parallel for a distance at least as wide as the grooves in the rails and stiles are deep.

A double-end machine for this purpose cuts two sides at a time. Panels are first run through so that the ends are cut, and then a second time, when the sides are cut. This operation may also be done on a shaping machine. This is the machine that was formerly used exclusively for this job.

Dowels as Made for Doors

The dowels used in doors may be made of oak, maple, birch, or other hardwoods. Yellow pine and gum are sometimes used. They may be purchased from plants that specialize on dowels, turned items, etc. The larger door factories generally accumulate rippings of hardwood, and these are worked into dowel stock. Generally speaking, it is wasteful to cut stock lumber specifically for making dowels.

In doors $^3/_4$ in. thick, $^3/_8$-in. dowels work best; $1^1/_8$-in. and $1^3/_8$-in. doors may have $^1/_2$-in. dowels, and $1^3/_4$-in. and $2^1/_4$ in. doors may use $^5/_8$-in. dowels. In some plants the $^1/_2$-in. size is used for doors up to $1^3/_4$ in. thick. In others a $^5/_8$-in. dowel is used on $1^3/_8$-in. and thicker doors. Where softer woods are used, they should be of larger diameter.

They are usually 5 to $5^1/_2$ in. long, and the ends should be tapered or chamfered for easy entrance.

The stock for making dowels should all be well dried; at least it should be as dry as or drier than the door stock. For $^1/_2$-in. dowels the stock is ripped about $^5/_8$ in. by $^5/_8$ in., or in general about $^1/_8$ in. larger than the dowel size.

This stock is run through any one of several types of dowel-rod machines. In these machines the stock passes through the hollow shaft of a revolving head. Stock is kept from turning by in-and-out feed rolls, while knives on the head cut off the corners and produce a smooth, round rod. A spiral groove may be cut on the rod during this same operation. Feeds run from 50 to 75 ft per min.

Cutting and pointing of the dowels is done on a turret type of machine where the rods are set in vertically and are passed by cut-off and chamfering saws. Such a machine may cut and point over 1000 dowels per minute.

Dowels with necks cut near the ends are sometimes used. A spiral or a straight groove on the dowel assists greatly in allowing air to escape when the dowel is driven. It is not unusual to split a stile or rail in driving a tight-fitting dowel unless the entrapped air and glue have a means of escaping.

To provide dowels with notches and the like for the glue to obtain a better hold is of doubtful value, but it is important to have a place into which excess glue may flow.

A properly glued dowel should break under tension before the glue joint should break. Failure in a test on dowels shows that the softer wood around the dowel generally fails first.

A dowel must not be too tight or it will split the door parts. A dowel's holding power may be attributed in some measure to the inward bending of the end grain of the wood forming the dowel hole. When the dowel is pulled outwardly, it is gripped by a sort of toggle hold. This grip can of course be exerted only on fairly tight dowels, and it may even be annulled entirely by the shrinking of a dowel or by the glue setting the fibers around the dowel.

The advantages of doweled over mortise-and-tenon construction are many, some of them being as follows:

1 An amount of clear lumber the length of the tenons is saved in the length of each rail that is used.

2 The machining and assembling operations are simpler and faster.

3 The holding power of dowels is greater per unit of glue area. In an oblong mortise one cannot count upon permanent glue joint along the sides of the tenon and mortise. Shrink and swell are quite apt to break it lose. A dowel hole is round and small. The shrink and swell of parts is generally so little that the adjoining wood will yield enough to prevent rupture of glue joints.

4 Several dowels are less apt to split a stile or rail than a long continuous mortise because there is a tie or bridge between the holes.

Assembling

The assembling of the door may be performed by locating the stiles, rails, mullions and panels, etc. in relative positions while the door stands on edge or lies flat upon the clamping machine. If the door is assembled outside the clamp, rubber-headed mallets are used to drive the parts so that the dowels of the rails are started into the holes of the stiles. When assembled in the clamp, the mallet often is not needed.

Glue must be applied to the dowels before assembly of the door. This may be done by dipping the ends of the dowels that project from the rails, into which they have already been inserted.

A faster way and one used almost exclusively is to squirt glue into the holes in the stiles. A special gang glue squirter is used for this. Properly spaced nozzles enter the holes, and pumps force in the glue at the right instant and in the proper amount.

If one follows the method of laying the door parts in relative

positions in the door clamp, the process will be readily understood. One stile is laid against one jaw, and the dowels at one end of the rails are started into the proper holes. The panels are entered in the proper grooves, and the last stile is put in place so that closing of the clamp will squeeze the whole door together. Some clamps operate at an angle, others are level.

A squaring device is used along one jaw. This consists of an inwardly spaced bar that contacts the door and a series of rollers between it and the main jaw of the clamp. This permits the side of the door to move longitudinally along the jaw so that when the door comes solidly together the squared ends of the rails will be parallel and tightly against the stiles.

A certain amount of dexterity is required in operating one of these clamps. If all joints do not enter readily, the dowels or panels must be assisted by crowding or pushing. A mallet of proper size or the hands of the operator are used. It is no small trick to learn to keep one's fingers out of places where they may be pinched or even crushed.

The operator controls the closing of the clamp with a foot pedal. He must learn to work with caution, and be ready to stop instantly. Care must be exercised to stop in time to keep from crushing the door. The capacity of a door clamp is dependent upon the character of the door being assembled and the skill of the operator. It may range from 20 to 80 doors per hour.

A clamp driven by screws at each end of the jaws seems to give most satisfactory results. Air clamps are not so easily controlled. If one end squeezes up more easily, the jaws at that end will move up more readily. Parallel movement of the jaws is essential, as is also slow but positive movement. It is good practice to have the jaws set slightly closer at the end where there is the greatest resistance—namely, the bottom end of the door where the rail is wider.

In assembling doors of complicated design and where there are a large number of parts such as sash doors, a certain amount of experience is required to make the parts come up properly. End-clamping is often necessary in addition to side-clamping.

After assembly and clamping the doors are stood aside until the glue on the dowels can take its initial set. Hot animal glue is generally used on the dowels. It seems to work best with the machinery available. The dowels should fit tightly enough in the holes so that they will not readily come out even if not glued.

Sanding Operation and Repairing Defects

Sanding of the doors may be done within an hour after they are assembled. Drum sanders are used for this purpose. Machines with four drums sanding each side are growing in favor over the three-drum types. Roll-feed machines are used almost exclusively. Double-deck machines are growing in favor where capacity is an item. Such machines may consist of two machines set in tandem or a single machine having six or eight drums built to sand both sides at one pass.

Drum speeds formerly were about 1300 to 1400 rpm. Since motor drive direct on the drums is available, the speeds have increased to 1750 rpm. The rates of feed will range from about 15 ft per min to 25 ft per min. Doors should be run through at a slight angle for best results.

The grades of coarseness of paper will vary with the kind of wood and the evenness with which the parts fit. If the joints are not level, more stock has to be removed and coarser paper is required. No. 2 grit is about as coarse as should ever be used. If too coarse paper is used on the first drum, it produces scratches too deep to be removed by finer paper on the following drums. Paper may vary from No. 2 on the first drum to No. 1/2 or No. 0 on the last drum. Experience alone will tell what is the best combination for different woods, character of work, and rate of feed.

The doors should now be completed and ready to start on the way to the user. All doors after sanding are carefully inspected. Occasional small defects can be repaired on the bench. Such defects may be loose panels, an open-face joint, or a wood defect that previously was hidden. By skilful "slivering" minor defects can be repaired satisfactorily. Sometimes the door must be torn apart and a new stile or rail put in.

It is one of the functions of the assemblers to guard against use of any defective parts, but certain defects may show up after the door is sanded.

Other benchwork consists of putting in glass or nailing in moldings, stops, etc. depending upon the kind of door.

The process just described covers ordinary doors built up of solid materials. Veneered doors are put through exactly the same process just as though the parts were solid.

Building Up Veneered Stock

It will be necessary to deal now with the details and methods of building up veneered stiles and slab doors. The operation of manufacturing plywood will be omitted, since this is an industry in itself, and many door manufacturers buy plywood panel stock cut to size, sanded, and ready to use in the assembly of doors.

Stiles and rails of dimensions identical to solid are built up of core, edge strips, and face veneer.

In the process of developing cut stock, many short and narrow blocks are produced. Much of this is free of knots and fairly straight-grained. This is commonly known as core stock. When lumber is high in price, it is economically profitable to assemble these blocks with edge strips and to veneer them for use as solid stiles, but the real occasion for the built-up stiles and rails is where the face of the door is to be oak, birch, mahogany, walnut, or similar woods.

Many of these woods are too expensive or are otherwise impossible of manufacture except by the veneering process.

Edge strips are accumulated from rips of 4/4-in. lumber. They must be of a length equal to the stiles or rails in which they are to be used. One exception is that a joint may be made in the edge strip of a stile if it is so placed in the finished door that it is concealed by one of the rails. Generally, they run full length.

The processes of building up core are as varied in number as the shops that produce it. The core blocks must be ripped to uniform width. In some shops this is done on accurate straight-line saws and the sawed edges are glued directly as they come from the saw.

In other places the blocks after ripping are dressed accurately to thickness and then glued with their flat faces. Still other plants form tongue-and-groove joints on the blocks and edge strips.

A properly fitting flat-faced glue joint is as strong as one with tongue and groove. The real reason for use of tongue and groove is that it helps place the blocks in even-face alignment when they are assembled.

In some instances the blocks are machined and run into bins and subsequently used as needed. One difficulty with this method is that drying of blocks after they are in the bins may make them of uneven thickness before assembly. A better method is to dry all the blocks and dress them to thickness just before assembly. They may well be dried to as little as 4 or 5 per cent moisture content, because the application of glue will raise the moisture several per cent.

The usual method of assembling the core and edge strips is on some sort of pan, caul, or table and clamping them up edgewise to be held till the glue sets.

The time needed for setting depends on the kind of glue used.

It may be of the casein, vegetable, or animal variety. Casein and starch glues seem to have been preferred in recent years.

When the core and edge strips are removed from the clamps, they must be surfaced to receive the face veneer. The wood of the core and the strips will be swollen next to the glued joints, and if not dried before being run through the planer it will dry out afterward and leave a depressed surface along the glue lines. For this reason the assembled core should be carefully dried before running it through a planer. Perhaps some of the newer glues may eventually change this.

Face veneer may be produced by sawing thin pieces from the logs or cants or it may be sliced off with knives. It also may be rotary cut by shaving it from around the log.

For the sawed veneer there is little choice as to which side should be outward. Sliced or rotary-cut veneer should have the same side out when glued as was out before it was cut. The reason is obvious, because in the slicing the veneer will be ruptured slightly due to bending. Such ruptured side should be next to the glue.

For door construction $^1/_8$-in.-thick veneer is generally used on stiles and rails. Thicker veneers have been proved by tests to be more durable in resisting exposure to weather. This is contrary to the common belief that thinner veneer will come and go with less danger of rupturing a glue joint. For slab-door construction, where the veneer is to be grooved as for plank effect, a thickness of $^1/_4$ to $^3/_8$ in. is needed.

Application of veneer to the core is generally performed by running the core through a glue spreader that applies glue to both surfaces. The core is then laid between the two face veneers like the meat in a sandwich. Successive assemblies of this kind are stacked up from three to six in height. Stiffener plates or cauls are inserted at intervals until the stacks of core and veneer are built up to the height capacity of the press. Generally this assembly is completed outside the press, and the unit is then rolled into the press, with suitable plank cauls and steel retainer beams above and below the stock to be veneered.

The entire stock is then compressed with a unit pressure of about 100 lb per sq in. of glue surface. Retaining clamps are applied, and the unit may be removed and set aside for the glue to dry.

Hydraulic presses are common for veneering purposes. There are many who favor screw-driven presses. A hydraulic press without an accumulator operates more slowly. A leaky pump also may make it uncertain as to exact pressure at the time the retaining clamps are applied. A screw press will maintain its pressure during clamping or overnight if necessary. Both types are in common use. For larger installations the hydraulic press with an accumulator may operate more rapidly. A paper by Mr. Francis on glue presses should be read in this connection.[5]

After removal from the retaining clamps the veneered stiles should be piled on stickers and run through a drying box to remove excess moisture added in the veneering operation.

The ends can now be trimmed and chamfered and any overhanging veneer be trimmed off, and the stiles and rails are then ready for making into a door in exactly the same manner as a solid piece of wood.

Slab doors are built up by much the same processes as for making veneered stiles and rails. Core and edge strips are built up in the same way. In fact, identical stile and rail core before veneering may be run through the sticker and tenoner.

The usual practice is to cut a groove on the inner edge of each stile and to form a tongue on the rails at the ends. Dowel holes are bored and a complete door is built up with rails solid one against another the full length of the stiles.

[5] "Hydraulic Presses for Plywood," by Harry G. Francis, Trans. A.S.M.E., 1931, paper WDI-53-12.

This assembly, without any veneer, is squeezed up in the regular door clamp. In this way a built-up core is obtained for the slab door that has exceptional strength and is fairly free from shrinking and swelling. Since planers are not generally available for this size, this core is usually run through a sanding machine to level all the joints to prepare it for the veneer application.

Cross-banding is applied on this core. It consists of $^1/_{16}$-in. veneer run crosswise of the door. Over this cross-banding the face veneer is applied at right angles or lengthwise of the door. It may be from $^1/_8$ to $^3/_8$ in. thick.

The remainder of the operation is to place this door under the press, and when the glue is set and dried, run it through the sander and trim the edges.

In applying face veneers of widths needed on a slab door, if of more than one piece, the edges are taped or glued to hold the joints securely.

Subsequent operations of grooving and cutting out and installing of glass are largely bench operations. For these operations portable power saws, routers, and sanders are used extensively.

Special Designs of Doors

The methods described cover the most usual types of doors. There are doors of special designs that need to be handled in exceedingly special ways.

A door has been marketed recently where the outside appearance is that of a number of planks with a groove at the joints between boards. The facing boards are cut on the back side with dadoes having angling sides to form a dovetail joint with rails which have a coacting rib threaded through them. The side stiles are then applied. They are doweled to the ends of the rails, and on the front they resemble in effect one of the facing boards. Such doors are not level-faced on the back and are used for garages, etc. To build these doors some very special machinery is required.

Mention has been made of the one-panel doors with an inserted frame surrounding the panel. The usual method of making these frames is to miter the corners and assemble them with clamp nails driven into saw kerfs. This is a much faster method and generally more satisfactory than the older form of splined or tongue-and-groove joint.

Another innovation is in the design of the lower edge of panels on exterior doors. In the usual construction the panel is slid into a groove in the rails. This groove aids water to run down the panel and get into the very heart of the door, where it may loosen glued joints and dowels and may even cause rotting unless the door is always kept properly painted.

The new joint involves a groove in the lower end of the panel which straddles a rib on the top of the rail, so that water is effectively directed outwardly.

For hand-trimming of doors there are a number of small machines available. In cutting doorstops, foot-power miter-cutting machines are used extensively. Other forms of miter cutters are provided with power-driven saws and generally produce more accurate and smoother work.

For cutting out openings, especially in slab doors, such as needed for lights of glass, portable motor-driven saws guided on suitable tracks clamped to the doors have proved to be great time savers. These saws can be set in place on the door and the saw let down to cut through the door and then moved forward on the track as desired.

These saws when fitted with an angle-faced grooving saw are very handy for cutting the vertical grooves of the modern and fashionable slab door.

For rounding the ends of circle-top and Gothic-top doors a modern router can be used to advantage. A form is placed on the

door, and the router bit cuts the door as the router is guided around the form.

Another machine that proves itself handy is the small portable belt sander which is run over the door like a flatiron. It is very useful and efficient in cleaning up special doors, repaired spots, or for touching up spots missed by the regular door sander.

A small power-driven hand jointer is convenient for use in trimming and repair of doors.

An operation commonly known as "slivering" is frequently resorted to when surface defects show up on doors after sanding. Rather than de-grade the doors, repairs can be made to overcome the bad effects of surface checks, pitch pockets, etc.

The tool used consists of a small mill-type file, ground at an angle and sharpened at one end. This tool is run with the grain along the crack or defect and forms a V-shaped slit. Into this is driven a sliver to which glue has been applied. The sliver is wedge-shaped about like the cross-section of a jackknife blade. Slivers are ripped by a saw from clear straight-grained lumber. Pieces are cut the length desired and fitted and driven into the slit. After the glue has set, the sliver is cut off even with the surface and sanded smooth. Such a repair is permanent and is impossible of detection if properly done and if the grain and color of the wood are matched.

In glazing doors the glass is ordinarily held in place by stops mitered at the corners and nailed in with brads. Such doors for exterior use should have the glass bedded in putty.

Grading rules for doors are quite general in their scope. Such a list of rules was prepared by the National Door Manufacturers Association and the Wholesale Sash and Door Association in 1930.

Handling to Avoid Damage

The handling of doors needs to be done with care to prevent damage. There are a few precautions that will help. Chamfering the edges of the stiles helps to prevent slivering if they are bumped. As a rule doors should be lifted or slid carefully on the edges upon a smooth floor or surface as the faces are easily marred.

In sliding a door on its ends it is very easy to split off part of the stiles. Chamfering the ends of the stiles is desirable. On veneered doors it is a necessary precaution. The stiles at both top and bottom usually are allowed to project about $^1/_2$ in. as a protection. These horns are to be cut off when the door is hung.

Slab doors should have a cleat or strip nailed on the ends and extending the full width of the door. If this is not done, veneer is apt to be damaged or torn off.

In storing doors in a warehouse there are three ways of arranging them. They may be stood on end, sloping slightly to lean against a post or wall. Care should be taken to see that about equal numbers of doors lean against opposite sides of a post, otherwise considerable side push is created that may cause a building to collapse. Where large quantities of doors are stored, they may be piled flatwise on top of each other. Strips should be laid down on the floor under each pile. Such piles should be only of a height for which the floor is safe. Where a large variety of doors of many sizes and kinds are stored, it is probably most convenient to build suitable racks, with either adjustable or permanent cross-members, on which to lay the doors. Space for 10 doors of a kind between the cross-members is a convenient size.

In all of these methods of storing it is of utmost importance to keep the top or exposed doors covered. This can be done very well by using old doors or fiberboard covers made for the purpose. This will keep the doors clean.

It is also important to keep excess light and especially direct sunshine from the doors, as this will discolor the doors quickly. Then, too, it is obvious that there should be a tight roof overhead.

Another good practice, when new doors are added to the stacks, is that they should be placed in new piles or underneath the older ones of the same size and kind. This will keep the doors moving and will prevent an ultimate collection of old and shop-worn stock.

The suggestions regarding storage of cut stock apply to doors in warehouses also, and much more so. Proper door-storage rooms should be as dustproof as possible. Floors should be sprinkled or, better yet, covered with wet sawdust before sweeping. There should be good ventilation, and the temperature should be controlled against extremes. It should not go below about 40 F in the winter time. In some storage houses the space just below the roof becomes abnormally hot in the summer time, and this is not a good door-storage place.

While humidity should be as constant as possible, this is not generally easily controlled. It is therefore important to provide ventilation and to guard against extremes in temperature.

It has been explained how low temperatures tend to increase the moisture content of wood. High temperatures will tend to produce excessive dryness. Anything that causes extremes in moisture content is harmful to doors. Aside from injuring the joints, it also may weaken the glue used in the door.

Molds and rot are not apt to be present in a warehouse that is kept dry. It requires from 15 to 20 per cent moisture content in wood to support fungus growth. Such extremes would occur only under a leaky roof or where doors are piled on dirt or on damp concrete floors.

Another enemy that needs watching is insects. The termite, sometimes called the white ant, is becoming an increasing pest in the more southern districts and is said to be working northward. By keeping door piles up from the floor they will generally be safe. These "ants" build a dried mud tunnel or tube from the ground up to the wood in which they bore. They must have this tube leading to the ground to obtain a proper water supply. They may bore up through the wood blocks or strips on which the doors are piled.

Powder post beetles are also a pest. There seems to be much to learn about them. It is not uncommon after a door has been hung and finished to find a pile of fine sawdust on the floor and small round holes in the finished faces of the doors. The holes may vary from $^1/_{16}$ to $^1/_8$ in., and they look as though bored by a high-speed drill. It is claimed by some that these beetles develop from eggs laid in the core of plywood and in veneered doors. It also is said that they originate in hardwood cores such as oak or chestnut. Attempts to prevent trouble along this line consist of putting a germicide in the glue, which should kill them as they attempt to eat their way out. The high temperatures of the re-driers used in plywood factories are also counted upon to kill the eggs or larvae of this bug.

In preparing doors for shipment much care is required to protect them from damage by the none-too-gentle truck drivers and freight handlers.

Crating of doors requires covering of both faces with thin lumber or sheets of fiberboard. Where crating is used, the only place where nails can be driven is at the edges, preferably at the top and bottom, and then only in the ends of the stiles.

An improved method is to use prepared fiberboard cartons that completely surround the doors and are held in place by tightly drawn steel taping. By this method no nails are driven into the doors.

In loading cars care must be taken to remove all nails from the car that might deface the door. Strips of board are laid down on which to pile the doors. Sheets of paper are used generously to cover the piles of doors. A recent practice is to line the inside of the box car with paper. Special paper covers are now made up for the entire under side of the car roof. It is highly important

to keep the cinders and dirt incident to freight traffic away from the doors in transit.

Another important thing is to pack the car full from end to end or to put in bracing or bulkheads, so that the doors cannot shift within the car. The usual method employed in freight-train switching or at the "hump" in the yards is responsible for many damage claims on shipments of doors.

The use of hooks for handling crates or bundles of doors is prohibited, and yet one may sympathize with the man that has to handle a bundle of doors as they are unwieldy.

The precautions concerning storage and handling apply from the time the door leaves the factory until it arrives at the place where it is to be installed. It is highly important that the same diligent care be exercised all the way to the house. The final truck driver should not attempt to deliver doors in bad weather without properly protecting them.

Precautions in Installing Doors

The builder or mechanic on the job has his responsibilities too. Doors should not be delivered until the house is completely dried out after plastering and is properly closed up to keep out the weather.

Installing a door is a well-known practice, yet there are a number of things that are often overlooked. The doors must be trimmed top and bottom, and the sides will need dressing for proper clearances. As already pointed out, it is possible for a door having two 5-in. stiles to shrink and swell a total of $^{5}/_{8}$ in. No such amount needs to be allowed for in practice. It may be assumed that a door will be hung when with about 10 per cent moisture and that this may vary about 5 per cent either way. This would mean that about $^{5}/_{32}$-in. clearance would care for most conditions of swelling. If the door frame is still swollen when the door is hung, this may be reduced. It is also best practice to bevel the edge of the door opposite the hinges to give swinging clearance for the outside edge of the stile.

The door should be "sighted" to determine if it has a bowed edge. If possible, such edge should be placed at the hinge side, and three hinges should be used. If the lock side has a bow in it, the hollow side should close toward the doorstop. All these precautions will help to hold the door in proper shape.

Clearance to swing over the floor should always be such that the door will not rub or strike.

A door should be properly protected against weather by proper finishing. If paint is used, nothing is better than an undercoating of good flake aluminum paint. This to be covered by paint to give the proper finish and color. The tops and bottoms of the door should be painted. In fact, it may be more important to protect the top and bottom edges than any other part of the door.

The use of stain only in finishing a door may bring dissatisfaction. Stains do not protect the wood surface from moisture changes and surface checking.

The use of stain filler and varnish is good practice and is to be recommended.

Recently there have appeared dozens of new paints, dopes, preservatives, and what-nots that are claimed to make any surface waterproof. Misunderstanding may be cleared up if one will distinguish between water in the form of liquid and water in the form of vapor. In samples of wood treated with these dopes the water runs off or stands in beads on the surface. But if these same samples are placed alongside an untreated control block and are exposed to dry and to humid air conditions, most of them will absorb or give off moisture as readily as the untreated sample. There may be a slight lag, but after a very few days generally all the samples are found to become dry or wet just as though no treatment was applied.

Experiments indicate that absorption of moisture by a door is most effectively prevented by two coats of properly applied flake aluminum paint carried by suitable vehicle, two or three coats of good paint, or three coats of good varnish. The value of the newer materials as an agent to prevent moisture change and reduce shrink and swell of wood, especially doors, needs to be better established. Some of these materials may have other desirable properties such as durability, preservation, quality of finish, etc.

The home owner also has his responsibilities. He can guard against the various forms of physical abuse. A sticking or binding door should be properly refitted. The door should be kept properly painted or finished. Doors exposed to weather and sun are necessary evils in modern architectural designs which are considered good. A front door without any sort of protection such as a canopy, stoop, or porch is in for a hard time, and will need frequent help from the painter.

Weather-stripping of doors has increased in vogue. It is unwise to say what kind of strip is most to be preferred. Surely a type that crowds against the door or opposes its closing should be avoided. It will prevent the door from latching properly and may even tend to warp the door. Types pressing against the edges of the door are most in favor. Felt or rubberized strips are effective, although they may not look so well.

Use of double doors in winter is always preferable. Weather-stripping a single door does not attain quite as effective results as with the double door.

Humidity in the home, too much or too little, is injurious to woodwork in general. Humidity may be as low as 15 or 20 per cent in some homes. This will shrink the woodwork and open up cracks. With the arrival of summer, and 50 per cent relative humidity, the cracks will tend to close again, but leave broken, dark-looking, ugly joints. Such conditions are common and doors suffer as a result.

In houses kept at 40 to 50 per cent relative humdity in cold weather, frost will form on single doors and windows. Storm doors and storm sash are the best remedy. Frosted doors are sure to suffer from swelling.

The house owner has his problem to know how wet or how dry to keep his house, but most of them are much too dry in the winter; 35 to 40 per cent is a good range.

The wrong type or kind of door is often used. For inside purposes almost any kind of good construction will be satisfactory. Veneered doors built up on core have a tendency to stay straighter and are preferred for this reason.

Exterior doors probably cause the door maker more grief than any other. They are subject to heat and cold, sun and rain, and many forms of abuse. All the causes are present to effect shrink and swell, warp and twist, rot and decay, paint and glue failure, and loosened joints. A door in such a place should be given a fair chance for its life. The modern door is the best ever made, but poor judgment is often exercised in its use.

The subject of doors is an extensive one, but many of the phases of discussion properly belong under other headings. Testing of doors, strength, durability, shrink and swell, fire resistance, etc., are subjects for the laboratory and not properly a part of the manufacture. The door has been followed from the sawmill to the finished home, but in a necessarily limited manner. It is hoped that enough of the ground has been covered to stimulate further study on this century-old industry. As we progress, there will most assuredly be changes that will still further improve the modern door.

The woods used in modern door construction are listed by the U. S. Department of Agriculture in bulletin No. 158, which gives the average comparative properties of the clear wood of species grown in the United States.

BIBLIOGRAPHY

Catalog of sash, doors, blinds, and moldings, 1891, Curtis Brothers & Company, Clinton, Iowa. This shows four-panel door with raised panels; section of construction at stile and panel joint; five-panel door with mold nailed on; ornamental glass door; door with raised ornaments; fancy door when "gingerbread" decoration was in style.

Catalog of doors, glazed sash, and blinds, 1903, Huttig Manufacturing Company, Muscatine, Iowa. This shows five-cross-panel plain-face door with raised panels; two-panel door using plywood for panels; one-panel door with plywood for panels; a fashionable front door of the early 1900's.

Catalog of architectural woodwork, 1932, Curtis Companies, Inc., Clinton, Iowa. This shows a modern front door with possibly the limit of present-day decoration; an English design with many panels; circle top or groove door; modern glass door; favorite outside door; two-paneled door with veneered stiles and plywood panels; English design of raised-panel door; one-panel door with molded insert frame around panel; pair of French sash doors; smooth-faced slab door.

[The illustrations of the doors included in these three catalogs, and referred to in the paper, are halftones and therefore incapable of clear reproduction. This limitation to reproduction also applies to the illustrations of the various machines described in the paper. It is assumed that those interested in the paper know the principles and appearance of these woodworking machines generally.]

Discussion

F. L. CROOKS.[6] We wish to take exception to statements as to the superiority of the dowel door over the mortise-and-tenon door. The author referred to hand-made doors in buildings of the Revolutionary days, made mortise and tenon. Could a better argument be used for the merits of the mortise-and-tenon door than these early American doors?

The author states that the dowel door is far superior to the mortise-and-tenon door. We are manufacturers of both doors, and feel that the dowel door has its place. The principal argument is that it can be manufactured at less cost than the mortise-and-tenon door, and in certain types it is just as good a door, will serve the purpose just as well, and will last just as long as if made mortise and tenon.

Competition demanded that we install dowel-door machinery to reduce our costs of manufacture, and when this dowel equipment was installed it was naturally supposed that we would have little use for mortise-and-tenon machinery. But we found that when we made a one-panel dowel door with 5-in. stiles and top rail and the door was molded with a $1^1/_4$-in. flush or raised mold, in a great many cases when springing the molding in to get tight miters, it forced the door apart at the top rail. On examination, we found that the dowels would be properly glued, as the fibers of the wood in the core would still be perfectly tight to the dowel, but had been fractured, allowing the dowel to pull out of the end of the rail. We have also had the same experience with two-panel dowel doors, especially where they were molded with a heavy flush mold, and in bradding in the mold, blows from the hammer on the nails would drive off the stile, which apparently had been weakened when the molding was sprung in place. This necessitated putting clamps over the door at the rails to stop this breakage while doors were being molded.

After several experiences of this kind with the dowel construction, and never having encountered this trouble with the mortise and tenon, the management was convinced that it was a mistake to make all types of doors of dowel construction. Our experience teaches that certain types of doors should be made mortise and tenon.

The one-panel door is finished with a flush mold and not solid mold, as we would not care to take a chance of making a large lot of these doors running up to 3 ft 6 in. wide, and have to depend on two dowels in the top rail carrying the weight of this door over the years. The constant slamming of the door would have a tendency to weaken the dowel construction, and we would expect to be called on to replace or repair doors made this way.

The six-panel door is another design which we feel should be made mortise and tenon. On doors of this type, the mullions are mortised into the rail with a tenon $1^1/_2$ in. long. If this door was made dowel construction, it would have to be a slip tenon $^1/_2$ in. long. If this type of door is made dowel construction, very accurate workmanship is necessary, or there will be a strain on the dowels in the top rail, and when assembling the door, in a great many cases the dowels are bent and break off when door is being assembled. Mullions mortised and tenoned in the rail make a very much stronger and more durable door than if put in with a $^1/_2$-in. slip tenon. If the door is mortised and tenoned, it can be driven together and clamped at both ends and sideways, and tight joints are assured.

The same conditions as described for the six-panel door apply to the ten- and fifteen-light doors. It is practically impossible to make this door of dowel construction and not bend the dowels in the top rail. This, in a great many cases, causes loose bars and mullions. If this door is made mortise and tenon, it allows the top and bottom rails to be drawn tight on the mullions, insuring much more rigid bars than if the door is made dowel construction.

Where the mortise-and-tenon door is more practical than the dowel door is on special doors, accordion doors as large as 3 ft 6 in. wide and 20 ft high, 3 in. thick, special paneled doors paneled differently each side, paneled doors 10 or 15 panels flush molding one side, one flush panel the other side, circle head, special small paneled two-side doors with one diamond light in center, solid doors of different designs and different kinds of wood from $1^3/_4$ to 4 in. thick, and many other types of special-design doors which are practically impossible to make of dowel construction and have them stand up over the years.

CHAIRMAN ARTHUR D. SMITH, JR. (president, Geo. W. Smith Woodworking Company, Philadelphia, Pa.): Our experience has been that mortise-and-tenon door construction is better in special-door manufacture, where the door is being built to architect's details. Unquestionably the dowel door is just as good and cheaper, and therefore more desirable for the stock doors that are made in Mr. Madsen's plant. I also hold with Mr. Crooks that the mortise-and-tenon door is the more practical one for a special-order proposition. We recently had a job in Philadelphia for 61 doors on one floor of a building, every door being different. It was a difficult job using the mortise-and-tenon construction, and it would have been even more so with the dowel construction.

Mr. Crooks said that where the door is a molded door rather than solid sticking, it is the practice to spring that molding in and set it in one corner, and then bend it and set it down in the other corner. He said that he had known where springing that molding in has caused an expansion that broke the glue joint of the doweled door. If a door was properly made, I cannot conceive of breaking the door apart by springing the molding.

P. J. SCHLOMER (Wilkin-Challoner Company, Oshkosh, Wis.): Mr. Madsen's paper was mostly on stock doors. Mr. Crooks, on the other hand, turns out special doors mainly, and I believe his statements are also correct. So far as the small molding having a tendency to break apart the glue joints, I do not believe that, because when the glue is squirted in, there is always sufficient glue left on the door, and that has a tendency to make a tight joint. So far as open joints developing, I do not believe that is the result of the boring of the holes in the doors. That

[6] President, W. D. Crooks & Sons, Williamsport, Pa.

would be more likely to come in the tenoning operation where the drill does not cut square.

V. J. Reed (Mattison Greenlee Company): If the machine operation is accurate enough so that no flaw can be found from the machine operating the bore, there is very little difference between the dowel and the tenon.

Chairman: Do you find that the dowel construction is more prevalent?

Mr. Reed: Yes; that is true in the West, but not in the East. It is a little easier to machine on the tenon than on the dowel. The tenoner can be set up very readily; and in the boring, if the machine is not accurately built, you are going to expand that part of the hole that is out of line. I would not say that my preference would be for a mortise-and-tenon on an order that had very few of a kind or size. Sometimes it would not be profitable to set it up for a few doors.

Mr. Kurkjian: Sometimes the Eastern designers of furniture for a hotel or an apartment use their own ideas of what is effective and strong, and will put in mortise-and-tenon joints; and I know of one manufacturer who took the order on the furniture and had it written in the order that they would not be responsible for the mortise-and-tenon joints holding after the fixtures were in. Their reason was the extreme dryness of rooms in a hotel and the moisture in the air from opened windows, so that this sudden variation affected the mortise-and-tenon joint. The dowel would have been guaranteed. In the Grand Rapids district the dowel construction of joints of furniture is general. On the other hand, in the Carroll district, I believe the mortise-and-tenon is used almost exclusively. An Eastern architect specifies mortise-and-tenon joints because most of those men are brought up under the old-school influence, and they still are of the opinion that the man who made the door by hand with a mortise and tenon, and had the tools to do that, was right. Now we have special machines to bore straight and hold the stock down. If the matter of the strength is brought in, I believe those doweled doors are certainly just as strong as any mortise-and-tenon doors were.

Chairman: The author states that it would be of interest and service if each type of door could be tested to destruction, to prove which is the most serviceable. In the East the building code provides for fireproof construction in certain buildings. The code will permit wooden doors as being fireproof, provided that the door will pass certain tests. I have witnessed one of these tests, and the door that withstood that test had mortise-and-tenon construction. At the same time, doors were tested in the same furnace with dowel construction, and some of them passed. It is my recollection that the average of the mortise-and tenon door was much higher than that of the dowel door.

C. Lindbeck (United Lumber and Supply Company, Jamestown, N. Y.): I think a mortise-and-tenon is ahead of the dowel construction, as it stands up better. In a building it does not make much difference, but in an ordinary test the mortise-and-tenon door will stand more. It depends on how they were put together at first.

William H. Gunlocke (Wayland, N. Y.): After many years of furniture manufacturing, particularly chair manufacturing, we have come to know something about the dowel construction of chairs. In most instances it is superior to mortise-and-tenon. I cannot picture a door constructed with the dowel permitting of the molding pushing the door or spreading the stiles. There was something wrong in that instance; either the dowels did not fit and were not glued or the man who was cutting the molding did not understand his business. We do not use a doweling machine, because we have no faith in such a machine. There are too many failures in a machine that glues and drives dowels. If you are boring a hole in a piece of wood, side or end grain, and take a glue stick and glue that hole and drive that dowel, and then with a pair of pliers extract that dowel, you will be surprised to learn what a small percentage of film or coverage you have on that dowel. Few people appreciate that a $^1/_2$-in. dowel has $1^1/_2$ in. of width that must be covered if it is going to hold. The dowel is longitudinal grain; you have $1^1/_2$ in. there; there is the same amount of area in the end wood. That is 3 in. of wood that must be covered with glue to give this a joint to do a good job. We have a large hot-air box, and when the dowels come in, we never think of taking the moisture-content reading. There are screens in this hot box, and those dowels are put on trays of this mesh wire. Over the front of the box is a hanging canvas.

When we drive those dowels in, they are absolutely bone dry—so dry that the moisture will expand the dowel after it is driven. As to the mullions in a mullion door not fitting, that is only poor work. No one calling himself a mechanic would do that kind of work. If the holes were bored as they should be and properly framed, this would not happen. As to doors taking on moisture, if the door is finished, there is not much chance of its taking on much. One of the worst problems in the furniture industry is the manufacture of a wood-seat chair, and it is almost incredible that wood-seat chairs 18 or 20 in. wide will expand $^1/_4$ in.; that they just pushed the back rails and slats apart. One of the things that we are most particular about is the coating of the seats around the outer side. You can do this better with a brush.

C. B. Norris (Haskellite Manufacturing Company, Grand Rapids, Mich.): The matter of the fire test interested me. We are making fireproof panels that might be used in door construction. These fireproof panels do not have any fireproof wood in them. However, they would stand up well in the test described. We have tried them with bunsen burners, and they compare very well with panels made up with fireproof lumber. They are made in the ordinary way, except that underneath the cross-banding we have glued a layer of asbestos millboard that has been impregnated with phenol formaldehyde. This will give the board strength so that it will not spread afterward. We have had rather good success in these panels, because after all no wood is fireproof.

Author's Closure

The discussion on the manufacture of doors centers upon the relative merits of dowel and tenon construction. When we consider special or odd doors we sometimes find constructions where only mortise-and-tenon joints are practical because of limitations in space, etc. In making odd doors, a few of a kind, which are identical to stock doors except for size, it is just as easy to machine the parts and bore them accurately for dowel construction as for mortise and tenon.

A modern door factory, using proper methods, gages, and equipped with proper and adequate machinery, has no difficulty in machining parts properly and boring holes accurately spaced for the dowels. Dowel construction can be made just as readily and economically as mortise-and-tenon construction when applied to the stock type of doors.

To call it an advantage of tenon construction that the tenon may be cut under-width to permit side movement to absorb inaccuracies is questionable. It looks more like toleration of inaccurate work.

To question the possibility of boring doweled holes accurately enough is to question the soundness of the method by which possibly 95 per cent of all stock doors are manufactured in the United States.

As to gluing dowels, some tests have proved that squirting glue in the holes is more satisfactory than dipping the dowels

before driving. Putting glue in the hole applies the glue to the walls. Glue applied to the dowel is largely scraped off the dowel by the square shoulder edges of the holes. A slightly pointed dowel tends to ride over glue on the walls of the hole. A combination of the two would be better yet. Perhaps counterboring the holes on a bevel would improve the holding of the dowels, but this is impractical where the dowels enter on a surface that is molded other than flat.

The real answer is to make some tests on the two constructions and determine the relative holding power of each method.

WDI-55-3

Laminating With Phenolic Resins

Some Practical and Theoretical Considerations Involved

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Phenolic resins as wood adhesives have qualities that have been long known, but commercial application has been retarded because of the cost. The method of manufacturing plywood is said to be revolutionary in the manner of applying the adhesive. The paper describes four ways in which these resins may be applied as an adhesive between two or more plies of wood or other material, these being as a varnish, as a film, as a dry powder, and as a colloidal solution. The author concludes that, despite objections raised against the high temperatures employed in laminating with resins, they constitute an ideal adhesive for thin panels.

PHENOLIC resins, so called, are synthetic products derived from the condensation and polymerization of phenol (carbolic acid) and formaldehyde or, in special cases, other aldehydes. The resulting resinous products may vary in their physical characteristics over wide limits, depending upon the proportions of phenol and formaldehyde as well as upon the procedure employed and the type and amount of catalyst used. The present commercial development and application of phenolic resins started with the inventions and discoveries of Dr. Baekeland about 1909, though the reaction of formaldehyde and phenol had been known for many years previously. In several of his early patents Dr. Baekeland disclosed that phenolic resins could be used as adhesives for wood. However, until comparatively recently, there had been no commercial developments along this line, though patents had been taken out in this country and abroad from time to time covering specific processes involving phenolic resins as wood adhesives. Commercial development has been primarily retarded, at least in the plywood field, by a cost consideration, and it is still the cost item that prevents their widespread use today.

There are several ways in which resins may be applied as an adhesive between two or more plies of wood or other material, some of which involve features that are patented or have patents pending on them. They are:

(1) *As a varnish.* The resin being soluble in alcohol, acetone, and other similar solvents, it may be made into a varnish and applied to the plies by brushing, dipping, or rolling. When this method is employed, it is next necessary to pass the coated plies through a kiln or oven to drive off the volatile solvent prior to hot pressing.

(2) *As a film.* In place of applying the varnish directly to the surface of one or more of the plies, a thin sheet of paper is impregnated with the varnish, the solvent is driven off, and the resulting dry film used as the adhesive by placing a sheet of it between each pair of plies and hot-pressing the resulting assembly.

(3) *As a dry powder.* The dry resin is ground into a fine powder which will pass at least an 80-mesh screen, and this dry powder is applied by special equipment to the surface of the plies, which are then assembled and hot-pressed.

(4) *As a colloidal solution.* The resin is not soluble in water, but it is possible to suspend it in water with the aid of a protective colloid, thus securing a stable solution that can be applied the same as the varnish. After the application of this colloid, it is necessary, just as with the varnish, to dry out most of the water prior to hot-pressing.

Because there is no apparent commercial future to method (1), on account of its involving the loss of relatively expensive solvents, it will not be discussed in detail. The other three methods will be taken up in detail later.

While there are several differences and variations in the various laminating processes involving phenolic resins, due to the different methods of application of the resin, they have characteristics in common, many of which vary from conventional glue methods.

In the first place, phenolic resins, in their reactive state, are thermo-plastic. By thermo-plastic is meant that they become plastic when heat alone is applied, without the necessity of a liquid being present. The usual glues are not, or are at least only partially, thermo-plastic. Animal glue and casein will melt or become plastic when heated in the presence of a very small amount of water, while with reactive phenolic resins no liquid is necessary to secure plasticity. Therefore, this characteristic theoretically permits a perfect dry-gluing process.

Second, phenolic resins are thermo-setting. They set or polymerize with the addition of heat, in contrast with the usual setting of glue by the evaporation of water or the setting of cement by combination of the water chemically with the cement itself. The setting or polymerization of the resin is somewhat similar to the setting or coagulation of a blood glue.

Since phenolic resins are both thermo-plastic and thermo-setting, they must be used with heat and pressure to make the bond, which means that a hot-plate press must be a part of the equipment in laminating veneer with resin. It is obvious that in laminating sheets of veneer that are not perfectly flat and that are rigidly deformed or buckled, pressure must be applied to bring them into contact. However, in making a bond with phenolic resins, pressure is also necessary to develop the proper bonding qualities of the resin. When, by the application of heat, a phenolic resin polymerizes and changes from its reactive state to its insoluble and infusible state, a multitude of small blisters is formed. In other words, the resin seems to bubble or boil. To prevent this and to secure a perfect bonding film between the plies, it is necessary to apply a combination of heat and pressure. Consequently, the hot plate press must apply both uniformly.

It is evident that these resins constitute an adhesive which, in its primary form, is dry and can remain dry on the surface of the assembled but unpressed plies. Moreover, they will become plastic and flow without the addition of any liquid, simply by the

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

application of heat and pressure, and they will then change into an insoluble and infusible form, making a complete and permanent bond between the plies. Thus we have an adhesive that theoretically is 100 per cent dry and is not dependent upon the presence of any liquid. This, obviously, is a big step forward in the adhesive art, since the swelling of the wood upon the application of wet adhesives, combined with the consequent shrinkage when the excess liquid is driven off, accounts for many of the problems and failures encountered in conventional gluing processes.

But beyond this, the resin, when polymerized or "set" by the heat and pressure to form the bond, is not only water-resistant, but is totally insoluble in water, and if the polymerization is complete, is insoluble in most other solvents, except concentrated alkalis. Also, it is infusible and is not further changed by heat until it begins to break down and char at temperatures in excess of 500 F. It is not susceptible to attack by fungi and bacteria. In fact, it is surprisingly inert.

There are problems of application, but before discussing them, let us examine the necessary requirements of pressure, heat, temperature, and time.

Pressure

When using phenolic resins, there are no requirements concerning the application of pressure that are not always present in the lamination of wood plies by any process. Since heat must be applied along with the pressure, some new factors are introduced, although they are common to all hot-press operations. The principal one is that, since there is only a thin body of material to be pressed between each pair of plates, there is very little cushion to take up press deflection, whereas in conventional cold pressing, the thick bundle of material will in a measure compensate for press deflection. Any hot press, in order to produce even passable results, must have rigidity far beyond that required in a cold press working on a bundle of stock. Therefore, the cost of the bare press must necessarily be higher than the cold press used with cold wet glues. It is not practical to convert a typical cold press into a hot press by the simple addition of some hot plates.

Since the pressing of any glue that is practically dry at the time of pressing involves no "squeeze out" problem, maximum pressures permitted by the wood may be employed, and since better bonds and more homogeneous panels result from the use of maximum pressures, it is usual to employ pressures from 150 to 300 lb per sq in. when laminating with resins. Due to the relative inertness of resins applied as film, it is necessary to employ higher pressure with films than with resin applied in other forms, other conditions being similar.

Heat

This thermo-plastic material requires a definite time-temperature cycle to produce polymerization, and therefore a definite heat input requirement, since it is necessary to maintain a minimum temperature for a definite time. There are two factors that determine the total Btu input required; one is the type of resin and extent of polymerization at the moment of heat and pressure application, and the other is the mass of the panel. Thickness of ply—that is, the distance of the glue line from the hot plate—affects the time of heat application, but the total mass of the panel being pressed also influences the total heat input, since it is obvious that the greater the mass, the greater the number of Btu required to develop a given temperature at the glue line. For example, if three plies of $^1/_{20}$th veneer are subjected to a pressure of 200 lb per sq in. between two plates having a temperature of 320 F, a bond can be secured in 50 sec with the proper resin applied as a dry powder, but if the $^1/_{20}$th center ply is replaced by a lumber core $^3/_4$ in. thick and the same two $^1/_{20}$th-face plies are applied to it, 120 sec will be required under the same pressure and temperature to effect a bond, since the mass of the core absorbs the heat and prevents development of the required temperature at the glue line until the temperature of the mass of the core has reached the point at which further transfer of heat practically ceases. The fact that it is necessary, in a short time, to heat up a lumber core to a minimum temperature of 250 F by pressure contact with hot plates, in order to secure the necessary temperature conditions at the glue line, constitutes one factor wherein phenolic resins fall short of the ideal adhesive. This rapid heating of a mass of substantially dry wood, like a lumber core, to a relatively high temperature, and holding it at the temperature long enough to cure even the most advanced resin, introduces stresses that cannot be other than detrimental. High temperatures may be safely applied to wood surfaces by the application of hot plates as long as evaporation is taking place. Time of contact is not a disturbing factor until evaporation ceases, but at that point rapid deterioration of the wood begins.

Temperature

Phenolic resins, in the reactive state, are not completely stable. Slow polymerization is constantly taking place. But in considering such resins as adhesives, this factor does not particularly enter. Temperatures at which plasticization and polymerization take place in producing plywood are what count. It is obvious that temperatures must be employed at the glue line that will, in combination with the pressure, be high enough to cause the resin to become plastic. However, it has been found that the melting point of a given resin must be materially exceeded if the proper rate of polymerization is secured to produce a good bond. The minimum temperature which will result in the necessary rate of cure or polymerization depends upon the particular resin used. The minimum temperatures for phenolic resins, in general, are given by a number of authorities on the subject at 290 F. The author has found, from testing a large number of phenolic resins for wood-bonding properties, that glue-line temperatures below 300 F result in material loss of strength. These tests include resins applied as varnishes, films, dry powders, and as colloidal solutions. On the other hand, some recommend plate temperatures as low as 266 F, which would result in even lower glue-line temperatures, as will be shown later. Since apparently the employment of increased pressures will permit of the use of slightly decreased temperatures, this difference of opinion as to permissible minimum temperatures may be due, in some measure, to differences in the amount of pressure used. Temperatures up to 350 F apparently have no detrimental effect upon the curing of the resin. At any of these temperatures the critical factor in the problem is, after all, the wood itself, rather than the resin, and the critical point with respect to the wood is complicated by the time element and the moisture content of the wood. The essential point to be brought out is that the use of phenolic resins as adhesives necessitates the application to the wood of temperatures at least above 260 F. Thus it is evident that such resins again fall short of the ideal adhesive. While it is possible, under certain conditions, to employ such temperatures for the required length of time, without the slightest detrimental effect on the wood structure, the ideal adhesive should not require temperatures above 212 F, thus eliminating all chance of blister from the formation of steam in the interior of the assembly.

Time

Time enters the problem from several angles, and it in turn is influenced by nearly every other factor. The time required to complete the bond has a vital bearing on the total cost of laminating. It affects the cost of the equipment, since the press

capacity required for a given daily production is in direct proportion to the time required in the press. It affects the labor cost, the floor space required, the inventory, and the carrying charges. It determines the service that it is posisble to render the customer. Duration of temperature and pressure is by far a greater factor in the effect produced on the wood than either temperature or pressure.

The time required to make a complete bond with phenolic resins is influenced and determined by numerous other factors, which will be covered later. With veneers containing 5 per cent or less of moisture, a plate temperature of 320 F, and a pressure of 200 lb per sq in., panels of three equal plies can be completely bonded with a suitable phenolic resin applied as a dry powder in the following times: $^{3}/_{64}$ in., $^{1}/_{2}$ min; $^{3}/_{20}$ in. 1 min; $^{3}/_{16}$ in., $1^{1}/_{2}$ min; $^{3}/_{8}$ in., 3 min.

When a similar resin is applied as a colloidal solution, these times are increased approximately 50 per cent, even if sufficient water has been removed prior to the pressing operation to bring the coated surfaces to a proper dry condition.

Published data on the time required with a certain resin film glue are as follows: "Thickness of the outer veneers up to $^{1}/_{16}$ in., gluing time 6 to 8 min; thickness of from $^{1}/_{16}$ to $^{5}/_{64}$ in., 10 min. Thickness of from $^{5}/_{64}$ to $^{5}/_{32}$ in., 12 to 15 min."

These figures convey a general idea of the speed possible with phenolic-resin adhesives applied to thin panels. The most obvious factor affecting the time required to secure a given result in any hot-pressing operation is the thickness of material interposed between the hot-plate surface and the glue line; that is, the thickness of the outer plies. To determine the effect of the thickness of the plies, it is necessary to determine the rate of heat transfer through wood under the particular conditions prevailing at the glue line. Fig. 1 shows this rate of heat transfer for $^{1}/_{64}$-in. and $^{1}/_{16}$-in. dry birch. Fig. 2 shows the rate of heat transfer through $^{1}/_{8}$-in. aluminum cauls. These tests were all made with veneers containing less than 5 per cent moisture. While the rate of heat transfer is influenced materially by the moisture content of the wood, it will not be considered here, since veneers containing over 10 per cent of moisture cannot be used successfully with resin in any form.

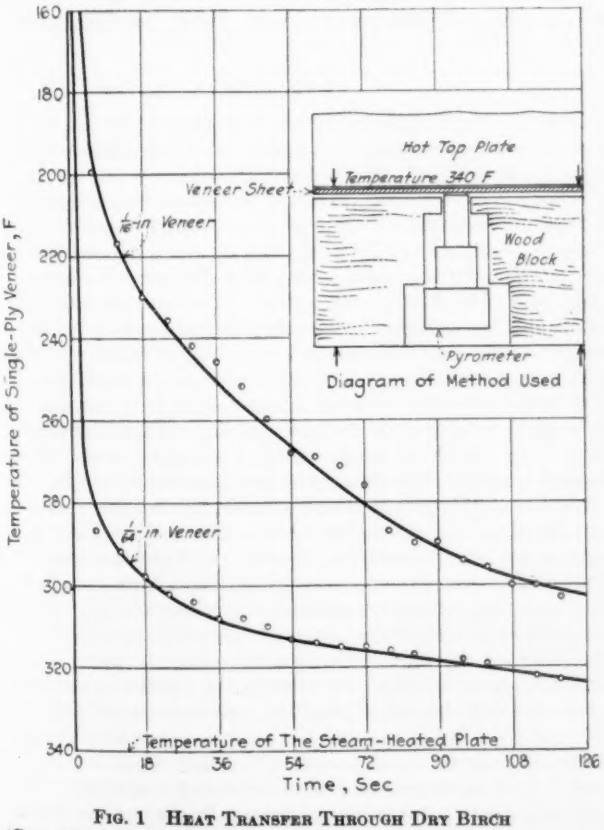


FIG. 1 HEAT TRANSFER THROUGH DRY BIRCH

(Curve showing rate of heat transfer from a steam-heated platen through a single-ply veneer sheet when held in contact by a pressure of 150 lb per sq in.)

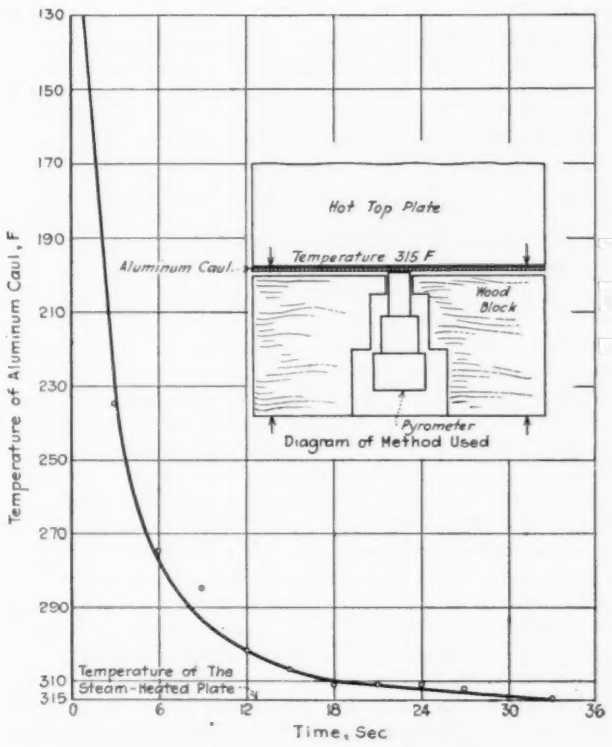


FIG. 2 HEAT TRANSFER THROUGH ALUMINUM CAUL

(Curve showing rate of heat transfer from a steam-heated platen through a $^{1}/_{8}$-in. thick rolled aluminum caul when held in contact by a pressure of 150 lb per sq in.)

Another important factor affecting the press time required to secure a complete bond is the character and condition of the resin at the time heat and pressure are applied. First, there is the fundamental fact that the time-temperature curve of all phenolic resins is not the same. Second, due to the method of applying the resin or to the influence of other prior operations, the extent to which the resin may have been partially cured or polymerized will vary over wide limits. Since the aim in any hot-gluing process is to achieve a minimum time of application of heat and pressure, the ideal resin is the one that has the fastest rate of cure. This is a problem of the resin manufacturer who secures rapidity of cure by the addition of various accelerators, just as in the rubber industry speed of vulcanization is increased by the addition of chemical accelerators. After the particular resin to be used has been selected, it must be determined how far the cure may be safely carried during its manufacture and still allow sufficient range, not only in the resin itself, but in its application as an adhesive. Allowance must be made for possible further cure during shipment and storage, and during closing of the press where heat is present and may reach the glue line before the application of the pressure. The latter is a very vital consideration, especially in a multiple-opening press where, with manual loading, there may exist a considerable time element between the insertion of the first assembled panel between a pair of hot plates and the moment of application of full pressure on

the completely loaded press. If due to contact of the panel assembly with the hot plates or to other causes the cure or polymerization of the resin progresses too far before the full pressure is applied, no bond will result. While there is no disturbing "assembly time" factor entering into the laminating of plywood with resin, there is this important factor of time of press loading and closing, even if protective measures in the form of metal or wood cauls are used. The whole matter of premature cure of the resin between the plies in the hot press may be best understood by comparing it with the making of steam in a boiler. If cold water is put into the boiler, it is necessary to apply heat for a considerable time in order to bring the water to boiling temperature, but if water at, say, 200 F is put into the boiler, very little heat and very little time are required to develop boiling temperatures. In the same way if resin, with a low cure, is applied to the veneer, it is necessary to apply considerable heat for a relatively long time to bring the resin up to the point at which final and complete polymerization takes place, but if the resin, when it is applied to the plies, has been "advanced" nearly to its "end point," very little heat and time are required to reach this "end point." Of course, the degree of advancement of a resin will in some measure influence its cost, just as hot water is worth more to a steam plant than cold water. Due, however, to the action of heat and pressure on the veneer and the cost of hot-pressing equipment, it is usually much cheaper in the long run to pay a premium for a highly reactive resin that is sufficiently advanced to give minimum pressing times.

Time required in the press is also vitally influenced by temperature. If we apply to the plies hot plates that have a temperature of 320 to 330 F, not only will the transfer of the heat through the plies be faster, but the cure of the resin will also be faster than if plates of a temperature of 260 to 270 F are employed.

Time of cure of a resin is also influenced by the amount of free liquid present. A given amount of resin in solution or in suspension will not, under identical conditions, completely polymerize in the same time as will the same amount of dry resin. Therefore, the greater the moisture content of the assembled panel prior to pressing, the slower the cure, and, conversely, the longer the time required under heat and pressure. It is immaterial whether this moisture content of the assembled plies be due to moisture carried in the veneers themselves or whether it is water or other liquid added to the assembly during the process of applying the resin. Due to this factor and also to the fact that the total moisture content of the assembly must be low, in order that blisters will not result from the hot-pressing operation, any method of application of the resin in solution or suspension must be followed by a drying operation to remove the liquid vehicle. Such a drying operation must be under careful temperature and time control to prevent excessive curing of the resin during this operation. If the cure is carried too far in this drying operation, no bond will result, largely because there will be no stage during the hot-pressing at which the resin will be in a plastic state. It is obvious of course that any method of applying the resin in a liquid vehicle introduces the same old problems of swelling and shrinking the wood present in all wet glue processes, irrespective of whether the bulk of the liquid vehicle is removed prior to the hot-pressing operation. It must be borne in mind that the introduction of any appreciable amount of water or other liquid that swells the cells of the wood is a serious detriment, and only introduces problems that otherwise might be avoided.

The amount of moisture present in the unpressed assembly at the time it is inserted in the press not only affects the time of cure of the resin in the hot press, but it also has a vital influence in other ways. At the temperatures and pressures employed any moisture present will be transformed into steam, and if this steam does not entirely escape from the panel being pressed, an internal pressure will be developed in the panel. This internal pressure is counteracted by the external pressure exerted by the press, but as soon as the external pressure is removed, upon opening the press, the internal pressure must relieve itself, and blisters form between the plies. In extreme cases the wood itself is ruptured. The formation of such blisters can be combated by reducing the moisture present, by providing means for it to escape or by reducing the temperature of the pressed panel below 212 F before the removal of the external pressure. The latter method is a remedy for a bad condition, while the elimination of excess moisture removes the cause. A heating and cooling cycle with the usual solid drilled steel hot plates is a slow and costly process, due to the large mass of metal involved. However, new types of hot plates have been developed which permit a heating and cooling cycle in less than 1 min, with a minimum requirement of steam and cold water. However, their use with resin adhesives does not completely give immunity to the use of high-moisture-content veneer. This arises from the fact that, while it is perfectly possible to completely cure a phenolic resin in the presence of considerable moisture, the resin so cured does not make a good bond with the wood plies. For example, if three sheets of $^1/_{10}$th veneer, containing 20 per cent moisture, are assembled with resin in any form and hot-pressed, and then cooled before the external pressure is removed, in order to avoid the formation of blisters, it will be found that the bond is only superficial. Apparently there is a definite limitation on the allowable moisture that may be present when laminating with phenolic resins. There is also some evidence to show that the maximum allowable moisture will vary with different species of wood, with gum requiring a lower moisture content than several other species.

Four methods of applying resins have been enumerated, each method having definite characteristics due to the method employed.

Film

In the film method, certain problems incident to the use of the varnish or colloidal method are assumed by the film maker. The laminater is furnished with a uniform product, in which the resin has been cured to a definite point, and in which there is a constant and definite resin content. In using a resin film, the spread of the adhesive is uniform and it is constant. Under some conditions, the fact that the laminater cannot vary the amount of spread might be a distinct disadvantage, since it is quite generally admitted that the minimum amount of adhesive to secure a maximum bond will vary with the thickness of the plies and with the different species of wood. It is reported that the resin content of one well-known resin glue film is such that 11 lb of actual resin are applied on each 1000 sq ft of glue line. This corresponds very closely with the average spread of actual dry resin, when the resin is applied either as a dry powder or as a colloidal solution.

While phenolic resins are thermo-plastic, it is necessary, in the manufacture of a resin film glue, to use a high-melting-point resin and one that is not too reactive, in order to minimize the tendency of the film to react and become sticky in the roll when stored and used under the extreme temperature conditions that are met in the Southern parts of this country. Therefore, to aid in the plasticizing of the resin in the film, advantage is taken of the small amount of moisture contained in the veneers to be laminated, and it is important that this moisture be sufficient to effect plasticization of the resin. Also, it is important that the amount of moisture is not excessive, for otherwise no bond will result. This requirement of a rather definite minimum and maximum limit to the moisture present in the veneers to be laminated with the film places this method in the realm of an exact technical process. The description of one such film defines

these moisture limits as follows: "(1) Cores should not exceed 5 to 6 per cent moisture content. (2) Crossbanding veneers, outer veneers, and fancy veneers require a moisture content of approximately 6 to 10 per cent. This is the moisture content that is found in veneers which have been stored for some time in workshops with normal temperature and moisture content, i.e., a room temperature of 70 F and atmospheric moisture of 60 to 70 per cent."

Since, in using a film, it is necessary that a definite amount of moisture be present, it is impossible to progressively press long panels in a short press, because the wood adjacent to the press during the first pressing dries out to such an extent for a short distance beyond the press that no bond will result over this distance when the next pressing is made. Therefore it is necessary to use a hot press having a platen area equal to the maximum size panel that it is desired to make.

In using a film-resin glue it is claimed there is no difficulty in making five-ply or seven-ply panels in a single pressing, though naturally the time in the press must be increased to compensate for the total thickness of plies lying outside of the inner glue lines.

Powder

When the resin is applied in the form of a dry powder, the laminater is furnished with a uniform product having a definite degree of cure, which may be well toward the "end point" of the resin, and a very reactive resin may be employed, due to the fact that the resin can be maintained in powdered form even under severe temperature conditions by the addition of special agents. Due to the high reactivity and advanced degree of precure of the resins applied as dry powders, extremely short times in the press will produce the complete bond. However, it is desirable with these resins to further speed up their cure by employing somewhat higher temperatures than are recommended for use with the film. Due again to the high reactivity of the resins applied in powdered form, the presence of a liquid is not required to secure the required plasticity under the action of the heat and pressure; consequently, there is no minimum requirement as to the moisture content of the veneers. Thus it is possible to progressively press long panels in a short press. This is an important factor, since it permits the laminater to purchase a press of the size of his average length panel and still be able, by progressive pressings, to make the small percentage of panels longer than his average.

While it is possible to insert a dry glue film between plies of veneer and have it maintain its position in relation to the plies during the various handling operations incident to assembly and loading of the press, this is not true of a dry powder. Therefore when the resin is applied in this form, it is necessary to provide some means to hold the powder in the same even layer in which it is deposited. The usual means of accomplishing this is to spray on the surface of the ply a minute amount of liquid, so that the dry powdered resin is applied to a slightly moistened surface. If a heavy spread of the dry resin is applied, it is also of advantage to spray a similar amount of the liquid on the deposited layer of the powdered resin. The preferred procedure involves the use of a liquid having a boiling temperature above the maximum temperature reached in the hot press and one that will combine chemically with the resin itself, so that no vapor is liberated to develop internal pressure and the consequent blisters from the action of the heat and pressure in the press.

Another factor arising from the use of a highly reactive resin in dry powdered form is that the character of the bond is influenced by the rate of cure. This makes it extremely difficult to make five-ply and seven-ply panels in a single pressing, unless the plies are $^1/_{16}$ in. or less in thickness.

Colloid

As previously pointed out, the application of any adhesive in a wet state is open to adverse criticism. While the problems arising from pressing a wet adhesive can be eliminated by drying out the liquid used as a vehicle prior to pressing, in the case of a resin adhesive this operation must be carefully controlled. However, when a resin has been applied as a colloidal solution and the drying operation has been properly conducted, the resulting film of dry resin on the surface of the coated plies is uniform and the resin itself may be in a highly reactive state and develop complete bonds in a relatively short time in the press. The recommended temperatures correspond with those with the powdered resin. It is also possible to progressively press with the resin applied in this manner.

In conclusion, it would appear that despite any objections that might be raised against the high temperatures employed in laminating with resins, they, on the whole, constitute an ideal adhesive for thin panels. Whether they or any other hot-press adhesive employing the same amount of relatively high temperature input be successfully and economically employed with very thick plies is still an open question.

Discussion

Charles B. Norris.[2] We have been doing some work lately on phenolic resins in connection with the design of a large hydraulic press for the mass production of panels 10 ft square and about $^3/_4$ in. thick. We also have adapted this method of laminating to the hot-plate presses with which our plant is equipped. We have been chiefly concerned with the resins as colloids dispersed in water. We have done some work with the varnishes, but very little with the powder. We have concentrated upon the colloid because it seemed to be the most easily handled of the three. The results of our experimental work seem to justify our choice. The author indicates that he has found it very difficult to use the colloid. In fact, he gives quite a list of rather serious obstacles in the way of its successful use. We are using it successfully in our plant at the present time.

The author says that gluing with the colloid "introduces the same old problems of swelling and shrinking of wood." Let us see of what these problems consist. In a section through a three-ply panel at one of its edges, the grain of the two faces runs parallel to the plane of the paper. The grain of the core is perpendicular to the plane of the paper. If the panel has taken on moisture since it was glued, the core would have swollen and bulged out beyond the edge of the panel. The faces have not swollen in proportion because wood swells very little with the grain due to increase of moisture content. The core grain was of course straight at the time the panel was glued. A compression stress is built up in the core due to the swelling of the wood. The swelling of the core is restrained by the faces, and hence the faces are in tension. The compression in the core is transferred by shear in the glue line, to the faces where it balances the tension. The shear stress is local to the edges of the panel. It is very high at the edge and decreases rapidly toward the interior of the panel. It is directly proportional to the compression in the core of the panel due to the swelling or shrinking, other factors being kept constant. There will be a certain moisture content of the panel at which the stresses within the panel are zero. This moisture content will be that which obtained in the panel at the moment the various plies were bonded together. Variations from this moisture content will give rise to the stresses described. Computation of the shear stresses involved is rather difficult,

[2] Mechanical Engineer, Haskelite Mfg. Corp., Chicago, Ill. Mem. A.S.M.E.

but the compressive stress in the core is very easy to evaluate. The shrinkages we can expect in different woods have been tabulated by the Forest Products Laboratory.[3]

We find that yellow poplar, for example, will shrink or swell about 0.28 per cent for a change of moisture content of 1 per cent. We may assume that the faces hold the core so that this swelling does not take place, since the shrinking and swelling of wood with the grain is negligible. If the longitudinal shrinkage of wood is computed from the volumetric tangential and radial shrinkages given in Technical Bulletin No. 174, a value of about 0.08 per cent for a moisture change of 1 per cent is obtained. However, the values given are not accurate enough for this use. The Forest Products Laboratory has found a value of 0.01 per cent to be about correct.[4]

The amount the faces stretch because of the tensile stress to which they are subjected is very small. Young's modulus for wood with the grain is about 20 times its value across the grain. (See "Report on Materials Used in Aircraft, 1920, by C. F. Jenkin, British Aeronautical Research Committee, His Majesty's Stationery Office, London, 1920.) Therefore if we assume that the faces are as thick as the core the extension of the faces is about one-fortieth of the amount that the core wishes to swell, or about 0.007 per cent for a moisture change of 1 per cent. The Young's modulus of poplar across the grain is about 80,000. The Forest Products Laboratory in Bulletin 556 of the Dept. of Agriculture, gives a value of about 1,600,000 with the grain; hence the value given above.

The stress, then, for a 1 per cent change in moisture content is about $0.0028 \times 80{,}000 = 224$ lb per sq in. It is evident that this stress may become quite high if the panel is allowed to assume a moisture content very different from the moisture content it had at the moment it was bonded together. The average moisture content of wood placed out of doors is about 12 per cent; therefore plywood which is to be used out of doors should be glued up at this moisture content if internal stresses are to be avoided. In the light of this fact it is not seen that the use of a dry glue overcomes the difficulties implied in the paper.

The author warns that the high temperatures used in making plywood with phenolic resins will cause serious internal stresses, particularly in thick panels. We have made successfully panels composed of nine plies of $^1/_7$-in. poplar veneer. The stresses set up by changes in temperature are similar to those set up by changes in moisture content. Coefficients of thermal expansion are given by Otis F. Hendershot,[5] and by the Forest Products Laboratory (project 134–8).

We find that when yellow poplar cools from 300 F to 70 F it will shrink about 0.354 per cent across the grain and about 0.023 per cent with the grain. The deformation in the core due to the cooling is then about 0.33 per cent. The stresses set up are, therefore, a little greater than those due to a change in moisture content of 1 per cent. It is evident that these stresses do not present a serious obstacle to the gluing of thick panels.

The author points out the danger that panels glued at high temperatures will blister when the press is opened because of the presence of moisture. This is a real difficulty. If water is held at a higher pressure than 67 lb per sq in. abs and heated to 300 F, the water will remain in the liquid form; however, if the pressure is reduced below 67 lb per sq in., steam will be formed and the temperature of the water will be reduced. Various pressures and corresponding boiling temperatures are given in standard tables of the properties of saturated steam. Exactly the same thing will happen to the water in a wooden panel during the hot-pressing process at the same corresponding temperatures and pressures. This can be shown by embedding a fine wire thermocouple in the panel and noting the temperature as the pressure is slowly reduced. The temperature will remain constant until the pressure is reduced far enough to allow steam to form. The formation of the steam reduces the temperature of the panel; thus the exact pressure at which steam forms can be determined. Experiments show that these pressures and temperatures check the values given for saturated steam exactly.

This fact suggests a method of opening the press so that blisters will not form. The pressure can be reduced rapidly to the critical pressure, and then reduced very slowly to atmospheric pressure. When the critical pressure is reached, steam will be formed at a rate depending upon the rate of reduction of the pressure. If the steam is formed slowly enough, it will force the press plates very slightly apart, and escape slowly without damage to the panel. This method completely solves the problem with respect to water. However, when panels are made with phenolic-resin glues, other gases than water vapor are formed, notably formaldehyde. We have not been able to find any data upon the pressure-temperature relations which obtain in the forming of these gases. We have found that we can open the press without blistering the panels if the pressure is decreased uniformly from its highest value (150 to 350 lb per sq in., depending upon the species of wood) to atmospheric pressure in a five-minute period. A more rapid rate of pressure reduction would probably be satisfactory, but not quite so safe. This method is so effective that we have made good panels without drying the colloid prior to the pressing operation.

The author states that it is safe to apply high temperatures to wood surfaces by the application of hot plates so long as evaporation takes place. Reference to tables of the properties of saturated steam will show (in the light of the foregoing) that if the pressure applied to the panel is 300 lb per sq in., no evaporation of water will take place (except a small amount around the edges of the panel) unless the temperatures is over 417 F. Evidently the evaporation of moisture has little to do with the temperature to which wood can be subjected in a hydraulic press without damage. We have found that the temperature at which the destructive distillation of wood starts depends upon the pressure applied to the wood. We have made panels successfully at very high temperatures. In one experiment two panels were made of Western pine at a temperature of 400 F. One panel was glued at a pressure of 200 lb per sq in., and the other at 300 lb per sq in. The panel glued at the lower pressure was turned to a golden brown by the heat. The other panel was hardly discolored at all.

We find that drying the colloid on the veneers before the pressing process has certain manufacturing advantages. We do not agree with the statement that "such a drying operation must be under careful temperature and time control to prevent excessive curing of the resin during the operation." The resin will not excessively cure if it is exposed to room temperatures. Reasonably rapid drying can be obtained at room temperatures if high air velocities are used. T. K. Sherwood[6] reports some careful work on the "rate of drying very wet blocks of pressed sulphite paper pulp containing over 100 per cent water, dried in a small tunnel through which air was forced by a blower." He finds that the rate of drying increases with the air velocity in a very regular manner and that it is directly proportional to the wet-bulb depression. We have run some tests upon the rate of drying of blood-albumin glue and the phenolic resin colloid under conditions similar to Sherwood's tests. We find that so

[3] "The Air Seasoning of Wood," by J. S. Mathewson, Tech. Bull. no. 174, U. S. Department of Agriculture, p. 11.

[4] "Longitudinal Shrinkage of Wood," by Arthur Koehler, Trans. A.S.M.E., 1931, paper WDI-53-2, p. 17.

[5] *Science*, Nov. 14, 1924, p. 456.

[6] "The Drying of Solids," *Industrial and Engineering Chemistry*, vol. 21, no. 10, Oct., 1929.

long as the glue is at all wet to the touch, the rates of drying are equal to those found by Sherwood. The rates of drying at air velocities between 1000 and 2000 fpm are high enough for commercial application, and no danger of excessively curing the resin will be encountered. The heat-transfer curves given by the author are interesting. The problem of heat transfer in flat slabs is a very old one. It was first solved by Fourier and recorded in his classic paper on the subject. H. P. Gurney and J. Lurie[7] have plotted the results of Fourier's complicated equations to non-dimensional coordinates so that they can be readily applied to certain problems. They give a special series of curves which apply to flat slabs. We have run some experiments which show that the diffusivity of yellow poplar is about 0.00128 cgs in the radial direction and that there is no surface resistance to the flow of heat between the plates of a hot press and the wood between them.

With these data available the temperature distribution in a poplar panel being pressed in a hot-plate press can be computed. The temperature will vary with the thickness of the panel, the point within the panel at which the temperature is computed, and the time. The author does not give the dimensions of the apparatus which he used in determining his curves; however, one would guess from his sketches that the data were influenced by the apparatus to a great extent.

The warning that a hot press "must have rigidity far beyond that required in a cold press working on a bundle of stock" is timely. This fact is often overlooked by panel manufacturers. It is easy to show that the variation in pressure upon the panel is large for small variations in the distances between the plates. The variation in pressure is directly proportional to Young's modulus of the wood and the variation of the distance between the plates and inversely proportional to the thickness of the panel. If we assume a Young's modulus of 80,000 and a variation in the distance between the plates of 0.001 in., the variation in pressure upon a panel 1 in. thick is 80 lb per sq in. A $^1/_8$-in. panel would have a variation of pressure of 640 lb per sq in. These figures show that it would be impossible to have any success at all gluing thin panels in a hot-plate press were it not for the fact that wood becomes somewhat plastic under the influence of heat and moisture.

Wilbur L. Jones.[8] The use of phenolic resins in plywood manufacture requires technical control and exactitude in advance of woodworking practice up to this time. There is a growing realization, however, that in order to make good panels there must be planning and sound methods promptly executed. The effect of one or two manufacturers offering waterproof, check-free panels will be to incite their competitors to emulation. There will be a brushing up of techniques all around, and perhaps in the end the hot press will become standard equipment in the plywood factory, as it is to a large extent in parts of Europe.

On reading the paper one wonders if there is not, perhaps, a fifth method of applying phenolic resins—a modification of the film method. Instead of resin-impregnated paper, would it not be feasible to use actual cores and crossbands carrying the resin in a film or coating on their surfaces? According to this plan, the resin manufacturer would furnish coated sheets of veneer all ready for hot-pressing. These coated veneers could be cut, inserted in the regular assembly, and pressed to a panel. This would avoid the disadvantages of the processes that the author calls Nos. 1 and 4, these being somewhat objectionable because they add water or some other liquid to the wood. As to process 3, the dry-powder method, the writer has never seen it in operation. It seems that the use of resin-coated crossbands and cores, as suggested, would possibly be more convenient to the plywood maker.

The greatest advantage arising from the use of precoated laminations might be realized in the making of five-ply and other multi-ply panels. The point is that a highly reactive resin could be furnished on the plies that will go near the center of the panel, where the heat has difficulty in penetrating. A more sluggish resin could be used next the face. It is almost necessary that a comparatively unreactive resin should be used under thin faces, especially of porous wood, because the resin must under no circumstances strike through. Resins may not be alkaline like certain glues, but they should not come through anyway, because they might still be noticeable on the finished surface.

If center plies were provided with a coating of highly reactive resin and crossbands with a less reactive resin, it appears that press times could be shortened and superior panels produced. Furthermore, the difficulties mentioned in connection with successive pressings on long panels might be diminished if resin-precoated crossbands and cores were used.

Perhaps it would be best if the resin manufacturer were to assume the responsibility of coating the interior plies, furnishing them properly spread as to density and reactivity of the resin. This is just a thought; it is just possible, however, that it is worth a trial.

A. J. Norton.[9] The simplest picture of what a resin is and what it does when used as a glue is a straight line extending from a molecule of phenol and formaldehyde at the beginning to the insoluble, inert set-up mass at the other end of the line. Each point on the straight line represents the formation of new chemical compounds, and without control it is possible to form many thousands of different products in one resin. It requires eight or ten hours under vacuum and pressure to bring the resin to the point where it is brittle. At this point it is soluble only in alcohol and spirit solvents, and a period of five minutes longer converts the material into an inert mass. It is the inertness of the product, with the high finish when heated under pressure, that attracts the attention to synthetic resins. It is readily seen that this extremely violent reaction must be controlled very closely in order to get a product which will set in five minutes to give the tough, inert resin.

In applying these resins as glue we turn to varnishes first because of our experience in laminating paper, and we felt that this was the proper method to use in woodworking. We found, however, that the varnish penetrated too much and the resin would not stay on the glue line. By making a very viscose varnish we could spread that varnish and by preheating it, could get it into the stage where it would stay viscose and remain on the glue line. This gave good gluing, but the cost was excessive.

The next attempt was to make powdered resin advance to the point where it would set quickly when heat and pressure were applied to the wood. This powdered resin is the type that the author spreads.

Previous to this in making a resin for binding sand cores, we found that where 3 per cent of powdered resin was necessary to give a certain strength, only 1 per cent of resin was necessary if this could be dispersed in a colloidal form to a very thin film. The next step in wood gluing, therefore, was to disperse the powdered resin into a colloidal form, which gave two advantages. First, the resin remained in the glue line, and, second, it could be spread thin enough and give good enough bond to help meet the price of common glues.

The film method of applying synthetic resins does not appeal

[7] "Charts for Estimating Temperature Distributions in Heating or Cooling Solid Shapes," *Industrial and Engineering Chemistry*, vol. 15, no. 11, p. 1170.

[8] Madison, Wis.

[9] General Plastics, Inc., North Tonawanda, N. Y.

to us because we do not feel that it can be manufactured cheaply enough, and in addition the paper content of the film is high enough to hurt the bonding properties.

In wood gluing it is not necessary to get the strength of the glue line stronger than the wood itself, and for cheap woods the colloid could be spread in a very thin glue line at a very much reduced cost. The powdered resin is very suitable for standard type of panels where the lower glue spreads are not necessary and where the cost is not so vital a feature. We also furnish the film form of material if anybody desires it, although at the high price it has no particular advantage to offer.

Author's Closure

Mr. Norris' interpretation of some of the text does not coincide with the thought the author tried to express. For example, the author had no intention of indicating that he "found it very difficult to use the colloid." On the contrary, the equipment at the author's disposal was exceptionally adapted to using colloid, and no trouble was experienced in handling it, other than that inherent in this form of adhesive. To those who daily contend with these inherent difficulties of a wet adhesive, they perhaps are not looked upon as such, and it is only in comparison with other methods that these troubles become so apparent. It is admitted that, for certain types of laminated work, the application of the adhesive in a wet state causes no trouble; for example, in laminating multiple plies of $^1/_8$-in. veneer. But it is still contended that the application of 12 to 14 lb of water to 1000 sq ft of $^1/_{24}$-in. or thinner veneers, and especially face veneers, will cause difficulties, which can be overcome by eliminating the application of this water.

The application of this water to the veneer obviously "introduces the same old problems of swelling and shrinking of wood." This water is supposedly applied evenly to the sheets of single-ply veneer, but the veneer will not evenly absorb this water, due to the non-homogeneous structure of the wood. The sheet, therefore, swells unevenly. The next step is to drive out all or part of this water, prior to pressing. This is, essentially, a redrying operation. It has become standard practice to redry single-ply veneer between hot plates, because this method is the most efficient and produces the flattest stock. In drying the colloid, hot plates cannot be used, nor can any appreciable temperature be applied. As Mr. Norris states, the drying must be accomplished by means of high-velocity air. This is not an efficient method in reference to time and power input, and the general run of single-ply veneer will not dry as flat by this method as it will if dried between hot plates.

The crux of the whole argument is: Why carefully redry the veneer to a moisture content of 4 to 8 per cent, then apply 12 to 14 lb of water per 1000 sq ft to the same veneer for use with the glue material, and then introduce a second redrying operation to eliminate this water? The question is pertinent, in view of the fact that the limitations of the second redrying operation prevent the use of recognized efficient redrying methods.

The author did not intend to imply that where gluing is done without introducing water there remains no problem of shrinking and swelling of wood. Everywhere there is moisture in varying quantity, and its results must be reckoned with, but why voluntarily add to the trouble, if it can be avoided? The diagnosis runs somewhat as follows:

Any variation in moisture content of wood results in swelling or shrinking. In wet gluing, water in the glue is absorbed by the wood, an increase of moisture content occurs, and swelling takes place. The same thing happens with colloidal glue. In the dry process, water is not introduced by the gluing process per se, and nothing is added to the problem of swelling and shrinking that did not already exist.

Moisture in wood creates a problem in nearly all its applications in the art, but not to the same extent that it has in glued-up work. Now if the gluing of wood can be accomplished without introducing moisture, the remaining problems, due to varying moisture content, are no more than those met with in other wood utilities.

It should also be noted that the sudden application of water to a wood surface, as when wet glue is applied, is quite different from adding moisture by the slow process of absorption in the air, and especially in hot pressing, where the time is exceedingly short, as compared with cold gluing. The sudden addition of glue and water introduces the old problems of shrinking and swelling in a particularly disturbing manner.

Mr. Norris' explanation of the problem of tension and compression in plywood, due to variable moisture content, is as helpful as any theorizing can be, when applied to so tricky a material as wood, but after all it is practical experience that convinces the average woodworker. Data obtained from a wood as homogeneous and ductile as yellow poplar do not mean much when applied to birch, maple, beech, and gum. The author does not believe that we know very much about the strain existing in any given piece of wood that has been processed. In his example of a three-ply panel, Mr. Norris states that the two faces are in tension, while the core is in compression, due to the fact that wood grain does not appreciably shrink and swell lengthwise. But he has stated only half the problem. When the faces of a three-ply panel are under tension with the grain, they are under compression across the grain, and the core, while under compression across the grain, is also under tension with the grain. These strains are also not confined to a single plane, but they lie in three separate planes, and that introduces new complications. Since the grain in the most homogeneous wood is never theoretically straight and never free from tension inherent in its make-up, it is never safe to prophesy what will happen when a disturbing influence like moisture is introduced, and the safest way is to avoid, as far as possible, that which produces trouble.

It is no doubt possible to dry the colloid on the sheets of veneer at room temperature. It is merely a question of time, but that is something to reckon with, because time means money. To produce high air velocity also costs money, and when all is said and done the whole thing is avoided if the resin is put on dry in the first place. High air velocity must also present some difficulty with very thin sheets of veneer, which curl readily when dried on one surface only, and they do not easily remain in position on a conveyor.

The evaporation of water within a panel being hot pressed follows the pressures and corresponding boiling temperatures as given in standard tables of the properties of steam, provided the exact specified conditions are attained. The conditions upon which the steam tables are produced do not always occur in a wood panel. They would, if each cell were a perfectly closed, elastic pressure vessel. Many of the cells are ruptured. There are faults or defects in the veneers, and Mr. Norris has pointed out the great difference in pressure resulting from as small a variation of distance as 0.001 in. between plates. The same difference in pressure would also result from 0.001 in. variation in thickness of the veneers. All these things are factors which make it extremely uncertain as to whether all the water is enclosed in a cell and subjected to the required pressure to prevent evaporation, and also as to what pressure is actually developed within any part of the panel. Consequently, in practice, a general slow evaporation does take place, and often a rapid local evaporation depends upon the soundness of the veneer.

There are plants operating on cottonwood whose regular procedure is to hot press the wet veneer direct from the lathe and turn out a dry panel. The moisture content of the panel going

into the press averages over 80 per cent and the dry finished panel 8 per cent. The pressure applied often exceeds 70 lb per sq in., and the temperature of the plates averages 260 F. Theoretically, the panel would not dry, due to the reasons given by Mr. Norris, but actually this practice produces commercial panels in daily operation.

The work of the author with phenolic resin has had, as its objective, the development of an extremely rapid process in order that the press investment can be kept at a minimum and one which would have a universal application. On the other hand, the Haskelite Corporation specializes in various types of plywood and is equipped with a highly trained technical staff which can develop technical control of plant processes. This situation is entirely different from that of the average plywood producer. Consequently, a procedure which may be entirely feasible for the Haskelite Corporation might not be well adapted for use by the trade in general.

Mr. Jones offers a thought which might result in an ideal procedure for plywood manufacture if it were not for some of the characteristics of phenolic resins and the economics involved.

The reactive resins are not stable, and reactive resin spread on sheets of veneer subjected to the heat of a summer day and the pressure which would be imposed on the bottom sheets of a pile would quite rapidly polymerize and lose their reactivity. This would result in no bond being formed when the veneers were hot-pressed. The economic factors which would mitigate against such a procedure would be the immense sums tied up in inventories and the extra freight involved in shipping the veneer from the cutting plant to a coating plant and thence to a plywood manufacturer. If the coating were done by the single-ply manufacturer, there would be no gain, only a shifting of the burden of coating from the laminator to the single-ply producer.

Mr. Norton has given a clear explanation of the formation of a phenolic resin which helps in picturing the final result in the finished glue line. It appears to the author, however, that some misconception might arise from the last of Mr. Norton's remarks. On a pound basis of resin, the dry powder and the colloid are practically the same in price. As light a spread can be applied with the dry powder as with the colloid, at least in a practical range, since the powder can be applied uniformly as low as $3^1/_2$ lb per 1000 sq ft of surface. Therefore, the dry resin would be just as applicable to the manufacture of panels in the lower price range as the colloid, and in addition the cost of drying the water applied with the colloid is eliminated.

In conclusion, the author wishes to correct an impression he may have given as to the limitation of hot-plate pressing to thick material. There is no practical limit as to the thickness of the final plywood which can be produced effectively with phenolic resin. But, in the opinion of the author, the lamination of single plies over $^1/_4$ in. thick, and particularly the lamination of lumber core panels, is something the practicability of which is yet to be established.

Fundamentals of Knife-Cutting Veneers

By JOHN N. and JOHN R. ROBERTS,[1] NEW ALBANY, IND.

In the last 25 years the per capita consumption of veneer in the United States has increased nearly 500 per cent. The manufacturers of veneer and its subproduct, plywood, are constantly finding new uses, better processes, and improved equipment. The principal methods of veneer cutting are undergoing marked changes and developments, requiring a clear understanding of the fundamental principles.

WHEN the first knife-cut veneer was produced is not definitely known, but before 1850 there were practical devices for producing thin sheets of wood veneer with a knife. The first machines were clumsy and crude. The early slicers had a straight cut without provision to avoid tearing the delicate cross-grain wood figure. The shearing or angling cut was not devised until about 1885. The first lathes had the cap, knife bed, and frame made of wood (see Fig. 1), and it was not until the later 1860's that the lines of the present-day veneer-cutting lathe began to take form. A great advance in the wood-working industry occurred in the period between 1870 to 1900, when all types of veneer-cutting machines were greatly improved and refined.

It is not the purpose of this paper to give a history of veneer cutting, but only to utilize enough historic background to show the necessity for the development of the devices that added so much to the quality of veneer and to the efficiency of the present-day machines.

A glimpse into government records reveals the size of the veneer-producing industry in the United States. The quantity of logs cut into veneer, by log scale (not surface feet of veneer), is shown in Tables 1 and 2, both according to species and states.

VARIETIES OF KNIFE-CUT VENEER

The three kinds of knife-cut veneer are shown in Fig. 2 and are as follows:

Rotary veneer, producing some 90 per cent of all veneer cut
Half-round veneer, an intermittent segment-cutting operation, used especially for face veneers
Sliced veneer, a flat-cutting process, used principally for face veneers or for special requirements.

Rotary Veneer

Rotary-cut veneers (Fig. 2-*B*) are principally used for non-decorative purposes, since little opportunity is afforded to develop the best grain and character figure. Rotary cutting may be speeded up more than the other processes, an especial advantage in producing cheaper grades for packages, cross-bands, and other commercial products, and it produces the large sheets so desirable in making structural plywood. When manned with experienced crews, the quality of rotary cutting may be excellent and entirely comparable with the other processes.

Veneer Utilization. There is an interesting economic factor in the use of rotary veneer that is shown graphically in Fig. 3. When a log is sawed into boards, the saw-kerf and slab losses exceed 25 per cent, with an additional subsequent loss from the planing necessary to smooth off the saw marks and reduce the boards to standard thickness. On the other hand, when a log is rotary-cut into veneer, these saw-kerf and slab losses are avoided, and the veneer is so smooth as to require only a sanding before finishing, and the only losses are in the initial rounding up and the final core. The conversion costs are strongly in favor of veneer, and the subsequent gluing up of thin veneer into plywood leaves the latter product less costly than solid wood on an equal-strength basis, but makes it somewhat more costly for equal thicknesses. Plywood has marked advantages, however, that permit the use of artistic face veneers, that increase its strength over solid lumber, that make it non-splittable, and that permit it to be bent and curved.

The evidence of census statistics proves that the per capita consumption of veneer in the United States has increased nearly 500 per cent in the last 25 years, while that of solid lumber has decreased over 50 per cent in the same period.

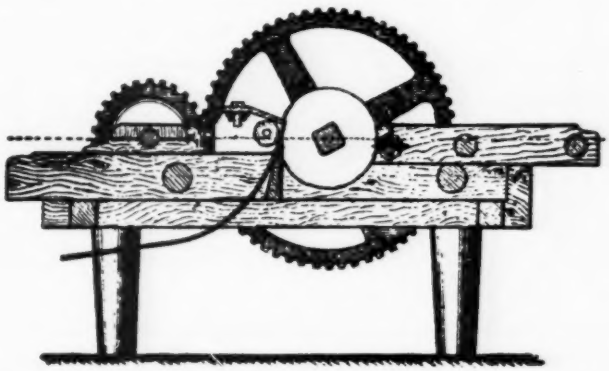


Fig. 1 Humphrey Lathe With Wood Frame

(Cutting pressure is exerted by roller, instead of bar. Face of knife is against log, as usual in early machines. Protruding top sections of table are pushed in as log diameter is reduced. Spur cutter for trimming veneer ends is opposite knife. Probably before 1860.)

Half-Round and Sliced Veneer

While rotary veneer is produced in large sheets, half-round veneer (Fig. 2-*C*) and sliced veneer (Fig. 2-*A*) are made in individual single sheets, each limited to the area of the flitch. These sheets are kept in consecutive order, and thus the almost identical character of two or four adjacent sheets permits artistic and mechanical combinations into beautiful face veneers, so much desired for cabinet, furniture, musical-instrument, and interior-finish design, a marked artistic advantage over solid wood.

Description of Processes. The half-round and rotary processes will be described together, as the two are produced on the same machine. While lathes are built primarily to cut rotary veneers, almost all lathes may be equipped with a stay-log attachment or

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[1] John N. Roberts is a native of Indiana. One of his first tasks, in the late 70's and early 80's, was to operate the lathe shown in Fig. 4. He has been actively engaged in veneer manufacturing during these 50-odd years and has originated and aided in the development of many of its improvements. His son, John R. Roberts, was graduated from Purdue University in 1918, and is now President of the Roberts-Dulaney Veneer Corporation.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

eccentric bar for holding flitches from which half-round veneer is to be cut.

The slicer is entirely different in design and operation, requiring a separate description.

ROTARY LATHE

A veneer lathe consists primarily of a rigid bedplate or frame, with sturdy fixed housings at either end. The housings are equipped with bearings for the main spindles. The log revolves on the spindles toward the knife, which is mounted on a movable assembly carrying the knife and pressure bar. This assembly moves in and out, at right angles to the spindle, on accurately machined and solidly mounted ways or slides built into the bedplate. The feed mechanism controls the rate of movement of this assembly, and consequently regulates the thickness of the veneer during cutting.

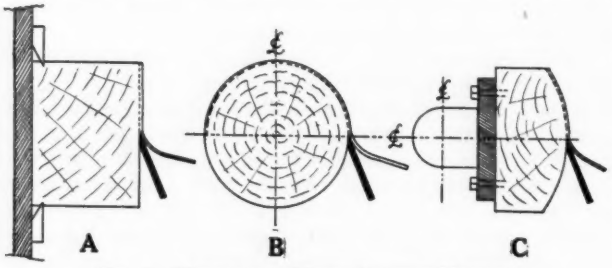


FIG. 2 METHODS OF KNIFE-CUTTING VENEER

(*A*, Sliced, flat shear-cut veneer. *B*, Rotary, continuous sheet from revolving log. *C*, Half-round, lathe cut when mounted eccentrically.)

The earlier wood-frame lathes (see Fig. 1) proved to be entirely too weak and yielding and were soon displaced by the all-metal lathe similar to those illustrated in Figs. 4, 5, and 6. These proved adequate for some years, but did not meet the exacting demands of later years. The major improvements for improved quality and increase of output were:

1 Additional strength and rigidity in the main parts of the machine
2 An efficient feed mechanism that would accurately control veneer thicknesses
3 Convenient and easily operated adjustments for the knife and pressure bar.

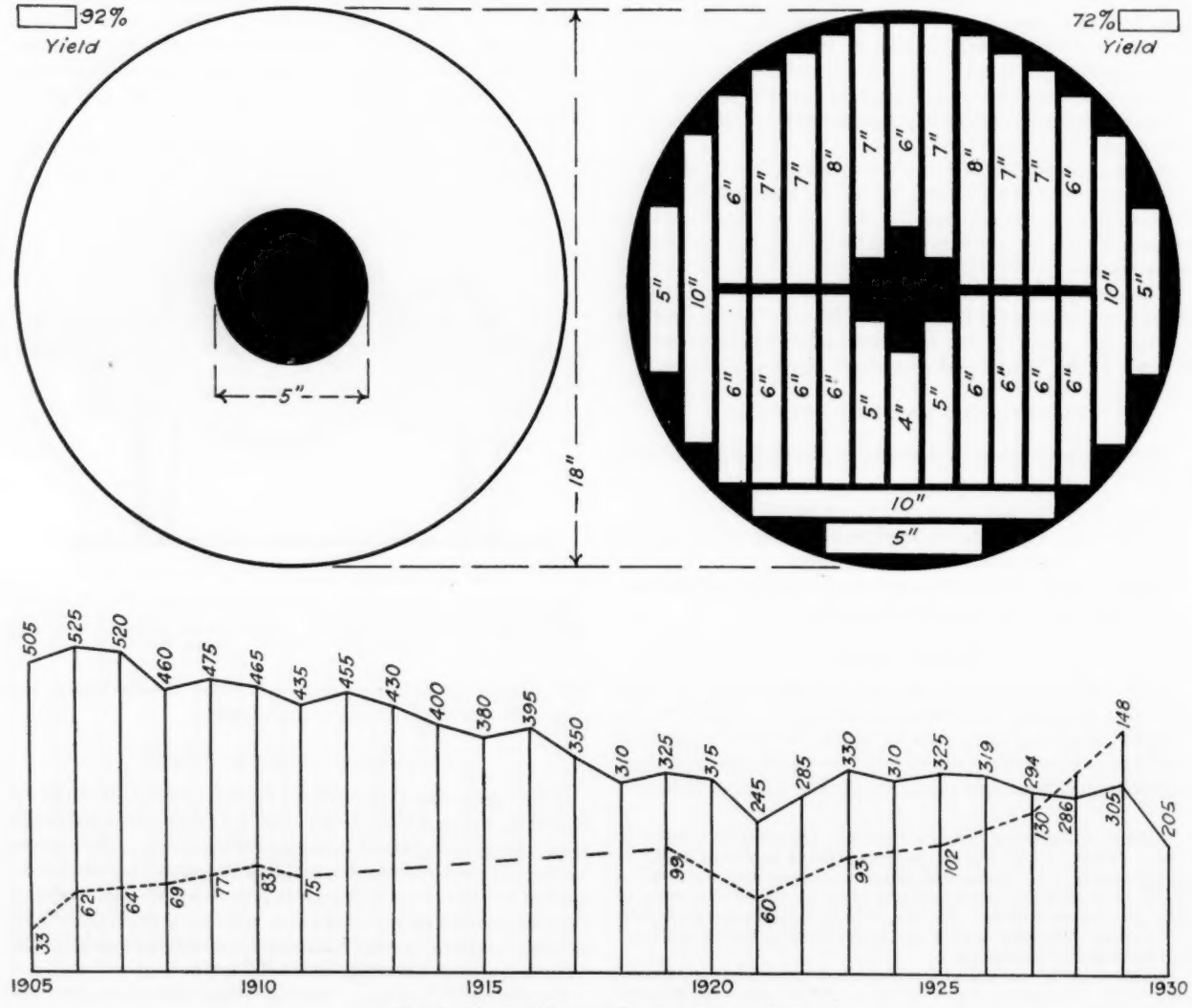


FIG. 3 CENSUS DATA ON VENEER PRODUCTION

(Above: Cross-sections of 18-in.-in-diameter cylinders showing the rotary veneer yield of 92 per cent of total area, compared with 72 per cent on sawn lumber. Unless lumbering operations are carefully managed, this lumber yield seldom exceeds 60 to 65 per cent. The standard Scribner-Doyle log scale allocates 58 per cent yield to such a log. Lower: Graphic chart of 25 years of lumber and veneer per-capita consumption in the United States. Lumber is given in log scale feet, which is practically board measure. Veneer is surface feet 1/16 in. thick, or log scale × 16.)

Strength Requirements

It is essential that a lathe be built on a sturdy and unyielding bedplate so as to give the machine rigidity. The bedplate carries the entire stress of mounting or chucking the logs. The housings at each end carry the main spindle bearings and are mounted on the bedplate. They must be heavy enough to withstand the tremendous torsional strains applied on the main bearings by the cutting stresses and leverage of the spindles.

Main Bearings. The main bearings on the early lathes were made of babbitt metal and were much smaller than those in use today. Even when the size of these bearings was substantially increased, the babbitt wore unevenly, broke out, and had to be frequently replaced. The bearings were next made of cast iron. This proved satisfactory, if they were given care and lubrication. One of the best bearings in use today is lined with fiber, which is less likely to score or wear the spindle shafts than cast iron and is easily renewable.

Knife Bed. The knife bed of an early lathe was probably the most rigidly built part. Much less importance was attached to the rigidity of the cap or pressure-bar holder, as its chief purpose at that time was to regulate the thickness of the veneer, without regard to its firmness and quality. These parts were made in various shapes of cast iron, superseding the earlier wood parts. On modern lathes the knife beds and caps have come to be very heavy castings designed to hold the pressure bar and knife in

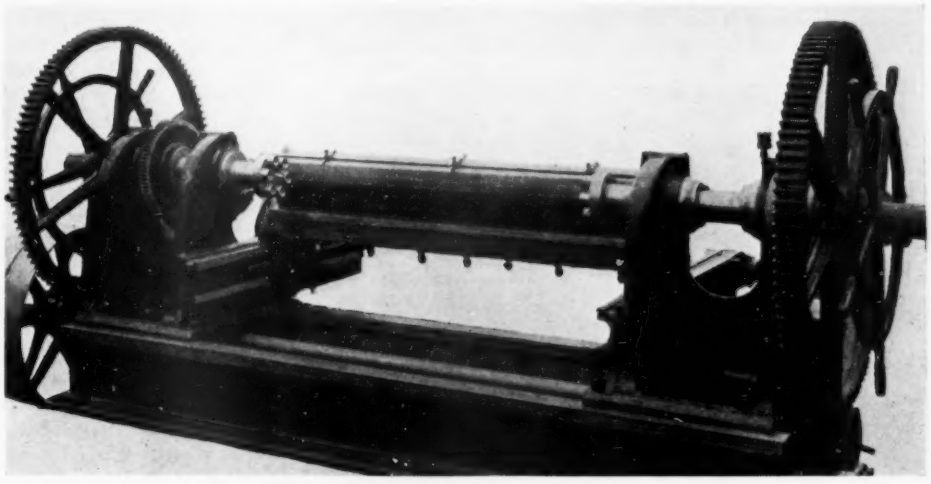


Fig. 4 Front View of an Early Lathe, Around 1870–1880

(Contrasting this with Figs. 8 and 12, notice the light bed plate, inadequate housings, and combination cast-iron and babbitt bearings; hand windlass wheels for end movement of spindle; bevel on knife against wood and pressure bar with practically no cap, and adjustments lacking except end screws; main bearings are split nearly vertical; feed screws are low with inadequate ways; spindle chucks have "vacuum" cups to prevent log twisting; ratchet and pawl feed is quite indistinct. This is the first lathe to establish the lines of modern design.)

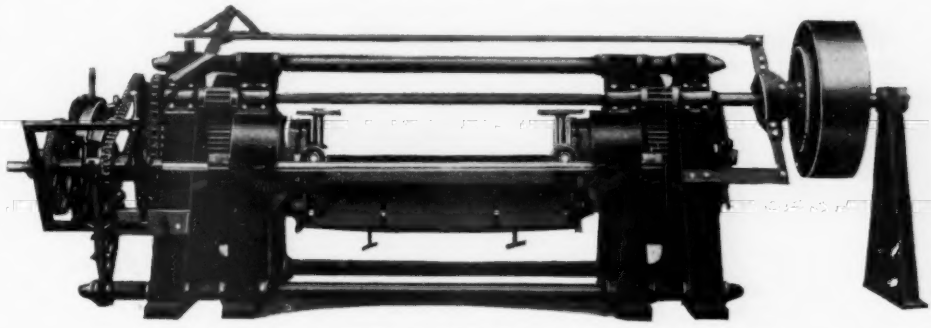


Fig. 5 Different Type of Mid-Period Lathe (1890–1900)

(Back view with feed mechanism at left, with sprockets and change gears. Cap, for pressure bar, is light. Knife bed very deep and heavy. Heavy rods or bars, two below and one above, substituted for heavy bed plate. Pressure screws are vertical rather than horizontal.)

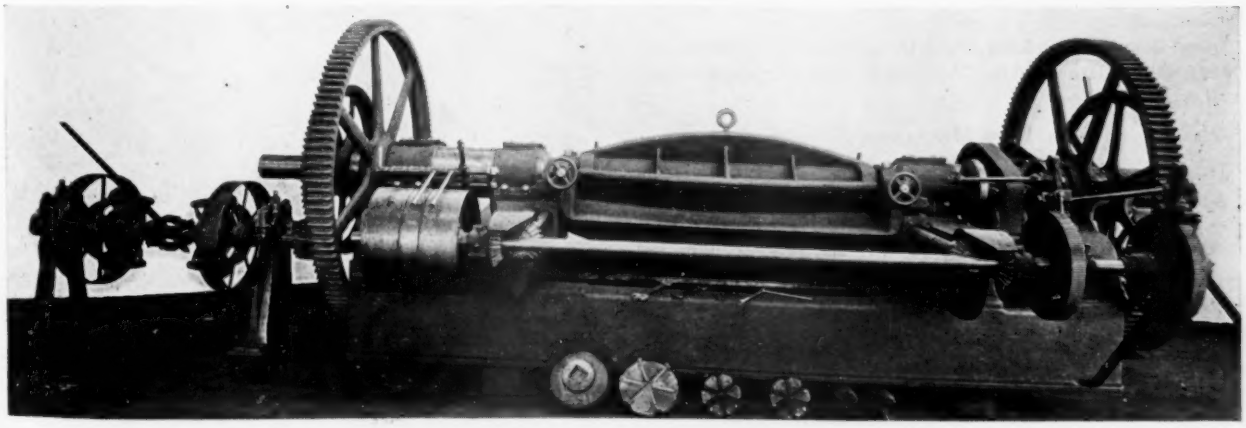


Fig. 6 Back View of a Lathe Developed in the 1880's

(Right: Double-ratchet feed mechanism, giving practically uniform feed. Center: Cross-shaft with bevel gears near each end connecting to feed screws with ratchet wheels at right end and double T & L pulley at left, with separate source of power to enter and withdraw knife from wood. Left: Another separate power source for chucking, with double clutch to move spindle in and out of logs. Note: Bearings split at approximately 45 deg. Knife bed and cap of sturdy modern design. Different size and shape of chucks shown below at center.)

alignment with each other and to be as nearly as possible free from spring or vibration.

Temperature Expansion. The heat from the log has some tendency to cause expansion in the knife bed and cap, particularly the cap, which may change the alignment between the pressure bar and the knife. The distribution of metal in these castings

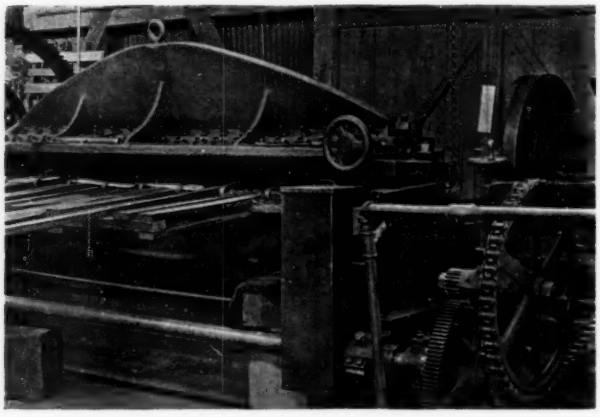


FIG. 7 COMBINATION GEAR AND CHAIN-FEED MECHANISM
(Back view of lathe, with sprocket wheel and gear train at right. Cross-shaft at lower left drives feed screws.)

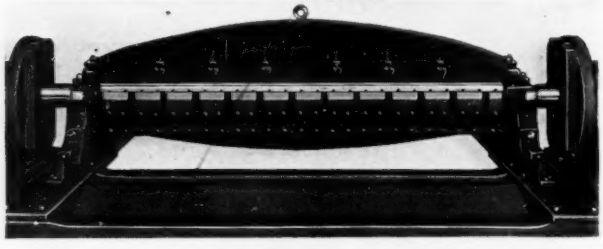


FIG. 8 MODERN STURDY LATHE
(Notice weight and rigidity of cap casting, above, of knife bed, behind knife, and massive bed plate, below. The sectional holding or clamping bar permits using narrow veneer knives, thus lengthening their useful service, as well as holding the knife more firmly than do the standard cap screws in knife slots.)

should be such as to prevent this as far as possible. In very long machines it is necessary to put a hog chain and adjusting screw on the cap, with which it can be sprung back into alignment with the knife bed.

There is nothing unusual about the strength requirements of other parts of a lathe, and they are designed according to standard engineering practice.

FEED MECHANISM

This device, controlling the "in and out" movement of the assembly carrying the knife and pressure bar, has undergone quite radical changes during the transition from the primitive wooden-frame lathe to the modern sturdy and nearly automatic veneer-cutting lathe. The principal types of past and present feed mechanisms are:

1 Hand control, a crude and early type requiring an experienced operator, but practically obsolete
2 Ratchet-and-pawl feed, originally an intermittent apparatus, but later practically continuous; seldom used on lathes, but standard on slicers
3 Combination gear-and-chain feed, continuous, and in general use on lathes
4 Gear-box feed, like the shift gearing of an automobile.

Hand Control. In the lathes built before 1870 the thickness of the veneer was regulated by the distance between the pressure bar, or pressure roller, and the knife. The feed on these machines was accomplished by a hand wheel, which operated feed screws to keep the pressure bar or roller pressed firmly against the log. In rounding up a log or in cutting half-round veneer, the wheel was marked, and the feed was advanced by hand for the thickness of the veneer, every revolution, while the knife was out of the cut. A hand-chain equipment, similar to those still in use on some veneer saws, was also used on half-round machines.

Ratchet and Pawl. On this earliest mechanical feed the screws that carried the knife against the log were operated by bevel gears from a cross-shaft. The cross-shaft was turned by a keyed ratchet wheel and was advanced by pawls, which in turn were operated by a crank geared to the spindle. The length of this crank was adjustable so that the number of teeth advanced by the pawls on each revolution of the crank could be varied in accordance with the thickness of the veneer to be cut. This is shown in Fig. 6. Of course, one pawl rocking back and forth gave an intermittent feed. A practically constant feed was gained by increasing the number of pawls that alternately engaged the ratchet wheel or wheels. The ratchet feed in its multiple form was satisfactory and had the advantage of making it possible to vary the feed and consequently the thickness of the veneer to any desired extent rather than to a limited number of set feeds, as is the case with the direct-geared or chain feeds in general use today.

While largely superseded on the lathe, this ratchet-and-pawl

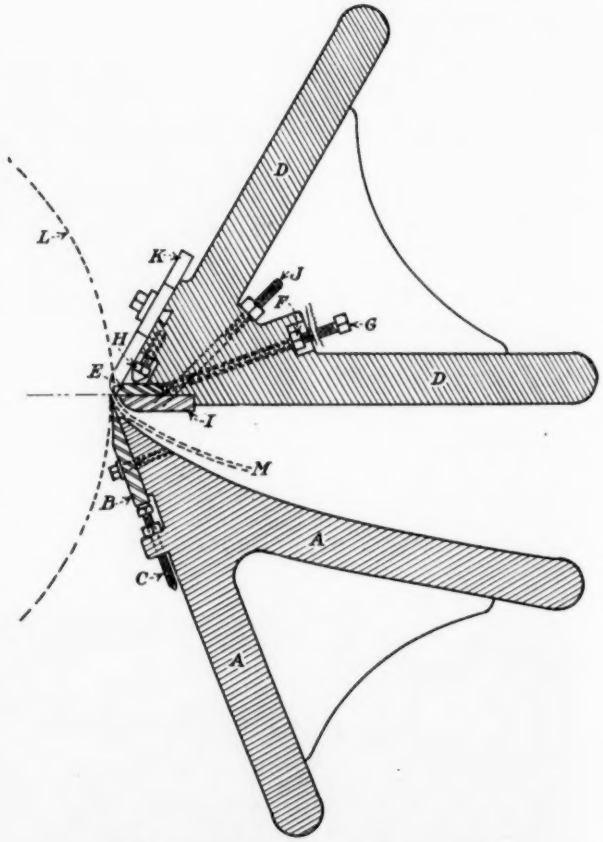


FIG. 9 DETAIL OF COMBINATION ASSEMBLY
(*A*, knife-bed casting; *B*, knife; *C*, knife-adjusting screws; *D*, cap or pressure-bar bed casting; *E*, pressure bar; *F*, pressure-bar adjusting bolts, in tension; *G*, pressure-bar adjusting screws, in compression—*F* and *G* alternate in cap casting; *H*, pressure-bar adjusting screws; *I*, pressure-bar auxiliary plate; *J*, pressure-bar auxiliary plate bolts; *K*, spurs, for trimming edges of veneer as cut; *L*, log; *M*, veneer.)

feed has become the approved method on slicers, where it meets the conditions, since the feed is on the back stroke when the knife is out of the wood.

Combination Gear-and-Chain Feed. This device was developed about the same time as the safety bicycle, which brought this power-transference method into many other mechanical applications. Sprocket wheels and gears (see Fig. 7) are changed for various speeds. Its full flexibility is limited by the number of available sprockets and the gear combinations, a limitation that sometimes prevents the desired accuracy in thickness. Its simplicity is in its favor.

Gear-Box Feed. This is a recent adaptation, and eliminates the delay and untidiness of gear changing, and can be entirely housed in an oil bath. While it gives great promise of surpassing the other feed mechanisms in convenience and flexibility, it has not been used long enough to become generally adopted. Such an enclosed feed mechanism has the advantage of permitting thicker initial cuts in rounding up the log, before setting the veneer thickness.

Separate Power for Withdrawing the Knife. In backing the knife away from the spindles with any of the feed mechanisms, the power connection is always disengaged and the feed screws are reversed and operated by some separate source of power (see Figs. 6 and 11-*D*).

Thickness Tolerances. In cutting veneer today for some purposes, it is desirable to be able to vary the thickness enough to take care of the shrinkage in drying, in order that the finished dry veneer may be of the required thickness. In view of the fact that the volume of this type of cutting is increasing, it would seem that there is an opportunity for the engineering profession to further develop lathe feeds that are both flexible in range and nearly automatic in operation.

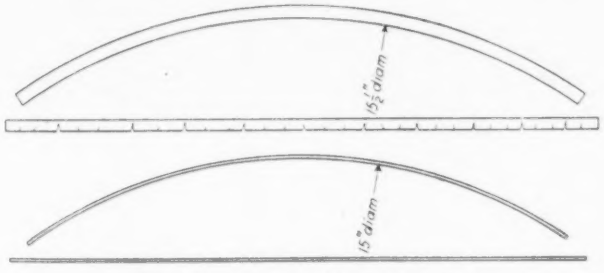


FIG. 10 TIGHT AND LOOSE CUTTING

(Above: Rotary veneer, 1/4 to 1/8 in. thick, with cutting checks due to inadequate pressure. These cannot be entirely eliminated, but in many grades of rotary veneer they are unobjectionable and cause the veneer to flatten better. Below: Rotary veneer, 1/28 in., standard thickness for face work. Proper pressure produces tight cut veneer and eliminates cutting checks, thus permitting both sides of the veneer to be used, as in matched faces.)

KNIFE AND PRESSURE BAR

Possibly the best idea of the development of this combination assembly—the knife, knife bed, cap, and pressure bar—can be gained by comparing the illustration of the early lathe (Figs. 4, 5, and 6) with that of a modern machine (Fig. 8). The details of this assembly are shown in Fig. 9. It would seem that the modern lathe builders have gone to extremes in putting so much metal in the knife bed and cap castings. As a matter of fact, while a much lighter machine will cut a log into veneer, it is possible on the heavy machines to increase the pressure to as high a point as the wood will stand without mashing. This pressure, properly applied, holds the wood fibers intact while the veneer is being cut from the log, and the result is a sheet of veneer that is not only smooth on both surfaces, but is internally sound and has its full strength value (see Fig. 10).

Knife-Bed Adjustments. Two castings, one at each end of the knife bed, rest on the feed slides and carry the knife-bed and cap assembly. There are two adjusting screws at each end, by means of which the assembly can be raised or lowered or tilted to bring the knife into cutting position. When the adjustment is made, the knife bed is held in position by tightening the center bolts, which hold it firmly to the carriage castings. These adjusting screws are shown in Fig. 11-*A*.

By raising or lowering the back screws, the knife can be tilted into or out of the wood. The angle of the beveled surface of the knife with a tangent to the surface of the log at the point of contact is called the "lead." "Applying more lead" or "putting in lead" means tilting the knife into the wood. Tilting it out of the wood is called "taking out lead."

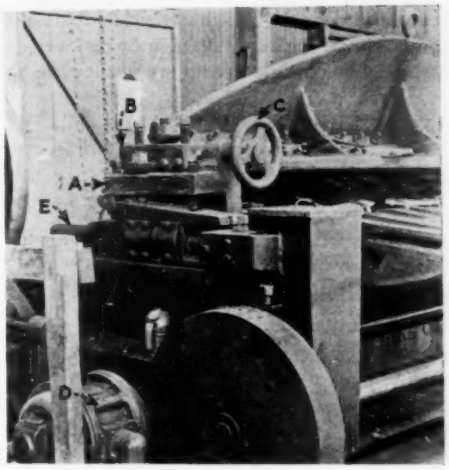


FIG. 11 KNIFE-BED AND CAP ADJUSTMENTS

(*A*, knife-bed casting, with two outside screws to adjust "lead" and center clamp bolt; *B*, cap casting, similarly adjustable: *C*, pressure screws to move cap toward knife; *D*, reversing motor to back combination assembly away from the log; *E*, feed screw, actuated by feed mechanism and advancing knife into the wood.)

Cap or Pressure-Bar Bed. The cap is mounted on top of the knife bed in the same manner as the knife bed is mounted on the carriage castings. The screws upon which it rests raise or lower the cap with respect to the knife bed (see Fig. 11-*B*). There are in addition, however, two "pressure screws" (Fig. 11-*C*), one at each end of the cap, which move the cap and pressure bar horizontally toward the knife and hold it in cutting position.

Feed Screws. The feed screws are operated by the feed mechanism and revolve in the carriage castings (see Fig. 11-*E*), moving the combined knife-bed and cap assembly forward or backward on the slides. The feed screws carry the full force of the pressure applied in cutting. It has been found best to locate these screws high enough to minimize the tendency of the knife bed and cap to rock back on the slides and spring away from the wood.

STAY LOG FOR HALF-ROUND VENEER

The stay log (see Fig. 12) is the adjustable bar mounted eccentrically between the spindles of a lathe on which flitches are held while being cut into veneer. Veneer cut in this manner is termed "half-round" (see Fig. 2-*C*). The purpose of this type of cutting is to produce a veneer with a more regular grain design than full rotary veneer, where the log is chucked centrally in the heart between the spindles and the knife follows the annular rings fairly closely as the log revolves. When half a log or flitch is attached to the stay log, the heart of the log is away from the center of the spindles, so that as the flitch is revolved, the path of the knife crosses the annular rings. The character of the half-round grain design can be varied by adjusting the stay log farther off center,

between the broad-grain design of the full rotary veneer and the striped-grain design of the sliced veneer.

It is possible to get the half-round grain effect without using a stay log, by chucking logs eccentrically or to one side of the heart. The main objection to this method is that a considerable part of the best wood cannot be cut into veneer and remains in the discarded core.

The design of the stay log must be such as to give it ample rigidity. This would not be difficult if there were no limit to its size, but the back of the bar must clear the knife at every revolution. Hence a reasonably light but rigid and compact stay-log attachment has taxed the ingenuity of lathe designers.

The flitches to be cut into veneer are fastened to the stay log

TABLE 1 SPECIES OF LOGS ANNUALLY CUT INTO VENEER

	Quantity (1000 ft log scale)		
	1929	1927	1925
Domestic hardwoods:			
Red gum	285,035	250,247	237,465
Tupelo	88,123	113,729	48,311
Yellow poplar	71,989	56,899	50,770
Cottonwood	51,742	43,456	39,869
Birch	47,715	51,331	51,490
Maple	35,617	24,706	23,922
Walnut	26,893	20,786	17,573
Oak	20,409	19,502	20,661
Elm	19,459	16,945	8,503
Beech	17,358	11,268	4,708
Basswood	13,920	11,885	9,413
Sycamore	5,809	6,192	4,836
Ash	1,506	1,262	1,103
Magnolia	1,238	4,666	3,786
Willow	1,086	[a]	[a]
Other	13,723	20,780	4,255
Total	701,622	653,654	526,665
Per cent of whole	62.8	68.0	71.8
Domestic soft woods:			
Douglas fir	162,415	101,220	101,869
Yellow pine	119,785	92,153	45,818
Spruce	38,198	53,642	20,033
Cedar	30,821	19,409	17,093
Western pine	22,717	13,379	4,876
Cypress	7,053	3,450	4,511
White fir	4,065	4,059	[a]
Sugar pine	3,588	[a]	[a]
White pine	2,613	3,027	[a]
Redwood	1,308	[a]	[a]
Hemlock	1,059	[a]	[a]
Total	393,622	290,339	194,200
Per cent of whole	35.3	30.2	26.5
Imported woods:			
Mahogany	3,741	11,954	7,471
Spanish cedar	3,917	4,829	5,149
Other	10,008	785	1,114
Total	17,666	17,568	13,734
Per cent of whole	1.3	1.8	1.7
Grand total	1,112,910	961,561	734,599

[a] Not separately tabulated.

TABLE 2 LOGS CONSUMED IN CUTTING VENEERS, BY STATES

	Quantity (1000 ft log scale)		
	1929	1927	1925
Washington	181,826	140,319	112,484
North Carolina	96,417	78,171	64,640
Florida	96,199	77,321	29,999
Mississippi	75,424	80,218	57,268
Arkansas	63,363	57,611	58,122
Oregon	55,644	32,091	29,793
South Carolina	48,960	35,311	29,669
Louisiana	47,617	35,012	34,395
Alabama	43,221	41,424	38,561
Georgia	40,385	38,646	11,412
Wisconsin	38,724	37,864	34,292
New York	37,377	27,364	24,095
Texas	32,488	29,216	5,711
Indiana	30,779	25,092	15,474
California	28,698	18,379	10,454
Tennessee	24,710	25,586	40,454
Virginia	24,688	33,319	26,416
Kentucky	21,437	24,193	23,309
Michigan	20,434	20,811	18,096
Vermont	19,056	12,217	10,255
Missouri	16,602	20,825	17,380
Illinois	15,821	11,414	11,967
Ohio	11,546	13,259	4,360
Other states[a]	36,174	45,898	25,993
Totals	1,112,910	961,561	734,599

[a] Less than 10,000 each.

TABLE 3 LATHE-SPINDLE REVOLUTIONS (PER MINUTE) FOR CONSTANT DELIVERY SPEED

Log diameter, in.	Constant delivery speed, ft per min			
	100	125	150	200
7	54.5	..[a]	..[a]	..[a]
8	47.7	59.8	..	..
9	42.5	52.7	..	..
10	38.2	47.8	57.2	..
11	34.7	43.4	52.1	..
12	31.8	39.8	47.7	..
13	29.4	36.7	44.1	58.8
14	27.3	34.1	40.9	54.5
15	25.4	31.8	38.2	51.0
16	23.8	29.8	35.8	47.7
17	22.5	28.1	33.7	45.0
18	21.2	26.5	31.8	42.5
19	20.1	25.1	30.1	40.2
20	19.1	23.9	28.6	38.2
21	18.2	22.7	27.3	36.4
22	17.3	21.7	26.0	34.7
23	16.6	20.8	24.9	33.2
24	15.9	19.9	23.9	31.9
27	14.2	17.7	21.3	28.4
30	12.8	15.9	19.2	25.5
33	11.6	14.5	17.4	23.2
36	10.6	13.2	15.9	21.2

[a] Over 60 rpm considered impractical. Under 20 rpm probably unnecessary.

TABLE 4 ACTUAL LATHE DELIVERY SPEEDS, ON 48, 36, 30, AND 24 RPM, WITHIN 10% OF STANDARDS

Standard delivery speed	Diameter of log, in.	Spindle rpm	Ft per min
100	8	48	100.5
	10	36	94.2
	12	30	94.2
	14	30	110.0
	16	24	100.5
	18	24	113.1
125	10	48	125.7
	12	36	113.1
	14	36	131.9
	16	30	125.7
	18	24	113.1
	20	24	125.7
	22	24	138.2
150	12	48	150.8
	14	36	131.9
	16	36	150.8
	18	30	141.4
	20	30	157.1
	22	24	138.2
	24	24	150.8
	26	24	163.4
175	14	48	175.9
	16	36	150.8
	18	36	169.6
	20	36	188.5
	22	30	172.8
	24	30	188.5
	26	24	163.4
	28	24	175.9
	30	24	188.5
200	16	48	201.1
	18	48	226.2
	20	36	188.5
	22	36	207.3
	24	30	188.5
	26	30	204.2
	28	30	219.9
	30	24	188.5
	32	24	201.1
	34	24	213.6
	36	24	226.2

with lag screws. The threads on these screws should be shaped so that they clear themselves and do not clog up with wet wood when used repeatedly. This involves a fairly sharp angle of thread toward the head and a much flatter angle on the under side of each thread when compared with the axis of the screw.

Variable-Speed Lathe Drives

Obviously it would be highly desirable on many lathe operations to have a constant flow of veneer on the delivery table, but there are certain limiting conditions that must be considered. Economy in time requires that all preliminary cutting, or "round up," be done at as high a speed or as thick a cut as practicable, since it is waste.

The average spindle speed of most production lathes is around 30 rpm, except for face veneer, which often is 20 rpm. On an average of 30 rpm the delivery speed will vary from 188 fpm on a 24 in. in diameter log to 63 fpm when the log is reduced to 8 in. in

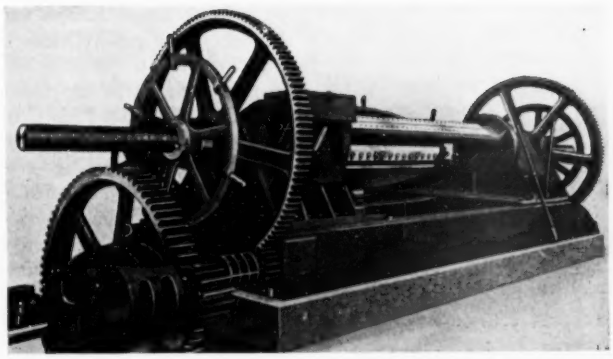


FIG. 12 MODERN LATHE WITH STAY LOG AND DIRECT MOTOR DRIVE
(Notice the massive bed plate, with broad single housings. Stay log is adjustable in distance from spindle center.)

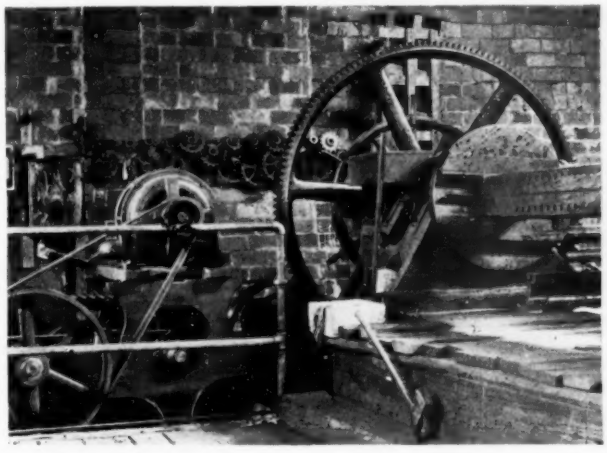


FIG. 13 VARIABLE-SPEED DRIVE FOR LATHE
(Motor above, connected to drive pulley on variable-speed device by V-belt. Variable-speed device, connected to main gear on lathe spindle, is not shown. Notice change gears for feed mechanism against the brick wall.)

diameter. It is difficult to arrange delivery clipper equipment and to man a crew to work efficiently over such a wide range. Various plans have been devised, none of them wholly satisfactory, to meet this condition. It is further complicated by the fact that $^1/_{20}$-in. veneer is light and easily handled but more fragile than thicker veneer, while $^1/_4$-in. veneer is hard to handle and must be cut more slowly, due to power demands. The species or character of log also has a bearing on speed; i.e., basswood can be cut faster than hard maple.

Hence the ideal arrangement would be three or four constant delivery speeds—at 100, 125, 150, and possibly 200 fpm. The speed requirements of such a program are outlined in Table 3. Such an arrangement should have the main drive connected with the feed-screw mechanism so that the operator would have no further concern about speed regulation after once starting the production cut subsequent to the round-up. This could be accomplished mechanically by certain variable-speed transmissions, although few such installations are yet in operation. It is also believed possible to devise electrical equipment to serve the same purpose.

Some veneer mills utilize a series of fixed speeds, such as 24, 30, 36, and 48 rpm, and set these speeds (by change gears or belts) for each log, or in the case of large logs, interrupt the operation to change speeds. The result of such a plan is shown in Table 4.

Variable-speed motors have been adopted for such a program with fair success, but many types of motors will race unless under approximately full load. Such racing is objectionable when entering or leaving a cut. The variation in power demand between $^1/_{20}$ in. and $^1/_4$ in. thicknesses also causes racing and is a handicap to a variable-speed motor. Frequency changers are preferable to a variable-speed motor, giving greater flexibility and more constand speeds under varying loads.

In the case of face-veneer operations or stay-log work, the speed-control problems are of a different nature, since wood characteristics must be considered. Delivery speeds, although still important, must be subordinate to veneer quality. On stay-log work the constant-delivery speed is scarcely a factor, since the total thickness of the flitch is rarely a large part of the total cutting diameter.

The problems of either variable speeds on the lathe spindle or constant veneer delivery speed offer an excellent opportunity for engineers to serve the veneer-cutting industry. The problems are almost entirely mechanical and demand adequate solution. Veneer operators are schooled and experienced in their ever-present problems of wood artistry and need the cooperation of engineers in the mechanical phases of their equipment.

One illustration of a variable-speed drive on a lathe is shown in Fig. 13.

SLICER

It is generally conceded that the highest quality of knife-cut veneer can be produced on a slicer. However, a preference for a sliced veneer (Fig. 2-*A*) is generally due to a desire for either the "across the heart grain effect" or "straight stripe grain effect" or for longer veneer. Due to the center support to the flitch table

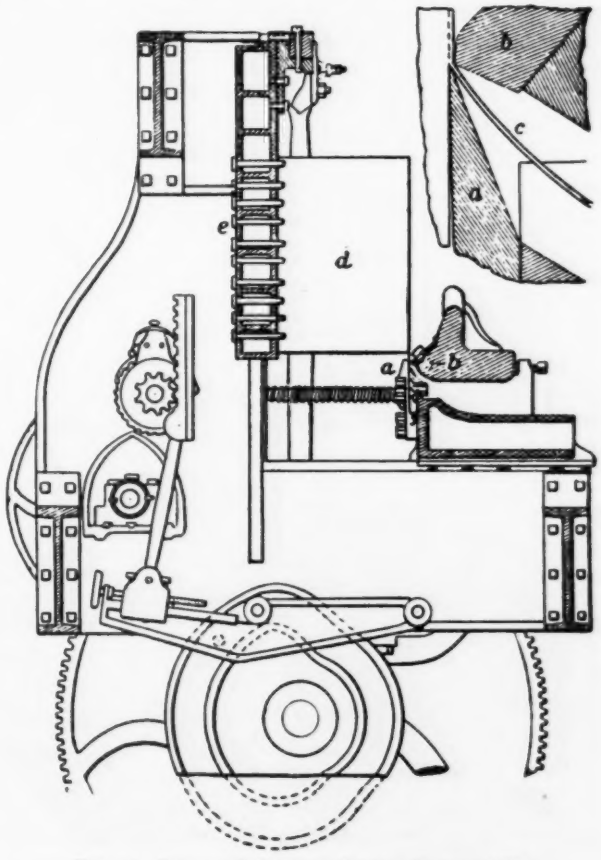


FIG. 14 PARKER & SLEEPER SLICER, BEFORE 1875
(*a*, knife, mounted with face against wood; *b*, pressure bar; *c*, sliced veneer; *d*, log. Ratchet-and-pawl feed mechanism, actuated by rack and pinion, shown against main casting. Feed screw shown below log.)

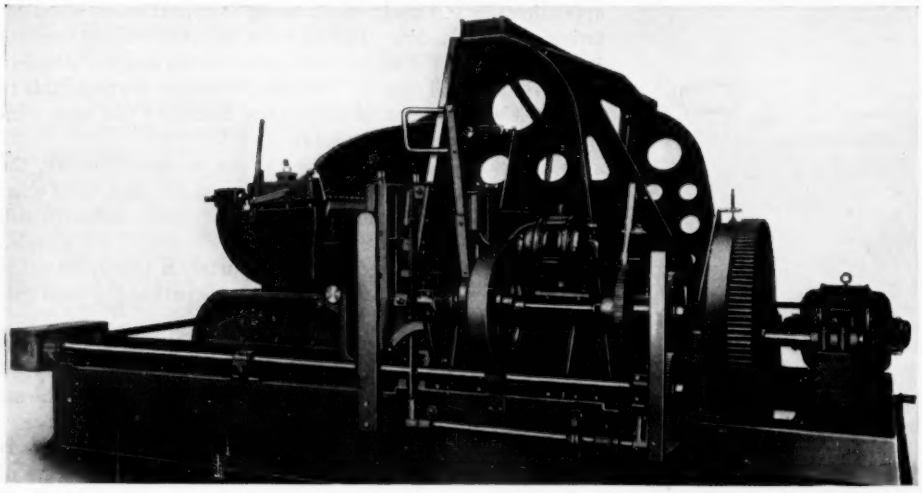


FIG. 15 VERTICAL-TYPE VENEER SLICER

(Knife bed and cap assembly at left above. Operating motor at right, and reversing motor against machine at center. Threaded feed screw at lower left. Three angling ways, for carriage travel, extend to cross-head across top of machine.)

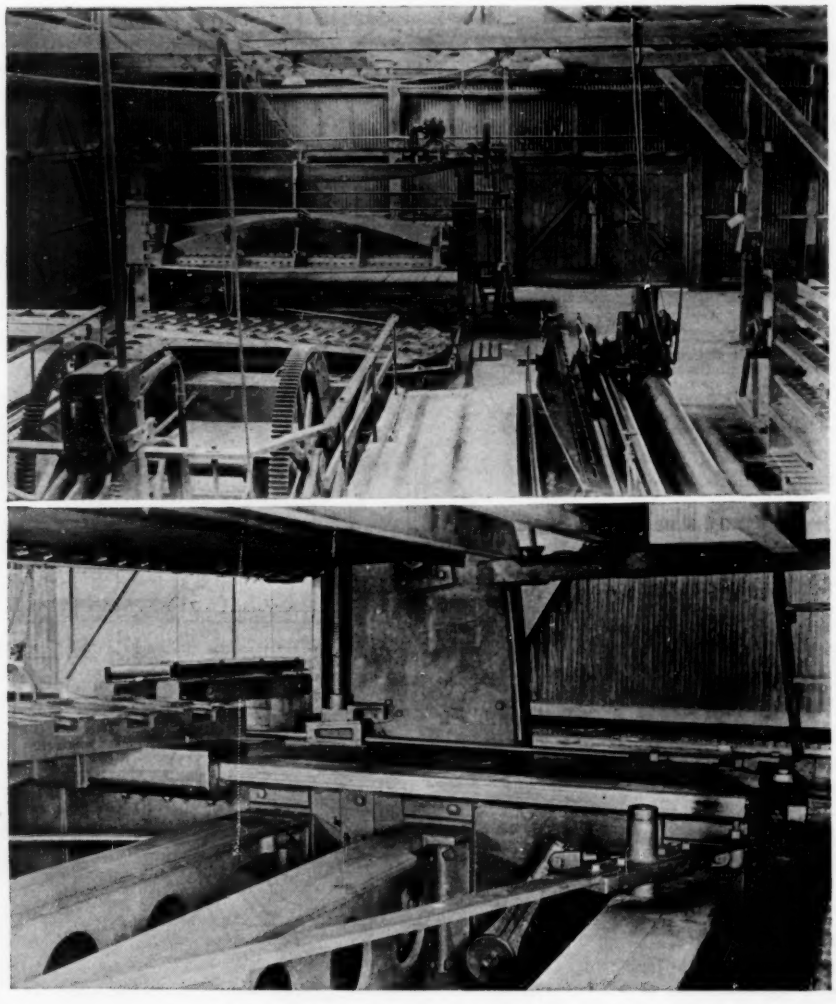


FIG. 16 HORIZONTAL-TYPE VENEER SLICER

(Upper: Front view, with angling flitch table in center, knife bed and cap assembly above. Driving gears at lower left, and end of knife grinder at lower right. Lower: Looking horizontally between the log carriage and knife, at upper left. Offset, tapered block, at center below vertical feed screw.)

on a slicer, longer flitches can be held more rigidly than is practical on a lathe. The length of slicers in use today generally runs from 12 to 17 ft, whereas lathes seldom exceed 10 ft in length.

The first slicer was probably little more than an elaborate splitting machine, possibly combined with a gigantic plane, cutting off somewhat irregular sheets for overlaying on solid wood.

One of the earliest known examples is the Parker & Sleeper slicer shown in Fig. 14, which had an all-metal frame, with a vertically reciprocating log carriage and a stationary knife, mounted with face toward the wood. The knife made a straight cut, without shear or draw, and represents quite a crude state of slicer construction.

VERTICAL AND HORIZONTAL SLICERS

Slicers have been built of various designs. They can be divided into two distinct types, vertical and horizontal. In the vertical type the flitch table or knife bed moves up and down. In the horizontal type the flitch table or knife bed moves horizontally.

There are machines of both types in use today, but in practically all cases in this country the flitch table moves and the knife-bed and cap assembly is stationary except for the feed and offset motion.

Vertical Type. The vertical machines built in this country follow the general principles of the Williams slicer, patented 40 to 50 years ago, and are the slicers in general use today.

Although the travel of the flitch on a slicer is reciprocating rather than rotating, as in the case of the lathe, the principles of veneer cutting are in both cases the same, and the construction of the knife-bed and cap assembly are similar.

The same requirements of weight and rigidity are applicable to the slicer as to the lathe.

The cutting stroke has a vertical motion of about 3 ft and a draw or lateral motion of about $1^1/_2$ ft. The draw gives a shear that eases cutting action and helps to prevent the flitch from breaking out at the back edge. The flitch table is mounted on three heavily built angling slides (see Fig. 15). The weight of the flitch table is counterbalanced by weights on the drive gears, which are connected with the flitch table by two pitmans.

The cutting occurs on the down stroke, and the feed takes place on the return, or up, stroke. The feed is of the single ratchet-and-pawl type and is capable of accurate adjustment to any desired thickness from zero to over $^1/_4$ in., which is above the practical usefulness of knife-cut veneer. As the feed takes place on the return stroke, an offset operates on the knife-bed and cap assembly that rocks it back to clear the flitch at the bottom of the stroke and returns it into cutting position at the top of the stroke. Regardless of how the offset operates, it is essential that the knife-bed and cap assembly be held rigid while the knife is in the wood.

Horizontal Type. The horizontal slicer has no draw cut and takes up more floor space than the vertical machine, but it has some features that are to its advantage (see Fig. 16).

The stroke is 8 ft long. The flitch table is pivoted at the center on a carriage in such a manner that the flitch can be turned in either direction to angles not exceeding 30 deg with the knife, to take advantage of the grain. This results in a flexible angling cut that will often be more favorable to wood grain than a fixed angle and results in smoother and sounder cutting. The offset raises the knife-bed and cap assembly directly on the feed slides so that the strain when cutting comes directly on the feed slides and not on the offset mechanism.

TOOLS AND TECHNIQUE

While marked differences have been indicated between veneer lathes and slicers in the frames, log (or flitch) mountings, and wood travel, yet there is a singular similarity in the knife and pressure-bar mechanisms. In fact, the similarity is so striking that the same crew, the same sharpening devices, and most of the adjustment principles can be used interchangeably on either type of veneer-cutting machine. Their descriptions will therefore be combined and their differences denoted.

Veneer Knives

For many years veneer knives have been made by the generally approved method of welding a strip of carbon tool steel on a soft iron or steel back, as shown in Fig. 17-*A*. All slots and holes for fastening to the knife bed are put in the soft back.

Knife Splicing. It is now possible to obtain a one-piece knife in almost any length desired, but ordinarily a jointed knife is satisfactory, and the extra expense of a single-piece knife is uncalled for. Most pieced knives are now welded so perfectly that it is practically impossible to locate the weld. Until this weld was developed, most makers put knives together with a dovetail key and threaded dowel pins, as shown in Fig. 17-*B*. Continued service is apt to cause this type of joint to become loose. This can be remedied by removing the dovetail key, filing the ends to give them clearance, and drawing the joint tight by replacing the key and peening the tapering sides of the dovetail.

Knife Mounting. In the early machines the knife was mounted with the bevel side toward the knife bed and away from the wood (see Fig. 17-*C*). This had three disadvantages: It was necessary to set the knife high enough for the bevel to be above the knife bed, which gave the knife insufficient bearing on the knife bed; it was imperative to clear the heads of clamping screws; and it permitted the cutting wear to come on the face side of the knife where it was difficult to grind off. On the other hand, the thrust of cutting came directly on the adjusting screws, with less strain on the clamping screws. The clearance was not so serious on lathes as on slicers, where the flitch passed the knife in a straight line parallel to the face and there was no curvature to the log to clear the bolt heads. This made it necessary to countersink the heads below the surface of the knife, which increased the difficulties of mounting the knife firmly.

Practically all modern veneer-cutting machines (both lathes and slicers) provide for mounting the knives with the bevel vertical (see Fig. 17-*D*) and against the log or flitch. This provides more rigid support for the cutting edge against the knife-bed casting and also supplies more adequate clearances for clamping and adjusting bolts. In the case of whetting or honing, if such becomes necessary, this method of mounting permits whetting the bevel rather than the face, which is the approved method.

There are now on the market several sectional knife-holding devices of the type shown in Fig. 8 that are designed to hold the knife more rigidly against the knife bed than is possible with the standard knife bolts. When properly installed and adjusted, these knife holders lessen the troubles caused by spring in the knife. The use of these sectional knife holders or clamps will also lengthen the life of a knife beyond the point where it can be held rigidly by standard clamping bolts.

Knife Temper and Hardness. Modern manufacturing methods

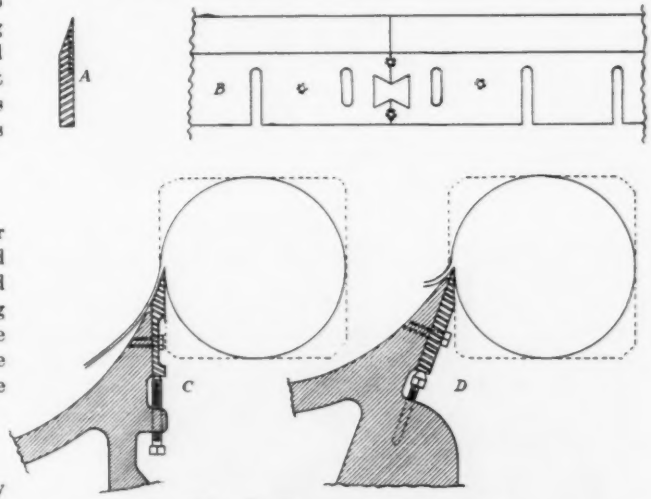


Fig. 17 Veneer-Knife Details

(*A*, showing hard-steel cutting edge welded to soft-steel back; *B*, method of splicing long knives; *C*, face of knife mounted vertically, with limited clearances, point of adjusting screw against knife; *D*, bevel of knife mounted vertically, with ample clearances, head of adjusting screw against knife. Relation of knife mounting in slicers shown by dotted line.)

and new alloy steels have permitted knives of more uniform temper and of tougher steel. It is possible to secure knives of almost any desired degree of hardness. The right temper depends on the kind of cutting and the wood to be cut. For face veneer, a hard knife should be used. If the operator uses care, a hard knife holds a straighter edge, which helps materially in keeping alignment of the knife with the pressure bar.

In cutting pine and similar woods, a softer knife must be used. The hard limb knots in the wood will cause a hard knife to break out. There is quite a difference of opinion as to the proper temper of knives, but the knife makers, with years of experience and contact with knife users, can be relied upon to furnish the proper knife.

Grinding and Whetting the Knife

Grinding machinery has been perfected to such a degree that the grinding of a knife is a simple matter. Originally the devices used in grinding were mostly contrivances developed by the various plants for their own use, some of them good, but all of them slow and requiring skilful handling to do a good job. This made grinding an unpleasant event and one that was as a rule put off until the last minute. In order to use the knife as long as possible, it was obvious that unsound practices, such as filing out nicks and using soft fast-cutting whetstones, would be tolerated. These practices are to some extent in vogue, but are disappearing

and will probably be forgotten in the next generation. A straight, thin edge is required to do good cutting, and to accomplish this, it is necessary to grind often and only to whet enough to keep the edge smooth.

Grinding. All modern grinders are equipped with a stationary knife bar and a moving abrasive wheel, mounted on a carriage that travels back and forth on slides (see Fig. 18). There are some grinders built so that the knife travels, but most of the grinders in use today are built with the wheel mounted on a reciprocating carriage, especially grinders for long knives.

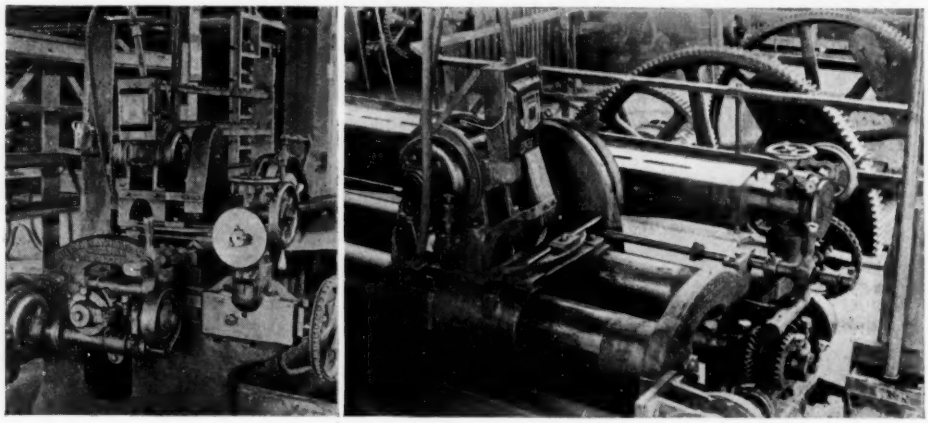


FIG. 18 VENEER-KNIFE GRINDER

(Left: End view of grinder, with abrasive wheel inside of guard. Right: Showing grinder-wheel carriage traveling on cylindrical ways. Knife bed with slotted boltholes in rear, with adjustments to control bevel.)

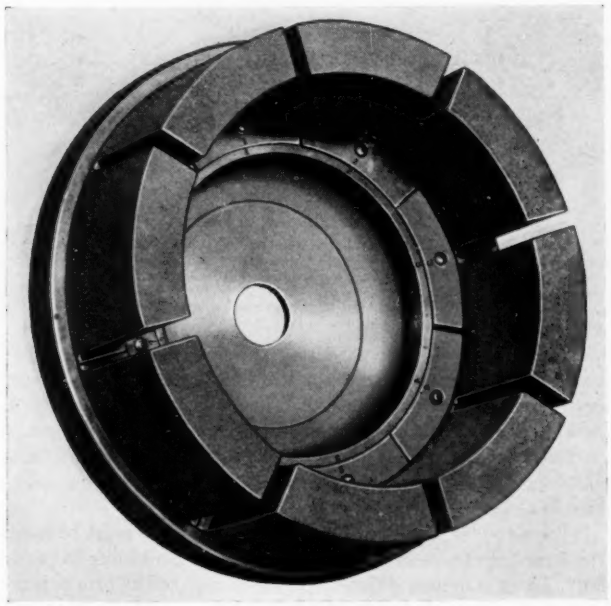


FIG. 19 SEGMENTAL GRINDING WHEEL

(Segments of abrasive wheel, mounted in holder, with approximately 1/2 in. clearance between blocks. Designed as a self-cleaning wheel with a minimum of glazing.)

The knife is bolted on the knife bar, the edges projecting about 1/2 in. above the bar. The bar is then brought up to the wheel and set so as to grind a 2-in. bevel on a rotary knife or a 2 1/2-in. bevel for a slicer knife. The wheel is then adjusted to grind not over 1/64 in. concave on a rotary knife and not over 1/100 in. concave on a slicer knife. The figures given are to be used on a standard knife 3/4 in. thick. It is not advisable to use more of a bevel or concave as there is danger of having the edge too thin.

The wheel used is usually a cylinder or cup wheel 14 in. to 16 in. in diameter and should run not over 650 rpm for a 16-in. wheel. The wheel should be kept burred off and not allowed to glaze. If a wheel is not cutting freely, there is danger of burning the knife.

A holder has recently been developed for segmental abrasive blocks, leaving 1/2 in. space between each segment (see Fig. 19). This is a great improvement over the solid wheel, for the reason that spaces between the segments allow the wheel to clear itself and thus prevent glazing.

Whetting. The wire edges should be taken off the knife before the knife is removed from the grinder. This is done by rubbing the whetstone lightly and squarely on the edge of the knife, removing all the wire edge down to the solid metal. Then the knife should be honed on the bevel side to an edge, being careful not to use the hone too much.

PRESSURE BAR

Most pressure bars (see Fig. 9-*E*) are made of mild steel, machined to fit in the cap, and drilled and tapped for the retaining and adjusting studs. The wear on a pressure bar is considerable, and it must be ground as often as the edge becomes rounded. Some cutters use a rounded bar on soft wood and a sharp bar on hard woods. It is the authors' belief that as sharp a bar as the wood will stand is better in all cases. In cutting soft wood, the angle of the bevel of the bar should be less acute to give the bar more bearing on the wood. This will prevent the bar from scraping the surface, equally as well as a more rounding edge, and the pressure will be much more sensitive, as it will be applied closer to the point of contact with the knife edge.

Tempered tool-steel pressure bars can be obtained today, at considerably higher cost. They require less frequent grinding and consequently are longer lived. It is a matter of opinion as to which bar is the most economical to use.

SETTING THE KNIFE AND PRESSURE BAR

Bolt the knife securely to the knife bed (the edge of the knife 1 1/2 in. or less above the knife bed). The best way to mount a knife is to adjust the setscrews on which the knife rests to about the proper height before putting the knife on the machine. Care should be used to have the bearing surface and the knife wiped clean so as to be sure that nothing interferes with having perfect contact of the knife with the knife bed.

Some of the older lathes were built with the back of the knife to the log and the bevel side to the knife bed, in which case the edge of the knife must be 2 in. above the knife bed.

The cap should now be placed in position, and the edge of the pressure bar and the edge of the knife should be in alignment vertically. This is done by means of the screws (see Fig. 9-*C*) on which the knife rests and the pressure-bar adjustments. The knife can be aligned to some degree by means of the setscrews, but short, low places will have to be taken out with the adjustments on the pressure bar. Some machines are built without a vertical adjustment in the pressure bar. In this case it is necessary to use shims to spring the bar up or down, as the case requires.

With the modern grinders and with proper care in handling the knife and pressure bar, the alignment of the knife and bar is a comparatively simple matter.

After the vertical alignment is accomplished, the horizontal alignment is necessary, which must be perfect and can be accomplished by the touch much closer than by any other means. After the knife is put on the machine and the pressure bar is adjusted, the knife bed should be leveled and the edge of the knife be brought to $^1/_{16}$ in. above the center of the spindles.

The same rules apply in setting the knife and pressure bar on a slicer as on a rotary with the exception of the height of knife edge above the knife bed. On slicers, the knife is carried higher than on a lathe, in some cases as much as $1^3/_4$ in.

There is no advantage in carrying a high knife, but it is important that the knife be afforded all the support possible.

Preparing the Wood for Cutting

There are few species of wood and not many conditions where logs can be cut into veneer cold—i.e., as they come from the stump without preparation. While it is admitted that cooking or steaming a log is a softening and toughening process that facilitates firm cutting, yet the processes of cooking or steaming have long been a controversial subject among veneer cutters, and the advocates of several different procedures seem to be successful, so far as veneer quality is concerned. The problem is one of heating the log through and making the fiber cuttable, without continuing the cooking to a point where the veneer fibers shred or separate. Some authorities advocate considering it a twofold problem of the temperature factor and the duration of preparation, believing this clarifies the issue. There are two principal methods:

1 Hot-water cooking vats, principally for hard woods
2 Steaming boxes, particularly for soft woods.

Hot-Water Cooking Vats. These are usually sub-floor tanks (generally covered with loose coverboards) of laminated wood or concrete, some 6 ft deep, as wide as lathe-log length, and from 20 to 40 ft long. Since duration of cooking is usually advantageous, these vats are generally in batteries, with conveyor chain overhead for removing the logs and transferring them to the lathe platforms. Live steam is discharged at one or more points below the water level, and often is under thermostatic control. Vats are exceedingly hazardous to workmen and need adequate guard rails and protective features.

Steaming Boxes. These are of wood or concrete, usually aboveground, where logs can be rolled in and removed by conveyor grabhooks. They are much easier to load and unload than are vats, and can be kept cleaner. The steaming process is usually more intense and shorter in time than cooking, so that fewer units are required, but the violence of rapid heating is hazardous on some species and rather ineffective on others. Doors to steam boxes, with in fact the entire structure, are subject to rapid deterioration from frequent and abrupt temperature and moisture changes.

General Preparatory Measures. The authors know of few veneer plants with an adequate supply of steaming boxes and cooking vats, and hence most operators are obliged to maneuver their scanty equipment and frequently to skimp on the best preparatory measures. The outline following is rather ideal, beyond full accomplishment in most plants, and will find many advocates, as well as some sincere objectors. Nevertheless, satisfactory work will result from adherence to the recommendations made. Heat should be applied slowly at first and increased toward the end of the cooking.

Poplar, basswood, and cottonwood, for utilitarian purposes, are often cut cold, if reasonably fresh from the stump. Soaking in log ponds is always advantageous. Gum is preferably cooked overnight at from 150 to 200 F. Coniferous woods, such as fir, pine, and spruce, are usually steamed 4 to 6 hours and sometimes longer. Oak, ash, and similar woods require boiling, usually from 12 hours up. Walnut, mahogany, and similar face woods cut best after cooking overnight, and sometimes longer, at temperatures from 125 to 200 F. Judgment should be used according to age and individual log characteristics. Too much heat is dangerous. Maple and birch should have similar treatment, but more rapidly to prevent discoloration, with temperatures up to 200 F. Stumps and crotches benefit usually by long soaking (several days) in warm (around 100 F) rather than hot water.

The important factors are to heat the log through and to find the happy medium between under-prepared or rough (sometimes fractured) cutting and over-preparation or fuzzy (shredded) cutting. The thicker the veneer to be cut, the higher the temperature required to reduce fracturing hazard. It is not at all unusual to remove a log from the flitch table to cool off or to recook it. Sometimes a flitch that cuts badly in one direction can be reversed and remounted so that it will cut satisfactorily. In general, long soaking at low temperatures, even in cold water in log ponds, is beneficial, and heat must always penetrate thoroughly but not too rapidly for the particular species or texture.

Operation

Operation, in the general sense of the term, is an individual problem in each veneer mill, and a procedure that might give the best results in one plant could be entirely inappropriate in another.

There are, however, a few fundamental rules that apply to veneer cutting in general. With the fit-up of the knife and pressure bar made as described, the cap should be raised to bring the edge of the pressure bar about $^1/_{32}$ in. above the edge of the knife; the cap should be pulled back so that a piece of dry veneer of the thickness to be cut can be slipped between the knife and bar. The lead should be flat; that is, the angle of the lead, as explained previously, should be at 0 deg.

With a flitch or log in the machine, it is ready to start cutting, with the principal adjustments in known positions. It is fairly certain that lead will need to be added, that the cap will have to be either raised or lowered, and that the pressure will have to be increased. After starting the machine and while taking off the first few sheets, adjustments should be made, such as putting in enough lead to hold the knife in the wood and adding pressure to preserve the texture of the wood (too much pressure will cause raised spots on the log). If the fit-up is not perfect, this can be relieved by pulling the pressure bar back at these points. If the cap is too high, adding pressure will not bring proper results. If the cap is too low, adding pressure will crowd the veneer too much in going between the knife and the bar, and cause a chatter, or washboard. This adjustment is very sensitive, and great care should be used not to change it too much. Since too much or too little lead will also cause chattering, it is well to get the lead set correctly before attempting to lower the cap.

On a lathe, the lead should be changed with the back screws so as not to raise or lower the height of the knife edge. The height of the cap is regulated with the front screws.

The back screws on the cap are used only to change the angle at which the pressure bar is presented to the log.

If changing the adjustments does not bring the expected results, it may be necessary to start all over. It is a good plan to mark the screws with chalk so that the amount the adjustments have been changed is always visible.

There are many other operating problems, subordinate to the fundamentals outlined, but their solution usually depends on the skill and experience of the operator and varies with the purpose for which the veneer is to be cut, the thickness of the sheet, the kind of wood, and the type of equipment available.

An idea of a typical lathe layout and crew may be secured from Fig. 20.

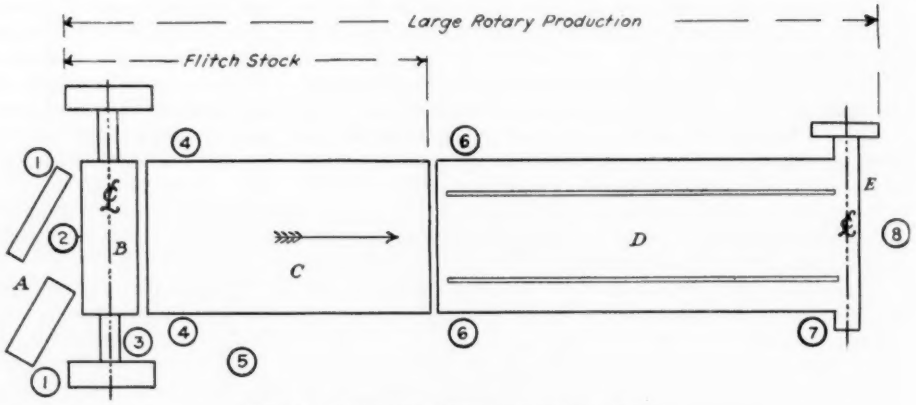


FIG. 20 STANDARD LATHE LAYOUT AND CREW

(*A*, log deck at floor level; *B*, rotary lathe; *C*, delivery table, sloping up to clipper chain; *D*, conveyor chain to clipper; *E*, clipper for trimming veneer. The standard crew, totaling 11 workmen, is as follows: (1) 2 log peelers, varies with lathes in service; (2) 1 log hanger; (3) 1 lathe operator and crew foreman; (4) 2 off bearers or breakers; (5) 1 waste collector; (6) 2 off bearers, for large production; (7) 1 clipper operator; (8) 1 clipper off bearer. A stay-log crew or rotary flitch stock usually eliminates the last four workmen.)

CONCLUSION

The dropping of sheets of beautiful veneer on the delivery table has a rare fascination for the artizan who can visualize the ultimate beauty of the wood in a handsome table or cabinet. The variations in color, shade, and figure are elusive and unending. Two trees never yet grew alike, and nature's artistry has never been equaled by man.

The modern American penchant for standardization has little place in the artistic value of veneer utilization. Beauty results from the intelligent use of color, grain, contrast, combination, and symmetry, and knife-cut veneer supplies a wonderful range for such artistic expression.

The full natural beauties of the forest products can be revealed only when the experience and artistry of the workmen combine the skilful flitching of the log with the intelligent cutting of each sheet of veneer from the flitch.

ACKNOWLEDGMENTS

The historic lathe and slicer illustrations, Figs. 1 and 14, were taken from "Knight's American Mechanical Dictionary." The photograph for Fig. 4 is from the archives of Sinker-Davis Company, Indianapolis, which has not built lathes since 1890; that for Fig. 5 was supplied by the Coe Manufacturing Company, of Painesville, Ohio; those for Figs. 6, 8, 12, 15, and 19 by the Capital Machine Company, Indianapolis.

Much credit is due to William Williams, of New York City, for his original idea for draw cutting on a slicer; to William Schwegman, of Indianapolis, for originating the pressure-bar adjustments that are in use today; to Louis Koss, of the Capital Machine Company, Indianapolis, for visualizing the need for sturdy construction and consistently working to that end; to H. H. Coe, of Painesville, for recognizing the advantage of placing the feed screws on the knife line, as well as for other original developments; and to I. E. Merritt, of the Merritt Engineering and Sales Company, Lockport, N. Y., for many of the refinements of modern lathe construction.

This paper has been prepared with the assistance and cooperation of Thomas D. Perry, the author of several plywood papers for A.S.M.E. Transactions.

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Discussion

ARTHUR ANDERSON, Chautauqua Plywood Co., Jamestown: Referring to what Mr. Roberts said in regard to the necessity of having a perfect adjustment of the pressure bar and knife, I have seen poor cutting done on curly maple. He said that as long as these adjustments are right a machine should cut good veneer. But is it not true that the logs should first be properly conditioned for cutting—and what is the proper way?

MR. ROBERTS: To get the best results in cutting veneer the wood must be in the proper condition. An operation preliminary to the cutting operation is in most cases necessary to obtain this condition. There are two common methods of conditioning logs: First, by soaking them in water of the proper temperature; second, by heating them with a direct application of low-pressure steam. In the case of hard maple the logs should be soaked for about 24 hours in water at a temperature of about 200 F. If too much heat is applied, the veneer will show heat marks and the log should be removed from the machine and allowed to cool before further cutting is attempted. A very keen, smooth-edged knife will help prevent heat marks. If poor cutting other than heat marks is experienced, the cause will probably be found in adjustment of the knife and pressure bar.

PAUL SETTER, Setter Brothers, Cattaraugus, N. Y.: Mr. Roberts spoke of the knife bearing against the log; at what angle is it set?

MR. ROBERTS: The angle referred to is the lead, as defined under the heading of "Knife-Bed Adjustments." It is best to start cutting with this angle at 0 deg and to add just enough lead to hold the knife in the wood and maintain even thickness. It is necessary to carry more lead in cutting thin veneers.

CHAIRMAN: Speaking of overcooking, is there some easy method of telling how to cook a log?

ANSWER: Our practice is not to get the wood too hot. You can, by additional pressure, get good results with the wood a little cool.

AUTHORS' CLOSURE

The questions reveal a decided interest in log preparation, a process that is admittedly far from standardized. The reason is obvious, since no set rules have ever been found to bring uniform results. Trees of the same species will require different treatment according to age, texture, growing conditions, and the season in which they are cut. Hence the cooking operation must in the last analysis be accomplished according to the results of observation and experience.

WDI-55-5

The Electric Molder and Its Efficiency

By ANTON VONNEGUT,[1] INDIANAPOLIS, IND.

The great variety of molder work, involving different methods of operation and various types of cutter-head equipment to meet requirements in all branches of the woodworking industry, accounting for a wide range of expectations regarding molder performances, has caused the development of diversified points of view for appraising molder efficiency. This paper attempts to analyze basic molder design requirements, as determined by fundamental molder functions and principles of molder cutting actions, with the hope of developing a more general understanding of reasons and requirements for electric molder efficiency.

ONE might define a woodworking molder as "a machine with a series of fixed and adjustable devices to guide lumber into relation with plane or formed knives of rotating cutter heads that are removably attached to a multiple number of spindles having lateral and logitudinal adjustment for the purposes of surfacing or forming and dimensioning various sides of lumber simultaneously, as it is pushed through the machine by means of its power-driven feeding mechanism." The qualification of "multiple number of spindles" excludes single-spindle machines. It includes two-head molders and says nothing about the relation of one head to another. Therefore it applies alike to the European type of molder, in which the first cut is made with the bottom head, as well as to the standard American spindle arrangement, in which the first cut is generally made with the top head. The definition also covers "flooring machines" and "planer-matchers," because they are single-purpose and semi-single-purpose molders. Some of them are made with infeeding and outfeeding mechanisms, and they are generally much heavier than standard molders. They are less flexible in the range and variety of their adjustments than the more widely used type of general-purpose molder.

The range and flexibility of adjustments and the accessibility and convenience for making and locking adjustments are the distinguishing characteristics of general-purpose molders. They must be designed to handle a wide range of work from smallest patterns of fragile stock up to dimensions of their maximum capacity. They are used to cut all varieties and conditions of lumber and in every branch of the woodworking industry. Sometimes molder work is a preliminary operation to other machine work, sometimes a finishing operation following preliminary work on other machines, and sometimes the molder is called upon to make a finished pattern from rough lumber. In one instance the main objective may be speed and maximum production without primary consideration of quality, and in another it may be that smooth surfaces and accuracy of pattern are considered far more essential than the amount of work produced. More often the requirements include a good balance between quality and quantity of molder work.

In spite of all the need for flexible adaptability, some molder service may call for long runs of a limited number of patterns. On the other hand, many short runs and great variety of patterns often create the necessity of using more time for making set-up changes than for feeding the stock through the molder. Relatively satisfactory efficiency for the extreme services are available with the same machine on account of interchangeability of a variety of special and universal types of molder cutter heads. The utility of heads is also augmented by attachments, and this has developed two distinct systems of molder operation to supply the different requirements of different molder services.

Though there are well-defined extremes of requirements and services in different woodworking plants, there are also mixtures of those conditions and requirements in most plants; and there is generally a slightly different blending in the mixtures of requirements and expectations associated with each molder installation. There is no definite standard to measure quality and quantity of molder work. The maximum output of a detail mill might seem ridiculous by comparison with the minimum output of a production mill, and a standard of excellent quality of work in one plant would not pass the most lenient inspection of another. Such differences do not only exist as between different branches of the woodworking industry, but also within the same field of manufacturing. For example, there may be more differences of opinion regarding the proper way of using a molder and the results that ought to be expected as viewed by two manufacturers of furniture than between a mill man and the manager of a toy factory. That condition helps to point out the danger of misunderstandings, for lack of an established point of view, in any discussion of electric molders. In fact, there is even very little agreement on the subject as between different manufacturers of identical types of molders.

A Common Point of View of Efficiency

There can therefore be no doubt of the desirability for first finding some basic or common point of view before proceeding with any discussion of electric molder efficiency. Each of us may be interested in a different detail or a different phase of molder design, operation, or efficiency, but whatever that interest may happen to be, it has some relation to the three fundamental molder functions of feeding, guiding, and cutting the stock. Fortunately, that applies absolutely alike to all styles, types, and makes of molders—belt-driven as well as electric—and it does not take any stretch of imagination to claim that the functions of feeding and guiding are a means to an end. That end, namely, cutting, therefore suggests itself as the most important function. It is the basic factor of all molder efficiency.

Of course, it is needless to say that an inefficient feeding mechanism or an inadequate guiding device destroys the value of satisfactory requirements for cutting and that feeding and guiding must be considered in determining molder efficiency under all conditions, but it is equally needless to defend the greater importance of the cutting requirements. Incidentally, it should be said that there have been some most interesting electric-molder feed-works developments during the last few years. Nearly all electric molders are now available with traveling feed beds in addition to roll feeds. One make of molder

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[1] Vonnegut Molder Corporation. Anton Vonnegut was born in Indianapolis, Ind. He is a graduate of Manual Training High School, 1901. He gained the M.E. degree at Cornell, 1905. Was engaged in machine shop and drafting work, 1905–1908. He was with the Vonnegut Hardware Company, Indianapolis, 1908–1910; president, Vonnegut Machinery Company, 1908–1930; president, Vonnegut Molder Corporation, since 1930.

Presented at the meeting of the Wood Industries Division of The American Society of Mechanical Engineers, Jamestown, N. Y., November 15 and 16, 1932.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

offers the three choices of roll feed, chain feed, and rolling chain feed.

There have been many improvements in guiding devices, particularly by way of increasing conveniences and speed for making quick set-up changes. It is also a temptation to mention materials and other refinements of electric-molder construction, but besides the limitations of time for those details, there is the very significant fact that none of those developments was a factor in the greater efficiency of the first electric molders by comparison with belt-driven machines.

The electric molder is therefore being discussed from the broad point of view that it is the same as a belt-driven molder except for a new means of applying spindle power and transmitting that power into the cut. That viewpoint is very well supported by the fact that several manufacturers are offering electric and belt-driven molders under the same catalog numbers, with no difference of specifications except as pertains to spindle drives. Furthermore, the relatively greater importance of power and spindle details is more or less self-evident as one becomes familiar with early experiences of electric-molder development.

Incentive for Designing the First Electric Molder

Paradoxical as it may seem, the incentive for designing the first motor-driven molder was the expectation of economical power consumption of a machine for an industry that had been notoriously indifferent to power economies, because it developed power from wood waste and refuse. Many woodworkers even took the attitude that power and the steam used for dry kilns, glue rooms, etc., cost nothing, because some form of burner and the expense of its operation would have to be maintained for refuse disposal. It must, however, be remembered that there was a rapid expansion of American industries between 1915 and 1920. Dry-kiln and power requirements soon exceeded boiler and engine capacities of many woodworking plants, and since power had generally been a matter of secondary consideration, those shortages were not often anticipated, and they frequently developed dangerously overloaded power plants before the facts were recognized. Such conditions naturally forced quick action in the purchase of motors to drive old line-shaft installations from public-utility power connections. Practically nothing had been done to correct the inefficiency of the old transmission facilities, and there were very unpleasant reactions against the monthly power bills, because the industry had learned to think of power as a low-priced by-product.

During the investigations resulting from that condition, attention was called to the larger power consumption of molders compared with other woodworking machines. Individual motors, coupled to countershafts, gave little relief, because of the many bearings and belts in connection with a long countershaft, a feed works, and four spindles of each molder. That data encouraged R. D. Eaglesfield, of the Vonnegut Machinery Company, Indianapolis, to undertake the design of the first electric molder early in 1916. It was installed in the plant of the Prest-O-Lite Company, near Indianapolis, with the gratifying result that it effected over 40 per cent saving of the power required by a new belt-driven molder of similar capacity, cutting the same amount of the same kind of lumber, at the same rate of feed. Considering the objective of the venture, the experimental machine was therefore a complete success.

Early Electric Molder Experiences

Subsequent tests have shown from about 40 to 60 per cent power savings for ball-bearing electric molders in comparison with babbitted-bearing belt-driven machines, and such evidence seems to justify the generality that the modern electric molder uses only about one-half the power required by old-style belted machines, while doing the same amount of work. The facts suggested that 25 to 30 hp, divided among four spindles and a feed works, would be more than sufficient for a fairly large-sized universal type of electric molder, because belt-driven molders were being driven by motors coupled to countershafts ranging from 10 to 25 hp, with only an occasional 50-hp installation for the heaviest service. Furthermore, the selection of motor sizes had been shown to be satisfactory by performances of twenty or more electric molders, when there came the startling report of insufficient spindle power in connection with one of those machines.

To understand that anomaly, one must remember the qualifying phrase, "at the same rate of feed," in the foregoing statements of power comparisons. Eliminating consideration of initial friction, windage, and electrical losses, molder power consumption increases in direct proportion to the rate of feed, and in the instance referred to, the purchaser was attempting to feed the electric molder approximately four times as fast as he had been feeding belted machines to do the same class of work. Thus the electric molder was forced to draw about four times as much power as would have been required if it has been fed at the same rate of feed at which the work had been done with belted machines and presumably about twice as much power as had been required to do the work with the old molders at one-fourth the rate of feed.

The work was being done in an automobile-body plant, and all of the patterns were to be covered in the completed product. The work had to be true to pattern, but there were no definite restrictions on the knife-mark spacings so long as the edges of the work were free from splinters and the surfaces free of torn places. Both types of machines were operated at spindle speeds of about 3600 rpm, because that was in the days before high-frequency molder operation. The work coming from the faster feed of the electric molder therefore had much wider knife-mark spacing than the work from the old machines, but it was acceptable—first, because there was more regularity in depth and spacing of those knife marks and, second, because there were fewer splintered and torn places in spite of the faster feed. The experience was the first conclusive demonstration that electric-molder efficiency included possibilities of increased production of greater proportions than the power economy for which the machine had been designed originally.

It had also been demonstrated, during early stages of electric-molder development, that the greater regularity in depth and spacing of knife marks produced a much better quality of work than was being obtained from belt-driven machines, and the electric molder therefore offered the threefold advantage of more economical operation, greater production, and better quality of work. All three divisions of efficiency had become a matter of fairly common knowledge even before the days of high-frequency molder operation and were evidently developed by smoother and firmer spindle rotation. Possibly that can be illustrated by comparison with another cutting action. We all know the contrast between stroking a pocket-knife blade or any form of edged hand tool firmly and quickly through a piece of wood and of shifting or twisting the blade hesitatingly before completing the cut. The one produces a smooth finish; the other a torn or splintered surface. It seems reasonable to assume that similar results would obtain with powered cutting tools and that the least lateral or longitudinal deviation in the line of cutter travel would account for torn and roughed molder cuts. Furthermore, even when the character of the wood is so favorable or the disturbed motion of the cutter is so slight as not to cause a torn surface, it is evident that the disturbance will affect the perfect regularity in the depth of each successive cut so as to destroy the effect of a perfectly finished surface.

Basic Reason for Electric-Molder Efficiency

The more nearly perfect spindle rotation of the electric molder was due to the three facts: (1) that the spindles were mounted in closely fitting ball bearings in contrast to spindles that floated on a film of oil, as is the case with friction bearings; (2) that ball bearings also restricted spindle end play; (3) that those spindles were being turned by balanced magnetic forces of the direct motor drive without being subjected to the pull and pound of high-speed belts. Those facts were evident, but they involved more than mere substitution. For example, the original electric molder incorporated substantially the same type of removable outboard spindle-bearing design as had been used with babbitted-bearing belted molders, but a friction bearing fits loosely and the spindle turns within the bearing, whereas the inside race of a ball bearing must be fitted to the spindle and turn with the spindle. With the old design there was the choice of fitting the ball race too closely for convenient removal and replacement or too loosely to provide firm spindle support. In one instance an operator gave evidence of the annoyance of replacing closely fitted bearings by deliberately filing down the spindle end diameters, thereby destroying the main advantage of ball-bearing spindles. At best, however, natural wear from bearing removal and replacement developed a sloppy fit that accounted for poor top and bottom head work after a few years, and in some cases after only a few months of service.

Subsequently, the problem was augmented by greater centrifugal cutter-head forces of high-frequency operation, but the fundamental requirements remained the same, and there followed the conclusion that the superiority of one electric molder over another electric molder depends primarily on its more nearly perfect cutter-head rotation coming from better spindle design and better means for applying the power to the spindles, just as the first electric molders had proved their superiority on account of a smoother spindle rotation than had been obtained from "floating" spindles that were subject to the pulling and pounding effect of heavy belts. Maximum molder efficiency depends upon the best solution of that problem.

Molder spindle design must provide for firm rigidity of the supporting parts that supply longitudinal, lateral, and angular spindle adjustments in combination with convenient accessibility for making and locking those adjustments. Then, besides answers to engineering questions pertaining to bearings, bearing housing, spindle shaft, motor, motor mounting, lubrication, ventilation, and the proper relation of all those spindle parts to one another, the problem also involved establishing standards and manufacturing methods to assist in obtaining the greatest possible accuracy of dimensions. An interesting sidelight on that point is that ball-bearing manufacturers hesitated to make any recommendations of tolerance allowances for fitting bearings in the first electric molder, because they knew nothing about molders and molder service. Those tolerances had to be worked out by molder manufacturers, and they have gradually been restricted, until present-day standards call for less than one-tenth of the allowances specified for the first electric molders. The desired results must provide the closest possible fittings in combination with uniform interchangeability of all the spindle parts.

It would be most interesting to go into the details that have been worked out in answer to those problems of design in the quest for maximum molder efficiency, because they have been the subject of so many claims and counterclaims among representatives of half a dozen molder manufacturers, but space forbids. The results derived from those designs must be examined instead of the details of the designs, and it is hoped that the foregoing generalities have served their purpose of emphasizing the importance of all that pertains to spindles in considering the following information regarding those results.

Molder Operating Methods

In spite of what has been said about the great variety of molder uses and the resulting diversity of opinions and points of view, there seems to be the prevailing interest, indicated by the frequency of the question, "How fast will it feed?" Theoretically, the correct answer would be, "As fast as the stock and finished work can be brought to and taken from the machine." However, the generality must be qualified by conditions. For example, there is greater possibility of crushing or breaking small patterns of fragile woods under the greater strains of higher rates of feed. That is only one of many other qualifications that, unfortunately, must also be omitted for lack of time; but beyond these, the rate of feed depends primarily upon the quality of work desired, and it is influenced by the method or system under which the molder is to be operated.

Two molder operating methods are determined by the two practices of "jointing" and "not jointing" the knives, or "multiple-knife finish" and "one-knife finish." The word "finish" is used because the term "one-knife" might suggest a single knife, and there must always be a balancing knife. There may be two, four, six, or even more knives per head, and all knives may be cutting, but if one knife projects only a fraction of a thousandth of an inch beyond the others, it wipes out the cutting marks made by the other knives, and the finished work shows only one knife mark for each cutter-head revolution, accounting for the term "one-knife finish." Jointed knives are attached to heads in substantially the same way as unjointed knives, but more care and time are taken to obtain approximately equal projection of all the knives. Then a firmly guided whetstone or emery stone is applied lightly to the knife edges, while the spindle is in operation, to equalize the knives, so that each edge projects exactly the same as the others, thereby effecting a distinctly separate cut and forming an individual mark for each cut; for example, four knife marks for each revolution of a four-knife head. That is comparatively simple for surfacing operations, because the jointing stone is drawn and guided along the length of a straight bar positioned parallel to the spindle. The same principle is sometimes attempted for formed work by guiding the stone along a jointer templet. More commonly a stone is conformed to the shape of the knives and then guided straight into the knives at right angles to the spindle. In either case the jointing of formed knives involves the preliminary time of making templets or forming the stones.

Theoretically, there is far greater production with jointed knives. At the same spindle speed, two, four, and six jointed knives will effect the same knife mark spacing as the one-knife-finish system, but at two, four, and six times the rate of feed, respectively. Practically, however, the theory must be tempered by the facts that the attainment usually involves more time for setting the knives besides the jointing time and that the amount of time is fairly well dependent upon the individual skill of each operator; that even a thoroughly skilled operator does not always joint the knives with uniform accuracy; that any slight difference of knife tempers may cause unequal wear and destroy the joint; that the gritty surface of an occasional piece of undersized stock cuts the knife edges and destroys the joint; and that jointing forms a "heel" or flat surface behind the knife edge, of which all parts are equidistant from the spindle center and which must therefore slide against and tend to glaze the surface fiber of the finished work.

The last point would suggest that since the object of jointing is knife-edge equalization it means that the process resolves itself into generating different widths of heels on different

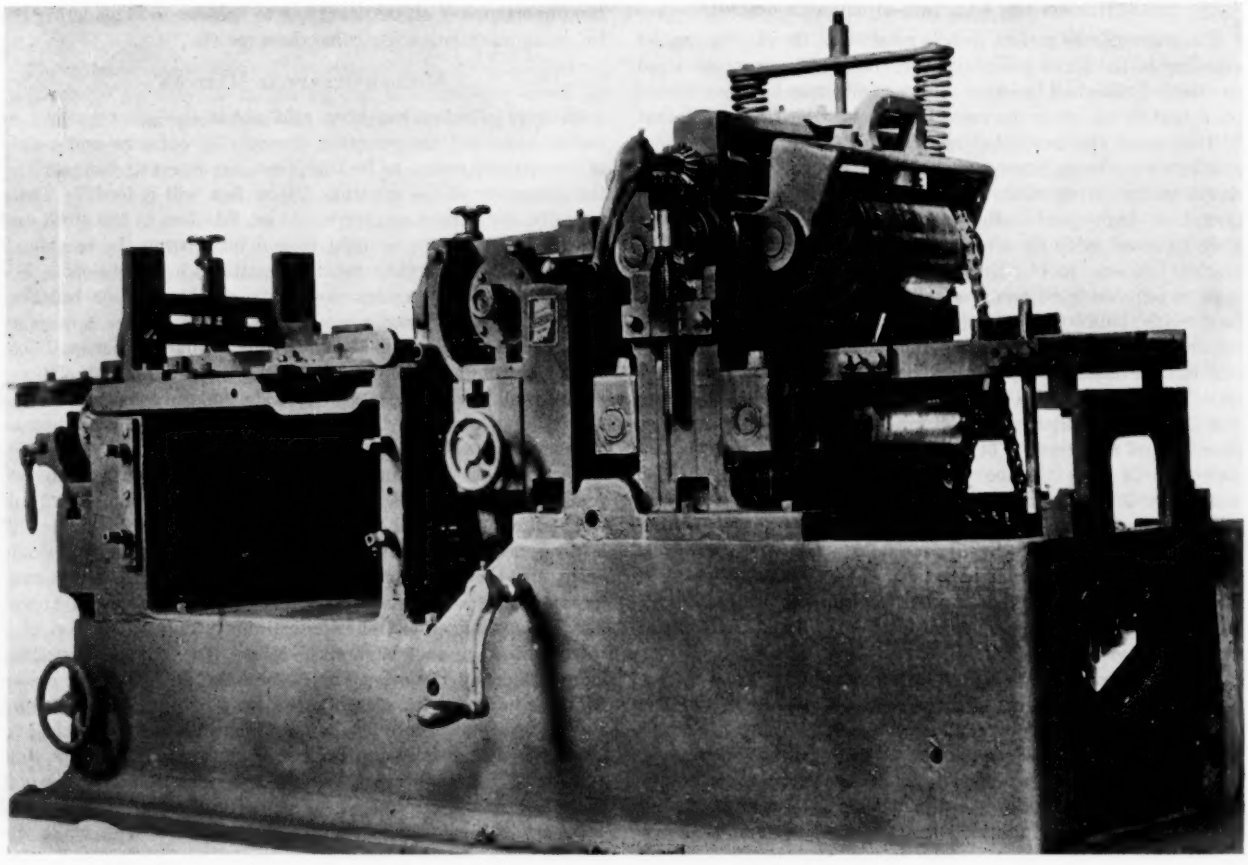


FIG. 1 FRONT VIEW OF THE FIRST ELECTRIC MOLDER

knives, depending upon their relative projection, and that the finished surface is therefore a series of slightly varying widths of heel marks. Possibly that sounds somewhat of the nature of hairsplitting, but the theory is supported by practice, because the desire for the very best quality of jointed work is fulfilled by generating the heels and then applying a grinder to the knife bevel to remove the excessive heels, without touching the edges. It takes time at best and is prohibitive for any except the most skilled and experienced molder operators. The system is commonly referred to as "fast-feed molder operation."

HIGH-FREQUENCY OPERATION

The feeding rates of one-knife finish are obviously slower for the same spindle speeds, but the standard 3600 rpm of earlier electric molders, operated from 60-cycle current, has been raised to speeds of from 6000 to 9000 rpm with 100- to 150-cycle power circuits developed through frequency changers from the primary lines or generated by alternators from almost any source of power. Assuming an average of 7200 rpm, the increased rates of feeds available with earlier electric molders have therefore been doubled. Higher spindle speeds, causing greater knife velocity, also account for cleaner cuts, and when the one-knife finish is used, the cuts are made with keen-edged knives that leave no heel mark. That system, in combination with the desired quality of spindle rotation, therefore makes each cut an exact reproduction of every other cut along that surface. The character of finish on all sides of the pattern is also similar, and the method offers much greater possibilities of maintaining a uniform standard of work at all times than a method that depends upon similarity of cuts made by different knives and upon the vigilance of the operator to maintain that similarity.

Besides the most uniform quality of work, the combination of identical repetitions of a keen-edged cut at high velocity, as occurs with one-knife-finish high-frequency electric-molder operation, self-evidently permits a higher percentage of running time, because it eliminates all lost time for jointing and preparation for jointing, and one is led to wonder whether it would not be better for the average woodworker to ask, "How much *good* work, according to my particular standards of quality, will my molder produce per hour, per day, and per month?" rather than, "How fast will it feed?"

CONFLICTING OPINIONS AND RECOMMENDATIONS

There is common consent to the opinion that no interior trim contract mill, small furniture factory, nor in fact any woodworking plant that intends to make a reasonable variety of patterns with one or a limited number of molders, can afford the time required for jointing and that high-frequency operation is to be preferred. Even in automobile body plants, where maximum output is paramount and where some machines, in batteries of a half-dozen to more than a dozen molders, could be kept busy on a single pattern or two, all molder work is being done with one-knife-finish high-frequency operation. Yet enthusiasts for the jointed-knife system are inclined to foster it with the argument that it is the means for faster feeds without the necessity of additional investment in a frequency changer (generally about 20 per cent of the molder price). In some instances that thought is even carried to the detail mill man by suggesting that high spindle speeds are dangerous with long knife projections; that all deep cuts must be made with spindles

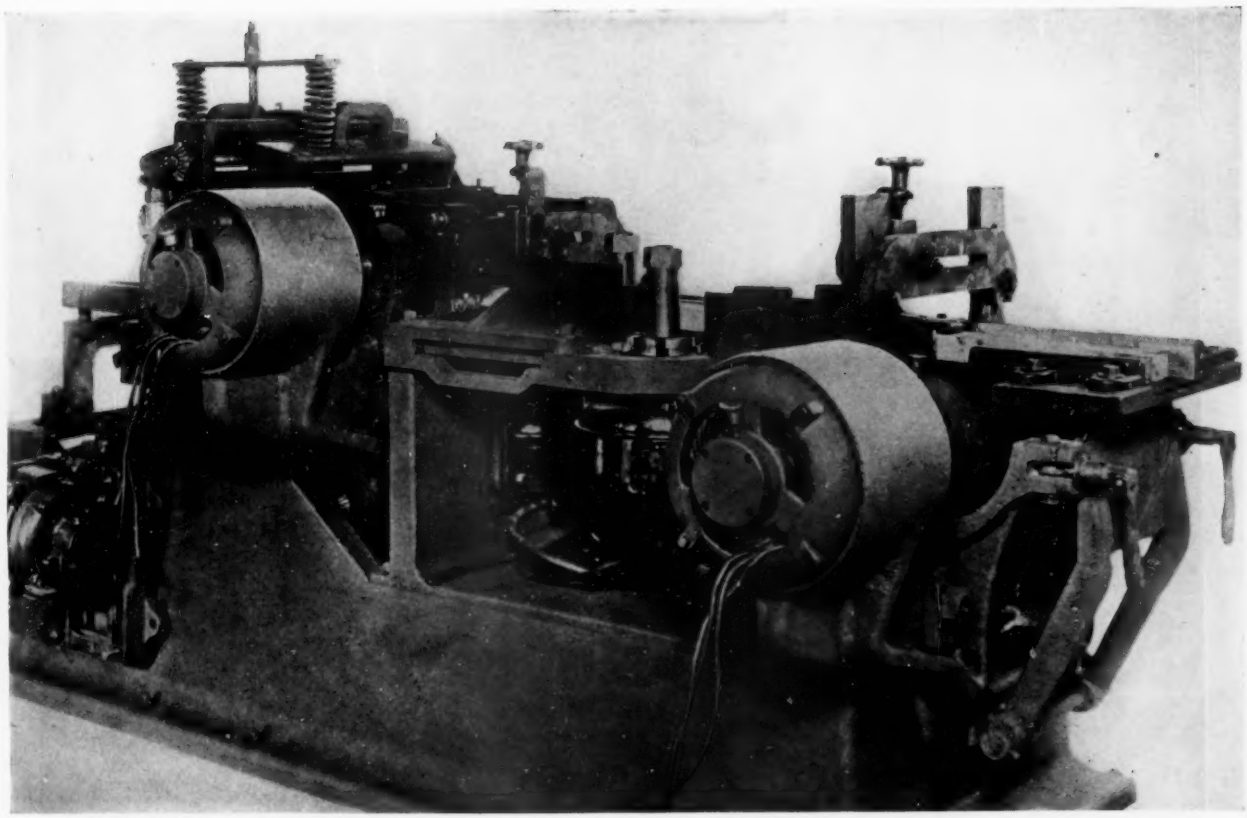


FIG. 2 REAR VIEW OF THE FIRST ELECTRIC MOLDER

turning at 3600 rpm; and that, since high spindle speed cannot be used at all times, the price of the frequency changer may be eliminated. The sponsors of such recommendations generally also maintain that knives cannot be jointed at 6000 and 7200 rpm.

There are intermediate opinions, but other molder manufacturers insist that maximum efficiency can be obtained only with high-frequency operation, because it is the means for making keen-edged knife cuts at from 50 to over 100 per cent faster rates of feed; that there is no more danger of throwing and breaking knives, with the smoother rotation of present-day cutter-head equipment, than existed at slower spindle speeds with inferior equipment of a few years ago; that speed enthusiasts can obtain maximum rates of feed by jointing knives with spindles revolving at 6000 to 7200 rpm; and that those expectations are being fulfilled at rates of feed approximating 200 fpm in both hard and soft wood operations. They do not overlook the fact that higher spindle speeds tend to accentuate difficulties of attaining and maintaining uniformity of multiple-knife finish, requiring the service of the more skilled operators, but claim that the success of that operation, like other phases of molder efficiency, is primarily dependent upon rigidity of spindle support and smoothness of spindle rotation.

It would therefore seem that conflicting recommendations regarding best operating methods for obtaining maximum efficiency resolve themselves back to the controversies about engineering problems and manufacturing methods relating to motor applications and spindle designs for obtaining the smoothest cutter-head rotation.

ILLUSTRATIONS OF SAMPLE WORK

Visibility of molder-knife marks depends partly upon the wood on which they are made, but mainly upon the angle at which the wood is held in relation to the light and the eyes but it is impossible to reproduce the knife marks at those angles photographically without showing distorted foreshortening, which would destroy the effect of the true relations of the marks. The accompanying reproductions were therefore photographed at right angles to the wood surface. All marks are therefore not distinctly visible, but since the photographs were taken under exactly the same conditions, they serve their purpose of showing comparative effects. Figs. 3 and 4 are samples of gum to show the similarity of cutting effect on a hard wood with the redwood of the other samples, which was used because it happened to be conveniently available when the work was ordered. Fig. 3 was made with a strong sidelight to show the curved character of the individual cuts, which is not noticeable with closer knife-mark spacing. All other photos were taken with the light coming from the vicinity of the camera.

It should be understood that some of the samples are not intended to illustrate results of normal practices. The conditions are deliberately exaggerated for the purpose of showing effects that are not ordinarily visible, but that nevertheless generally exist in some degree. For example, Figs. 3, 4, and 5 show one-knife finish, 7200 rpm, 190 fpm, with greater knife-mark spacing than would be desired for any practical purposes, but the uniformity of depth, spacing, straightness, and parallelism of clean cuts, with untorn surfaces, are a good means of demonstrating the effect of smooth spindle rotation, which accounts for perfect work with closer spacing of the same character of cuts, at slower rates of feeds.

One heel of four knives that produced the multiple-knife finish of Fig. 6 was deliberately emphasized to illustrate the effect of heel marks that always exist, though seldom so noticeably. In fact, that mark happens to show up more plainly

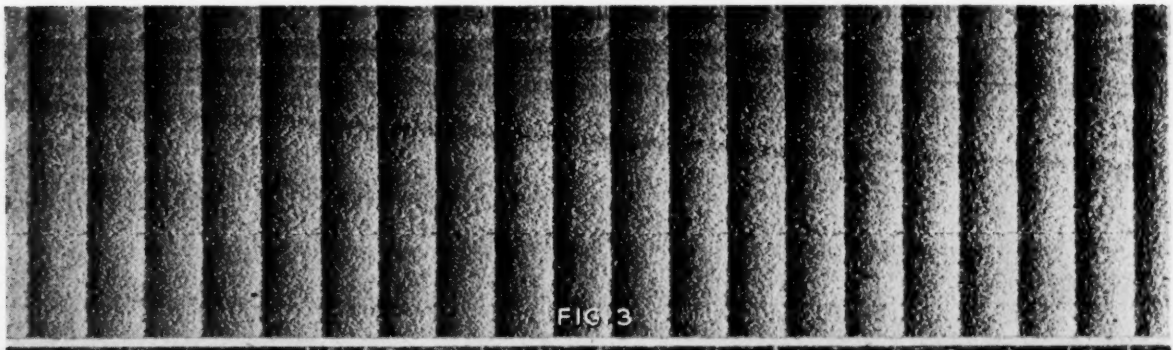


FIG. 3

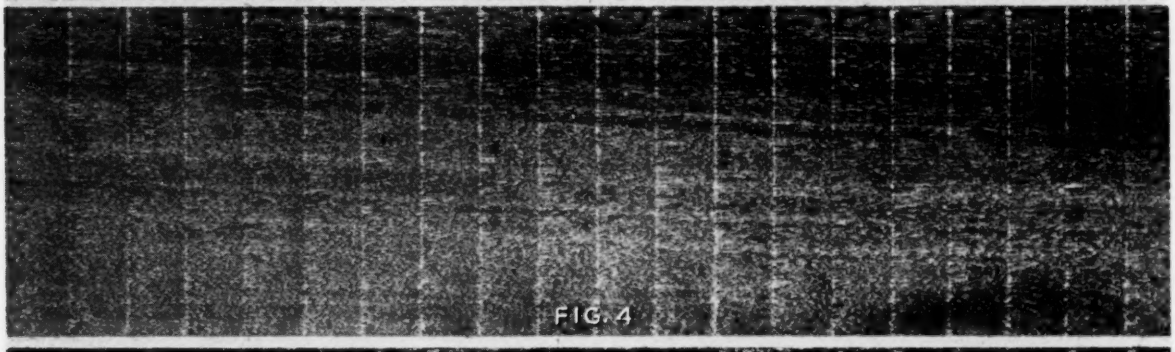


FIG. 4

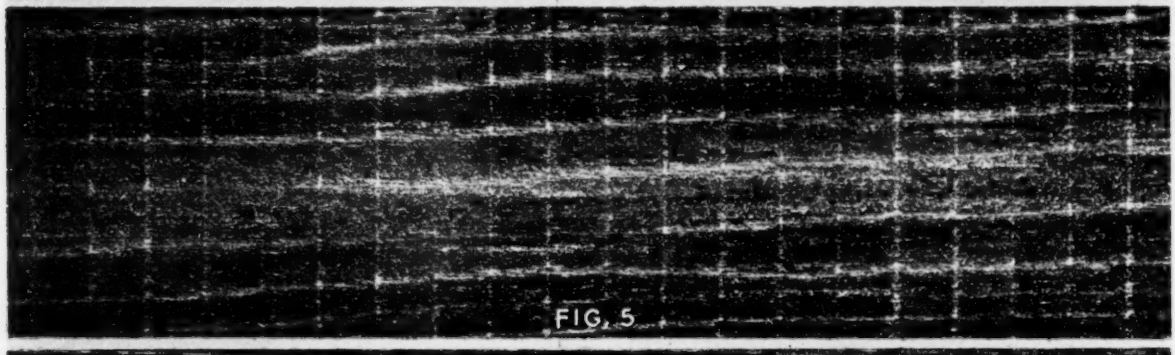


FIG. 5

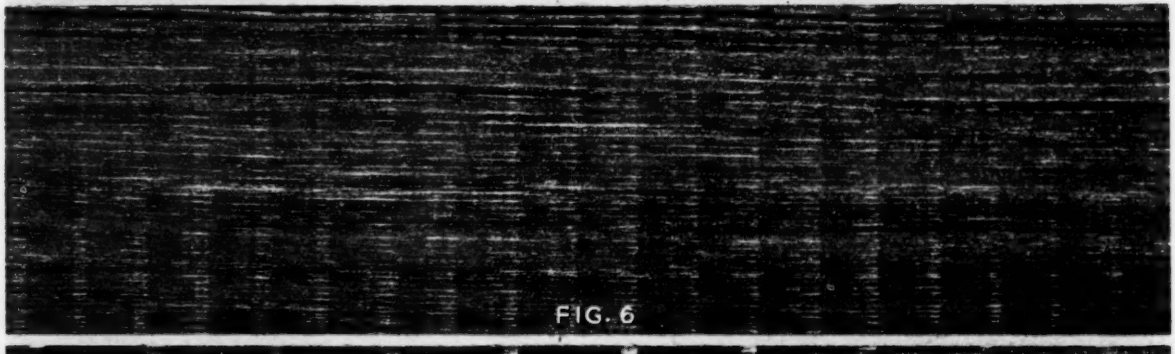


FIG. 6

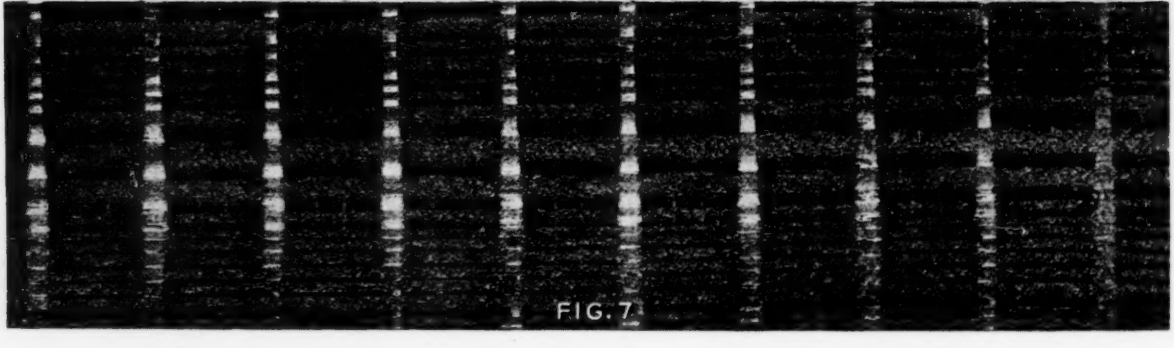
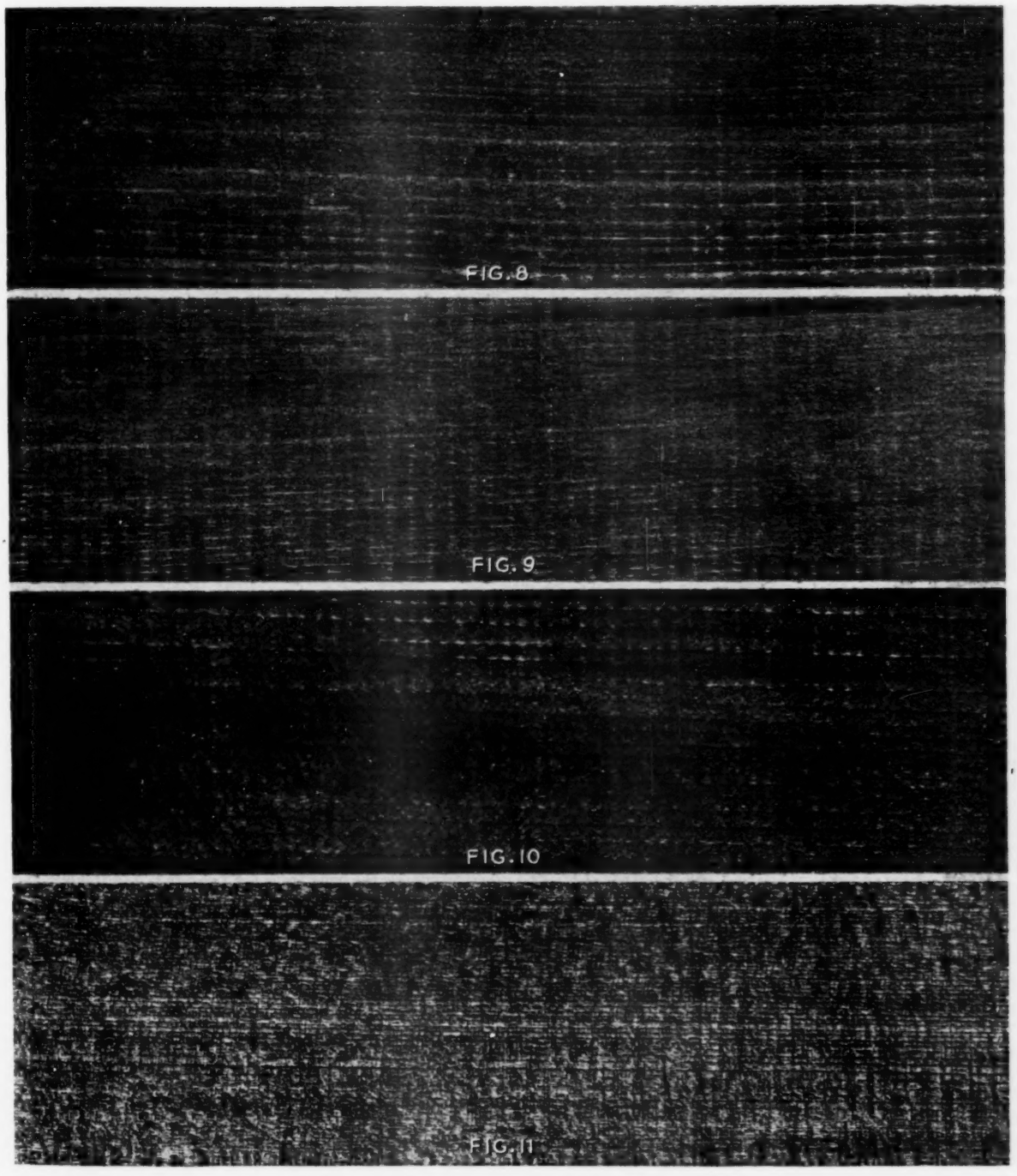


FIG. 7



FIGS. 3 TO 11 (*See preceding page and this page*)

FIG. 3 GUM, ONE-KNIFE FINISH, 7200 RPM, 190 FPM, WITH HORIZONTAL ILLUMINATION. FIG. 4 SAME AS FIG. 3, GUM, 7200 RPM, 190 FPM, WITH NORMAL LIGHT. FIG. 5 REDWOOD, ONE-KNIFE FINISH, 7200 RPM, 190 FPM. FIG. 6 REDWOOD, FOUR KNIVES JOINTED WITH ONE EXTRA-HEAVY HEEL, 3600 RPM, 95 FPM. FIG. 7 SAME AS FIG. 6, BUT FED AT 190 FPM. FIG. 8 REDWOOD, FOUR PROPERLY JOINTED KNIVES, 3600 RPM, 190 FPM. FIG. 9 REDWOOD, FOUR PROPERLY JOINTED KNIVES, 3600 RPM, 140 FPM. FIG. 10 REDWOOD, SAME KNIVES AS FIGS. 8 AND 9, BUT AT 6000 RPM, 190 FPM. FIG. 11 REDWOOD, SAME AS FIG. 10, 6000 RPM, 190 FPM, QUARTER-SAWN

in the reproduction than was expected by the appearance of the original sample, and the work shown in Fig. 7 was immediately made under exactly the same conditions, but at twice the rate of feed, before realizing that the glazed heel mark would show up as plainly as it does in Fig. 6.

Fig. 8 shows work that was done at the same spindle speed and same rate of feed as Fig. 7 and with the same four knives, but with knives properly jointed. If the intermediate knife marks of Fig. 7 were as plain in the halftone as they were in the sample work, it would be seen that the knife spacing of Figs. 7 and 8 are equal, despite the great differences of the two effects.

Fig. 9 shows multiple-knife finish, 3600 rpm, like Fig. 8,

but at 140 fpm instead of 190. Work shown in Figs. 10 and 11 was also fed at 190 fpm and with the same jointed knives as shown in the two preceding figures, but with spindles turning at 6000 instead of at 3600 rpm. The contrast of Figs. 10 and 11, with no knife marks showing in the latter, is because the direction of grain in the quarter-sawn board of Fig. 11 favored the finished appearance of that work. The contrast is an excellent illustration of the many variables that enter into the difficulties of establishing definite means for measuring molder efficiency.

Other Considerations for Efficiency

After all, however, the final analysis of efficiency must include investigations that begin ahead of and end beyond the molder. It involves questions of how much work must be done before the stock goes to the molder and of how much sanding and how many other subsequent operations will have to be required, with consideration of losses due to inspection rejections. There is no intention of considering those phases of molder efficiency, and the following three examples are offered merely by way of illustrating the suggestion that molder efficiency involves even more than economy of power and quality and quantity of work.



Fig. 12 Seven-Sectioned Desk Seat

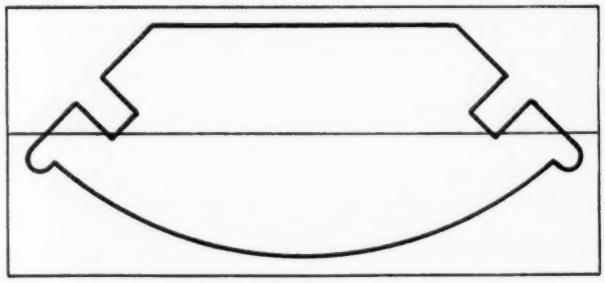


Fig. 13 Corner Cabinet Post

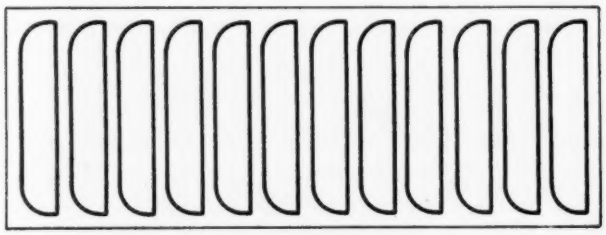


Fig. 14 Thirteen Roof Slats

Fig. 12 is a school desk seat in which the tongue, grooves, and slightly angular edges of seven pieces of stock are matched and glued to make the pattern. Before the days of electric molders, the seats required several operations for surfacing and thicknessing the top and bottom sides of each piece of stock, then glue jointing each edge separately, and running the glued-up seat pattern through a special machine having a formed cutter and formed feeding rolls before attempting the final sanding. With an electric molder it is possible to surface and thickness the top and bottom sides and glue-joint both edges of each piece so accurately in one operation that the pattern is ready for sanding as soon as it comes from the glue carrier. Of course, there is a substantial reduction of machining operations to account for considerable saving, but a far greater earning is effected by eliminating the finish-forming operation, because that accounted for much breakage, with resultant loss of labor, material, and salvaging expenses. The molder method also produces a more uniformly satisfactory product.

Fig. 13 shows a cabinet corner post. The two grooves must be of uniform width along their entire length, at 90 deg to each other, and the two surfaces into which they are cut must of course form a similar angle and must be perfectly straight along their entire length to provide for the mechanical requirements of construction. The large curved surface calls for a substantial removal of material requiring about an inch of knife projection to cut the two rather delicate border beads, which must be done without any splintering or tearing. The least blemish of the three curved surfaces would fail to pass inspection.

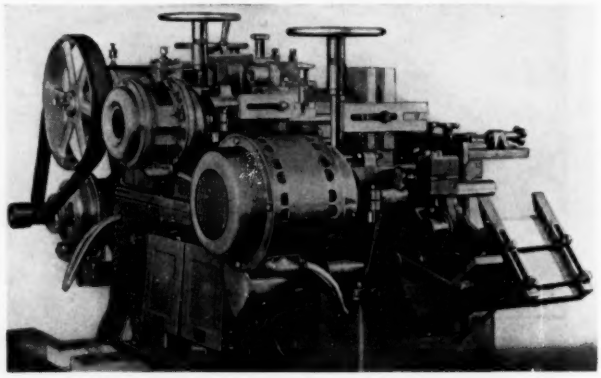


Fig. 15 30-Hp Motor, Developing Approximately 50 Hp at 6000 Rpm, on Bottom Spindle of a 6-In. Electric Molder

The pattern is made of built-up walnut. Thus loss on account of rejections would include the labor of gluing and of surfacing stock to prepare for gluing, besides the labor of molding and the cost of a comparatively high-priced lumber. Furthermore, the finish requirement dictates the best possible quality of work, because excessive sanding, to correct imperfect machining, would tend to change the contour of the two beads and thereby destroy the effect intended by the designer in the finished product. The example illustrates that good quality of molder work is also rewarded by less lumber waste and savings in preliminary and subsequent machining operations, as well as the better appearance of the finished product.

Fig. 14 shows a cross-section of stock, about 2 in. by 5 in., to be ripped into 12 automobile body roof slats of about $^{5}/_{16}$ in. by $1^{3}/_{4}$ in. Each has a quarter-round edge made with 12 cutters between 13 saws, on top and also on bottom spindles. It was known that the saw friction would require much power, and a 6-in. molder was originally equipped with oversize 20-hp top and bottom head motors, each developing approximately 33 hp at 6000 rpm when operated from a 100-cycle circuit, but even that was not sufficient to permit the desired rate of feed. The top-head load was relieved by throwing more of the slotting cut on the bottom spindle, and the latter was then equipped with a 30-hp motor which developed almost 50 hp at 6000 rpm. That provided the necessary power to feed 75 fpm, which was the maximum permitted by facilities for inspecting and handling the finished work. It illustrates one advantage of the greater power available with electric molders. Imagine belt sizes, belt speeds, and the pull and pound on the spindle to transmit 50 hp to one spindle and the resultant effect on the accuracy of making the 13 saw cuts with a belt drive! It is admittedly an extreme example, but exactly the same principles prevail, only in lesser degrees, with belt drives of smaller capacity.

Discussion

WARREN RICKETTS (Curtis Machine Corporation, Jamestown, N. Y.): In our experience, based on selling molders, there are just three fundamentals—feeding, guiding, and cutting. There is just one question that is put to me, especially when trying to sell an electric molder to a man who has a belt-driven molder. He will say that with our direct-motor-driven molder we have 10 hp on the top head, 10 hp on the bottom head, 5 hp on each of the side heads, and a 3 hp for feeding, and that nevertheless it is asserted that he will save power by having his machine with a total of 33 hp when he is driving his present machine with a 15-hp motor.

ANSWER (by the author:) This brings out an interesting part of the history of the electric molder. I did not stress, as I should have done, that perfect spindle rotation is the important thing of any molder. The big difference between an electric molder and a belt-driven molder is mainly a matter of better condition due to a better means of power transmission. In a belt drive, we have a belt pull on a pulley and on a spindle. It is driven at a very high rate of speed; there is a disturbing action there. In an electric molder, you have a self-centering, invisible magnetic force surrounding the motor and turning it with a self-centering action. There is a big advantage for truer spindle rotation. I cannot speak specifically on the power consumption of a ball-bearing, belt-driven molder and of an electric molder, because I have never made the test. The first suggestion of building an electric molder was with the idea of saving power. The first test required 40 to 60 per cent of the power required to do the work with the belt-driven molder. The electric molder will require about 50 per cent of the power required by a belt-driven babbitted-bearing molder. There was a test like this with a 6-in. electric molder equipped with a total of 33 hp, but when we turned on all four spindles and the feed motor, the power consumption was 3.2 hp. Now that 3.2 hp was consumed by electrical losses of the motor, by the action of the cutters against the air, and by the mechanical friction losses all the way through the machine. We ran it in competition with a 4-in. belt-driven molder which consumed 9.4 hp running idle. When we continued taking measurements of the power used to make an identical cut on the two machines, the electric molder required about 10 hp total to do certain work at 90 fpm, and the belt-driven smaller machine required over 17 hp. The question may come up why we put such large motors on the small molders. The answer is that one never knows how much power is going to be needed. One of the old troubles with belt-driven molders was that one did not have the power that he wanted. But as every electrical engineer knows, the motors only draw current in proportion to the cutting load.

R. L. TAYLOR (Jamestown Table Company): I have often wondered why they have not added a fifth or surfacing head. That is following our research. Why not add another head and therefore cut out an operation in the plant? Another thing, if there were a level or jump head on the molder, so that it could do fluting on a post that you do not care to carry all the way through, it might be a good thing. Is any work being done on that?

ANSWER: It depends of course upon the nature of the pattern, to begin with. If the problem were that of putting in a cut on the top of the post, it is comparatively simple. We have such a machine operating at Memphis, Tenn., but it is worked by gravity. There is an auxiliary head. As the end of the stock comes to a certain point it hits a little wheel, which turns the grooving saw into the cut. The point of entry into the cut is determined by a certain distance from this end. But this can become very much complicated and may develop need for a special machine. I believe the S. A. Woods Co. has a machine of that type. It is an auxiliary machine, not a part of a molder. The stock has to be turned to a 45-deg angle to do the work on that machine. As a universal machine, it is difficult to reply to the question. It might be too complicated. As long as you have a simple pattern to be duplicated, it would be a comparatively easy thing, but to build it to be used in many plants, I would say that it could not be done.

MR. TAYLOR: Would it not be possible to put a straightening device ahead of the molder to accomplish the same result as with the Oliver straitoplane?

ANSWER: Mr. Douse, of Grand Rapids, after becoming enthused over the Oliver straitoplane, asked why we could not do the same thing with a molder. We said we would try, and in 1927 we built such a machine for the Widdicomb Furniture Company, and as far as I know, it has worked out to expectations. For a year we made no more, so as to discover any deficiencies. In 1928 we built two more for use out on the Pacific coast. They are all giving satisfaction.

CHESTER L. BABCOCK (chairman): In the jointing of knives, what is the top speed at which you can joint effectively?

ANSWER: That is being done at 6000 and 7200 rpm. Whether it can be done at higher speeds, I do not know.

CHAIRMAN: Is that a test or would you recommend it in a furniture factory?

ANSWER: I am recommending it, but my general recommendation is to operate at one knife finish if you possibly can. I have recommended it to such an extent that I have been accused of being afraid that our molder would not hold up on joints.

CHAIRMAN: Would you do it on 1000 ft of linear molding?

ANSWER: We have not had much experience to encourage jointing, but on a pair of machines that were running a quantity of soft wood, one of our customers succeeded in feeding the molders at 250 ft with satisfactory results. They cut it down to 200 because that enabled them to handle the work better.

CHAIRMAN: Would you recommend a run of one knife or would you joint the knives in one head in a simple pattern?

ANSWER: I recommend one knife finish, even though it is a long run. That would slow down the rate of feed, but you will get a better quality of work. Some shops do good work with jointing where they have long runs to justify it. If you have a job of 50,000 ft, four-side surfacing, by all means joint, at whatever speed justified at the rate you can maintain.

CHAIRMAN: Would you not be safer at 3600 than at 7200?

ANSWER: The great trouble with jointing is the carelessness of the operator in not maintaining the joint equally true on all four knives on one head and all sixteen knives on four heads. The man that can joint at 3600 rpm and get good results can also joint at 6000, given the proper machine.

H. A. JOHNSON (Kling Factories, Mayville, N. Y.): We run our sticker at 6000 and get better work and more feeding speed than we did at 3600.

ANSWER: Speaking of four-knife work, one thing that has not been mentioned is that you would not think of running your shaper at 3600. What is the difference between a shaper and a molder? A molder has four spindles, but the principle of cutting is fundamentally the same. You want the same results with a molder as with a shaper.

CHAIRMAN: On account of the spindle speed, not much work has to be sanded.

MR. RICKETTS: I would like to ask about the sample of work done with four knives cutting at 190 fpm. Would it be possible to increase the speed of the feed if you increased the speed of the head?

ANSWER: There are certain other conditions that must be taken into consideration. That particular sample was run on

a 6 in. diam head, and 7200 rpm is about a maximum for that. If you tried to speed that cutter head up to 8400, you would have a knife speed that might be dangerous. We know that a 6-in. with good knives is safe at 7200. However, you could do that same work on a smaller machine, with smaller head diameters, and operate those heads at approximately 8400 rpm with exactly the same safety as in operating the larger head at 7200. Power consumption increases in direct proportion to the rate of speed. There is also a limit to the rate that you can feed due to the increase in strain that tends to splinter the stock.

Different Methods of Taking Lumber Out of Wind

By ARMEN S. KURKJIAN,[1] GRAND RAPIDS, MICH.

Boards are generally sawed out of the green log, and although as they come off the saw carriage they are quite straight, the saw has cut some fibers crosswise, others longitudinally, or across the grain in knot formations. The strains and stresses under which the cells adjacent to the saw were confined are now released, and the exposed surfaces of the board absorb moisture or lose it, this producing warp or wind in the lumber. In the drying of this lumber the fibrous cells often become rearranged, warping it out of shape. This condition is not objectionable for certain building applications such as baseboards and trim, where the stock is nailed in place, but when making furniture, musical cabinets, case goods, etc., it is important that the permanency of the joints be assured. For this reason machinery has been devised to overcome the warp or wind in boards to be used in these ways. The paper describes these mechanical processes.



FOR the purpose of this discussion the antiquated handtool methods of taking stock out of wind will be omitted, and the paper will be confined to the mechanical methods by the use of power-driven machinery. Because of the very nature of the substance and growth of trees it is impossible to find two logs exactly the same. This is true not only as between different species of trees, but as between trees of the same kind. The cellular construction or the fibrous mass forming the trunk of a tree consists of woody fibrous material, chemicals, and liquids commonly termed "moisture content." The distribution of these substances throughout a tree trunk is affected by many factors.

The influence of these factors changes the position of the wood fibers, causing strains and stresses in the cells that form the trunk of the tree. If one could in some manner reduce the moisture content in logs to 4 or 5 per cent before the log is sawed into boards, he might hope to have lumber less susceptible to being in wind or warp; but in general practice, boards are sawed out of the green log, and at the time of sawing as they drop off the saw carriage they are quite straight, although the saw has cut certain fibers crosswise, sliced other fibers longitudinally, or cut across the grain in knot formations to achieve straight-line sawing. The freshly cut cells and fibers of the two sides of the board are now exposed to the air, and the strains and stresses under which the cells adjacent to the saw cut were confined are now unequally liberated, causing a rearrangement of the internal stresses. This action, combined with the unequal loss or gain of moisture by the freshly cut cells, produces warp or wind in the lumber which cannot be physically prevented. Therefore there must be solved mechanically what cannot be solved physically.

Boards freshly sawn from the log are not usable until the moisture content is reduced to 3 to 7 per cent.

During both the natural air-drying and the kiln-drying methods of seasoning lumber of all kinds, as moisture leaves the lumber the fibrous cells often become rearranged in the board, so that the resultant dry board is warped out of shape and is said to be "in wind;" that is, when the board is laid on a perfectly flat table, only certain parts of it will touch the top of the table and other parts will extend away from it.

This warp or wind is not objectionable in certain applications. For example, in the home-building industry, in the case of baseboards, trim, etc., when the stock is finally finished and nailed in place, the nails draw it to an approximate straight position regardless of the warp or wind in the lumber; but in the manufacture of furniture, musical cabinets, case goods of all kinds, radio cabinets, etc., the straightness of the stock used is very important so as to assure the permanency of the joints; therefore lumber which is used in these industries must be taken out of wind.

The first concern should always be to minimize the warp or wind while the long boards are cut up into the necessary useful lengths for the article to be manufactured. Very often a long board which may have two or three twists endwise and a certain amount of warp crosswise can be so handled by the cutter that the resultant shorter pieces will be comparatively straight; they will at least have far less warp in them than the long board. The "wind" lengthwise is minimized by crosscutting certain lengths out of certain parts of the board. The warp crosswise and some of the warp endwise may be minimized by ripping the stock into narrower strips than the original width of the lumber.

In order to take stock out of wind it is very essential that in the planing process the lumber be passed over the cutters without any pressure to flatten out the board. Therefore the commonly known roll-feed, single- or double-surface planers cannot be used to take stock out of wind, because the pressure of the rolls flattens out the boards before a cut is taken.

The first mechanical method used for taking stock out of wind was to pass the warped board over the tables of a hand planer and jointer, the operator holding the board down by his hands only; the pressure of the hand on ordinary stock would not be sufficient to change its contour while it goes over the cutter head, and, depending upon the thickness of cut taken, either one, two, or three passes will eventually make one side of the lumber perfectly straight by removing the protruding portions

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

only. The objection to this first or hand-planer method of taking stock out of wind is that the board, being held only by the hands of the operator on a table, the length of which is normally less than the length of the board being jointed, will naturally have the first end held down against the table while the tail end, being unsupported, may be warped downward or upward, and in its turn going over the cutter head will change the position of the stock while it is being planed. And so, ordinarily speaking, the full amount of the warp would have to be taken out of one end or the other of the stock before one side is perfectly straight. Another natural objection, of course, is the slowness of the operation and the fact that any hand operation of this kind is neither efficient nor safe.

The second mechanical method introduced was that of having a set of sectional spring-loaded rollers superimposed in a frame directly above the rear table immediately following the cutter head in a hand planer and jointer. In this method the operator,



FIG. 1 LONGITUDINAL DIAGRAM OF BOARD WITH EXAGGERATED WARP OR WIND

(Showing how the longitudinal wind is minimized or eliminated by cross-cutting the long board into shorter pieces, taking the cross-cuts at the points of maximum bend.)



FIG. 2 CROSS-SECTIONAL DIAGRAM OF WIDE BOARD WITH EXAGGERATED CROSS-WARP OR WIND

(Showing how the cross-warp is minimized or eliminated by ripping the board into narrower pieces before any attempt is made to face-joint or plane.)



FIG. 3 EXAGGERATED SKETCH OF DRESSING LONG BOARD

(Showing how the facer dresses comparatively straight stock down to a flat by an assumed single pass.)

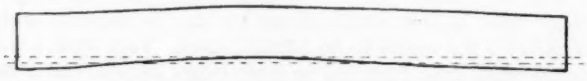


FIG. 4 TWO PASSES REQUIRED FOR BAD WARP IN BOARD

(With the machine set for average run of stock, two passes were required in this case. Had the depth of cut been set sufficient to clean up this and many pieces similar to it, the larger proportion of straight stock would have similarly suffered an unequally heavy cut.)

with his hands, advances the stock across the cutter head until the front end passes under the power-driven sectional rolls, which then in turn drag the stock forward and complete the passage across the cutter head. This method speeded up the process, but required that the front end of the stock be always held down; and furthermore the pressure of the sectional rolls necessary to feed the stock through was usually sufficient to flatten out the stock, making it impossible to secure an accurate face joint. But in many industries where the products were cheap and labor saving was the chief problem, this method was used effectively.

The third mechanical method of taking stock out of wind employed a "swing-out" feeding arrangement consisting of a series of sectional rolls mounted in a frame supported at one end by a vertical cylindrical bearing. The whole unit was arranged so that it could be swung around to a position immediately above the feed-in table, when power feed was required for the jointer; but when power feed was not required, this could be swung out of the way and the machine used as a regular hand jointer. This method was an improvement over the second method, because it tended to normalize the position of the board over the "feed-in" table before any cut was taken; however, the whole affair was not very satisfactory, because it lacked rigidity and definiteness and the pressure exerted downward was a considerable amount, and there was not a sufficient length in the feeding mechanism to hold the stock in an approximate mid-position while it was being fed forward.

The fourth mechanical method of taking stock out of wind consisted of superimposing on the normal hand planer and jointer an endless-chain feeding mechanism of a length almost equal to the total length of the tables of the hand planer and jointer, about one-half of this feeding arrangement being located directly over the front table and the remainder over the rear table of the jointer. In this case the multiplicity of the feeding fingers or cams, with slight pressure in each, simulated the hand method of shoving the board forward. Very little vertical pressure is applied, and since there are many fingers and the feeding mechanism is of sufficient length, the whole board is kept in practically the same contour and the same plane while being fed forward. In the first part of the machine the board is entirely rough, and in the latter part the already cut part of the board glides smoothly over the rear table of the jointer. The success of this method rests particularly in applying very little vertical pressure and having the dogs or feeding fingers come down on the upper face of the lumber just enough so as to have the forward-feeding effect without any flattening effect on the lumber. The first feeding device of this kind employed spring-loaded, vertical, chisel-point feed "fingers." The chief duty of the fingers, which contacted the upper face of the board, was to maintain the stock in a constant plane relative to the machine table. The power for moving the board forward across the cutter head was furnished by the row of fingers which would contact the rear end of the board. This method of using the facing planer has been very widely known and is one of the most successful mechanical means of taking stock out of wind. This method went far to overcome the objections to the former mechanical methods, but there remained one important point which had not been changed—namely, the necessity of running the stock, after it had gone through the facing planer, through a single-surface planer so as to cut down the top of the board, removing the superfluous stock and making the top straight and parallel with the first or bottom side, which had previously been face-jointed on a facing planer or jointer. Aside from the extra time and extra space required there is another important objection in the practical working out of running the stock through a facing jointer and then through a single surfacer to finish its rough planing, and that is the fact that usually in various factories stock is run through the facing planer and piled on a truck, and these truckloads stand around for a while before they are run through the roll-type planer; and since one side of the stock has gone through the facing planer and this has opened up the pores of the lumber afresh while the other side remains rough, the boards actually change their shape or again assume wind while they are waiting to go through the planer. This is due to the fact that the freshly opened pores will absorb moisture more quickly and to a greater degree than the rough side of the lumber.

Up to this point in the development of the art of mechanically taking lumber out of wind it was always necessary to first work on one side of the lumber and get that to be as nearly in a plane as possible, and then as a second operation the other side of the board was planed in a roll-feed surface planer of some kind to reduce the board to uniform thickness to make

the second side parallel to the first. These two separate operations were carried on for many years until about eight years ago, when a further advance in the art was made. This combined these formerly two separate operations into one tandem operation. Two machines were combined in one to effect sequential planing wherein the operations of face jointing and sizing down to uniform thickness are performed on the board in one pass through the machine, in one operation, at one handling, as follows:

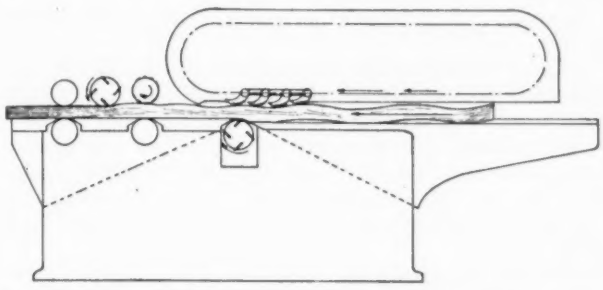


FIG. 5 PROGRESS OF BOARD IN WIND OR WARP THROUGH STRAITOPLANE

(This machine is described as the fifth or most modern method of correcting crooked lumber. The action of the cam dogs does not flatten out the warp. They feed the board in its natural contour across the bottom cutter head for face-jointing, which is immediately followed by the planing-to-thickness action of the top cutter head, so that the board leaves the machine as a flat slab, straight and sized to the uniform thickness desired.)

The fifth and most important method of taking the wind out of stock mechanically is by the use of a straitoplane, which may be described as a combination of a facing planer and a single-surface planer. In the first part of this machine the stock is fed with the least vertical pressure, but with a powerful forward pressure, by means of an endless chain having a multiple number of cam dogs. The stock moves forward over the long front table, and then, after the bottom of the board passes over the bottom cutter and is face jointed, the multiple cam dogs cause the stock to hug the rear table until in turn it glides under the sectional roll and is fed under the sectional chip breaker, the top cutter head, and the back pressure bar, and through the two outfeed rolls (similar to the arrangement in a single-surface planer). This finishes the work of planing the top side of the lumber, bringing it down to uniform thickness and having the top parallel to the bottom, which has been face-planed straight in the front part of the machine. In this method the face-jointing work and the single-surfacer work are done almost simultaneously on the stock, and therefore there is no time element left during which the stock may warp by having one side pick up moisture more than the other side. The result is that the boards which come out of the straitoplane are straight, having both sides planed perfectly parallel and to uniform thickness. They are easily piled on the trucks flat and remain flat. The straitoplane method of taking lumber out of wind is still very new, being only about eight years old, and therefore is the least known of any of the methods; hence, sometimes, people who have not seen the machine in operation advance an assumed objection to it by saying that since the stock passes through the machine once and the machine is set for the average boards, there might be waste in having the stock go through the machine and not having the bottom entirely cleaned while the top is planed off to the thickness that the machine is set. The fact is that in the straitoplane the front table is 6 ft long and the feeding mechanism over it is of such length that the boards up to 6 or 7 ft long are held in mid-position or in balanced contour over the front table before any cutting is done, and therefore only about one-half as much of a cut is necessary to take any certain amount of wind out of the board. Take for example, a double-twist warped board about 6 ft long and 10 in. wide, which has about $^1/_4$-in. wind in it; that is, when one end of the board is held down against a flat table, the other end of the board sticks up $^1/_4$ in. above the table. When this board is fed into the straitoplane, the cam dogs balance the same in an equipoised fashion over the 6-ft-long table so that diagonally opposite corners project only $^1/_8$ in. above the table and the board is pushed across the bottom cutter head without changing either the contour of the board or the plane of feeding. Therefore although the board has $^1/_4$-in. wind, a cut of only $^1/_8$ in. deep will face-joint the stock perfectly. In common practice it is not necessary that all of the rough spots be removed from both sides of the board in this first-cut stage, and therefore an extra thickness may be left in the board for the finishing cut, which is taken off either in a double-surface planer or single-surface planer. In actual practice in over 30 different woodworking industries the claim is that the straitoplane actually saves thickness rather than destroys it. In other words, out of the same grade and type of kiln-dried stock, the boards when finished can be kept thicker than the former method of face-planing in one machine until one side is all cleaned up and then surface-planing in another machine.

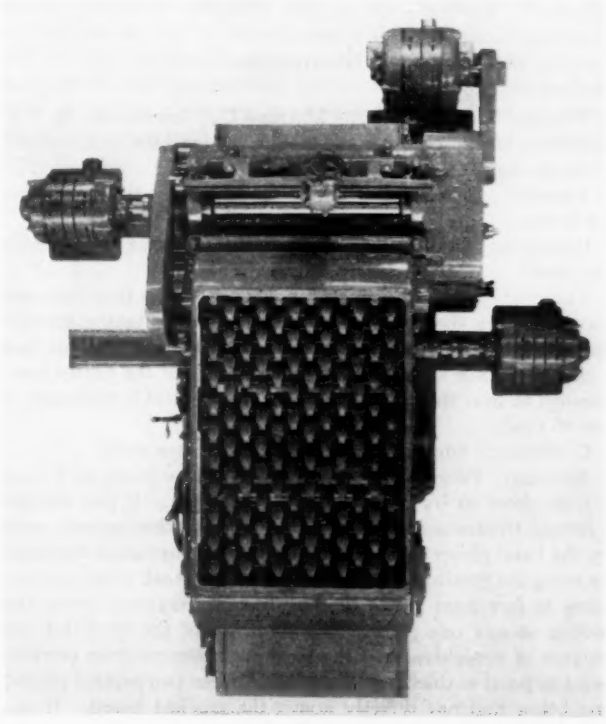


FIG. 6 MULTIPLE-CONTACT FEED MECHANISM OF STRAITOPLANE

(Knife-sharpening unit is shown over upper cylinder ready to sharpen the knives.)

In conclusion, the most modern or straitoplane method of taking wind out of stock has the following distinctive features in the operation of the machine:

1 The front table is 6 ft long, making it possible for stock up to 6 ft long to lie completely over the table before any cut is taken, and hence the feeding plane of the stock is not changed while going through the machine.

2 The endless chain is extra long, extending almost completely over the front table and at least one-third of the rear table. This makes it possible to equalize the position of the

stock with respect to the tables and to maintain that equipoised mid-position while being fed across the cutter head.

3 The cam-shaped multiple-feed dogs are pivoted from a fulcrum point well in the rear so as to minimize the vertical component and to augment the horizontal component of the forward feeding force, and hence the contour of the stock is not changed while going through the machine, and the result is more perfect face-jointing.

4 Immediately after the bottom of the board is face-jointed, it glides over the rear table and enters into the rear or single-surfacer part of the machine, where the top of the stock is planed down parallel to the bottom and the stock leaves the machine in the shape of straight, flat slabs planed on both sides to the uniform thickness desired.

5 The face-jointing operation on one side and the sizing or thickness-planing operation on the other side of the stock are performed almost simultaneously, in one pass through the machine. Thus, the freshly cut pores or cells on both sides of the stock are exposed to the air simultaneously, and since the action of the moisture in the air is the same on both sides, after the stock leaves the machine, there is greater surety of the stock remaining permanently straight.

Discussion

WARREN RICKETTS (Curtis Machine Co., Jamestown, N. Y.): In cutting stock, when it goes through the machine, is it brought down to size?

ANSWER: It is brought down to the size that the machine is set to cut.

QUESTION: If there is any wind left in it, what is done with the piece?

ANSWER: If the stock is still in wind, it means that there are rough spots in the board on both sides, and by taking another cut all wind can be taken out. Usually if the machine has had a chance to take a cut all over on both sides, if the knives have touched all over the face of the stock, the whole of it is straight—out of wind.

QUESTION: Supposing you have four-quarter stock?

ANSWER: Four-quarter stock may mean as much as $1^1/_4$ or $1^1/_8$ in. down to $^7/_8$ in. thickness in the rough. If you can get a certain thickness out of any board by any other process, even by the hand planer and jointer, you can get that same thickness by using the Straitoplane. The thickness of stock which you are going to face joint is not the physical measurement across the section at any one place. The thickness of the stock for the purpose of straightening is the total measurement from extreme point to point of thickness measured between two parallel planes, one below and one directly above the crooked board. If the machine is set for $^{15}/_{16}$ in., and the stock comes out with a little warp, the knives not having touched all over, no fault can be found. This is not a lumber-putting-on machine; it is a lumber-taking-off machine. The fact that there are rough spots on both sides shows that the machine is a lumber saver. If the board is put through the Straitoplane and does not clean up on both sides, it most likely does not come out straight. In that case it is run through again. We always get the proper thickness consistent with the stock.

QUESTION: Supposing you go through that process, and then put the board in the single planer, and it comes out not straight at the top, what do you do with it?

ANSWER: If you take enough of a cut on both sides, you will get your first thickness on this machine on any board.

CHESTER L. BABCOCK (President, Babcock Machinery Co., New York, N. Y.): Supposing a board is $^5/_{16}$ in. in wind, and you have the machine set for a definite thickness, what are you going to do with this piece—will it be spoiled?

ANSWER: If this board has a perfectly straight bottom, and the top of it varies as much as $^5/_{16}$ in. in thickness, the top cutter will make it straight, and you will get a perfectly straight piece out of it.

QUESTION: What if it is $^5/_{16}$ in. and the machine is set for only $^1/_8$ in.?

ANSWER: If this board has an exaggerated wind of $^5/_{16}$ in., then with one-half the cut, this machine will take twice the warp or wind out of it. If it is set for $^1/_8$-in. cut, then when the $^1/_8$ in. protruding part from each face is removed, it is equal to taking off $^1/_4$ in. by other methods, where the first part is pressed down first, and then gradually the hind part comes along.

QUESTION: But if it is a "dog" board—an unusually thick piece?

ANSWER: If I understand you correctly, you refer to a board in a truckload of 4/4 stock which is perhaps $1^1/_2$ in. thick at one end and $1^1/_4$ in. at the other end. If the operator knows his work, he does not put that board through the machine during the run of the truckload, but puts it on top of the machine. When he gets through with the load, he shifts the lever that moves the top of the machine, say, $^1/_8$ in., and he lowers the front table so as to take a heavier cut, and then he puts this special board through, and it comes out planed down properly. In a Straitoplane when one desires to reduce boards of great variations in thickness down to a common given thickness, the operator merely moves the front table up or down, but does not change anything else. The cut will be divided between the bottom and the top cutters as the set-up of the front table occasions.

QUESTION: What thickness do you usually set the bottom for? You must have some definite amount.

ANSWER: If I have 4/4 stock and I am taking a light cut on both sides, and I want to have a $^{15}/_{16}$ in. thickness left in those boards, then I will set the bottom knife cutter for a $^1/_{16}$-in. cut.

QUESTION: When you set the bottom head for the thickness you want to take, what do you do?

ANSWER: First you look at your order, and then at the truckload of lumber. The chances are that the operator will set the machine so that the bottom cutter will cut just a little under $^1/_8$ in. These boards all have some wind, and even though you set it for a $^1/_8$-in. cut at the bottom, it will not take off $^1/_8$ in. because of the wind in the stock. If it is all straight stock, you will set it at $^1/_{16}$ in. Any operator by just looking at the pile can tell how to do this.

QUESTION: If the stock is good and straight, can you set it for $^{30}/_{32}$ in. and will you get it straight?

ANSWER: If you do not have thickness enough in the board to clean up by any other process, it will not clean up by this machine. Say you have a truckload of 4/4 lumber. Assume that it has gone through the drying kiln and is supposed to clean up to $^7/_8$ in. Now in this truckload there may be some pieces that are nearly straight. With the Straitoplane set for the average board, you will take a thicker cut on the bottom of the straight boards than on the bottom of the crooked boards; and naturally you do not take as much of a cut at the top of the straight boards as at the top of the boards that are in wind because the latter have greater protruding portions; but in both cases you will get the thickness you want. In the Straitoplane, if the piece is straight, it is not going to be spoiled, because you will surely have enough material to get the thickness that you want.

QUESTION: Would you lose your thickness?

ANSWER: If you have lost the thickness, then it means that you could not have gotten it by any other process. The Straitoplane removes stock equal to only half the wind.

ARTHUR D. SMITH, JR. (President, Geo. W. Smith Woodwork-

ing Co., Philadelphia, Pa.): If I have a power feed jointer and place that in tandem with the single planer, what results will I get with the Straitoplane that I would not get with the two machines?

ANSWER: In jointers with feeding attachments you do not take as heavy a cut. Some of those boards would not have had sufficient cut at the bottom.

QUESTION: Why cannot one take a heavier cut?

ANSWER: Usually jointers with feeding attachment are built and recommended for taking a series of light cuts rather than one cut of sufficient thickness to remove the wind.

QUESTION: How does the price of the Straitoplane compare with a jointer with feeding attachment?

ANSWER: This machine face joints and sizes stock up to 36 in. wide. If you were to take the price of a 36-in. jointer with feeding attachment and add to that the price of a 36-in. planer, the price of those two would exceed the price of the Straitoplane.

QUESTION: But I have one planer.

ANSWER: Then of course your immediate outlay of money (since you already have a planer) would be about 40 per cent more for a Straitoplane than for a jointer with feeding attachment.

QUESTION: Would I get the same results by using a jointer with feeding attachment and a single planer in tandem as by using the Straitoplane?

ANSWER: You will get the same results, but if you have a lot of narrow stock, in the case of the Straitoplane two men feeding at one end and two men receiving at the other end and trying to pile, cannot keep up with it. In the case of the jointer and the planer in tandem, your boards will get mixed up and go in sideways instead of straight, and furthermore to do the same amount of work you would need extra men, which would be an extra expense.

QUESTION: Would we get the same jointing, with the same loss of thickness? Is there anything to be gained from the standpoint of thickness in stock from the one machine that you would not have in two?

ANSWER: I do not see what difference there would be. You should get out of one as much as out of the other.

AUTHOR'S CLOSURE

The questions by Mr. Ricketts and Mr. Babcock are based on the theory that if the Straitoplane is set for the final finished thickness of the stock and all of the excess material is cut away from the top and bottom of the board in its first pass through the Straitoplane without cleaning up both sides, the board cannot be used for its original purpose. This would be a fact if all of the conditions assumed are obtained; but in practice these conditions are not encountered. The Straitoplane is used chiefly for the first or rough planing operation. In practice it is never set to plane stock down to the ultimate cut once through the machine. Considering stock ordinarily used in any woodwork, where the straightness as well as smoothness is desired, the Straitoplane will face joint and size the stock to a uniform thickness in one pass through the machine, and it is left for the finishing or second cut either in the Straitoplane or in a finishing planer to remove all the rough spots from both sides of the stock and finish it to the ultimate thickness desired.



WDI-55-7

Moisture Content, Specific Gravity, and Air Space in Wood

By J. D. MACLEAN,[1] MADISON, WIS.

An exposition of the physics of wood is given in the paper. It describes the cell structure of wood and the physical changes that take place as it is dried to oven-dry condition or impregnated to maximum absorption. Formulas and graphs give the relation that exists between percentage of moisture content, specific gravity, water content, total weight, and percentage of air space. The data are of interest to those engaged in wood research or wood physics or in kiln drying, wood preserving, or fireproofing.

THE material wood contains wood substance, moisture, and air or gas. The wood substance includes both the cellular structure and extractives of the wood and usually has a variable quantity of water absorbed in the cell walls. When wood is green or wet, the water also occupies a part or all of the cell cavities. In addition there is usually more or less air in the wood cells depending on the density of the wood and the moisture content. In the present discussion the volume occupied by air or gas will be designated as air space.

The purpose of this article is to present graphic methods and simple formulas worked out at the Forest Products Laboratory from which the relations among moisture content, specific gravity, and air space in wood may be better understood and applied by the wood user to the many practical problems arising daily in the routine treating and seasoning of wood. For example, the amount of preservative that can be forced into the wood will be limited by the amount of air space, and when the moisture content and specific gravity are known, the approximate amount of air space can be computed. When this air space is known, it becomes possible to compute within reasonable limits about how much preservative can be absorbed as a maximum.

Moisture Content

The amount of moisture in wood, which is designated as moisture content, is commonly expressed as a percentage of the weight of the oven-dry wood. For example, if W is the weight of a sample before oven drying and D is the weight after oven drying, then the moisture percentage M is $(W - D)/D \times 100$. In order to obtain the weight of the wood when in an oven-dry condition, a small sample of the wood is heated at approximately 100 C (212 F) until the weight becomes constant. If the wood contains volatile substances that distill at the drying temperature, the term moisture by definition will also include these distilled volatile substances in addition to the water. In the following discussion, therefore, the terms "water in the wood" and "moisture content" include these volatile materials as well as the water.

When the weight of the oven-dry wood is used as a base, the moisture percentage shows the number of parts of water by weight to the number of parts of oven-dry wood by weight. For example, if a piece of wood has a moisture percentage M of 75, there are by weight 75 parts of water to 100 parts of oven-dry wood, or the water weighs three-fourths as much as the oven-dry wood. Again, if a timber has a moisture content of 100 per cent, the weight of the water is equal to the weight of the oven-dry wood. If the moisture content is 125 per cent, the water weighs one and one-fourth times as much as the oven-dry wood, or in other words the moisture is five-ninths and the oven-dry wood is four-ninths the total weight.

Since seasoning takes place most rapidly longitudinally, samples taken from near the ends of timbers that have been seasoning may show a moisture content considerably lower than samples taken at a point where end-drying would not affect the results. Furthermore, samples containing more or less sapwood may have a moisture content widely different from that of the heartwood. Samples should be weighed immediately after they are cut from the timber and as soon as they are taken fron the drying oven.

Plants impregnating wood with preservatives treat both round and sawed timbers, and it is therefore desirable to know the moisture content of the sapwood separate from that of the heartwood. Table 1 has been prepared with that in view. The moisture content values given in Table 1 were obtained at the Forest Products Laboratory and are average values determined as soon as possible after the trees were cut. These values probably show very closely the average moisture content of the living trees from which the samples were obtained. Individual timbers of any particular species, or timbers obtained from a given locality, however, may have a moisture content varying considerably from the average value given in Table 1. Variability in moisture content will also be found in wood from different parts of the sapwood and of the heartwood of the same tree.

Table 1 shows in general that the differences in the moisture content of the sapwood and heartwood are much greater in the softwoods, or conifers, than in the hardwoods, mainly because the heartwood is usually drier in the conifers than in the hardwoods.

Shrinkage and Fiber-Saturation Point

Water contained in the cell cavities of wood is commonly called free water, whereas that in the fiber walls is known as imbibed or hygroscopic moisture. The cell walls remain saturated until the free water has been removed. Any reduction of moisture in the cell walls causes the wood to shrink, and the amount of shrinkage will depend upon how much of the hygroscopic moisture is removed by seasoning.

The fiber-saturation point is defined as the moisture content of the wood when the cell walls are still saturated but no free water is present in the cavities. This moisture value varies for different

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 AVERAGE SPECIFIC GRAVITY AND MOISTURE CONTENT VALUES FOR 28 SPECIES

Name of species	Place of growth of material tested	Average percentage of moisture content when freshly cut		Specific gravity based on volume when green and weight when oven-dry	Specific gravity based on weight and volume when oven-dry
		Heartwood	Sapwood		
Hardwoods:					
Ash, white (*Fraxinus americana*)	Ark., N. Y., W. Va., Mass., Vt	45	40	0.55	0.64
Beech (*Fagus grandifolia*)	Ind., Pa., Vt	58	79	0.56	0.67
Birch, yellow (*Betula lutea*)	Pa., Vt., Wis	64	68	0.55	0.66
Chestnut (*Castanea dentata*)	Tenn., Md	120	...	0.40	0.44
Elm, American (*Ulmus americana*)	Wis., Pa., N. H	92	84	0.46	0.54
Gum, black (*Nyssa sylvatica*)	Tenn	52	58	0.46	0.55
Gum, red (*Liquidambar styraciflua*)	Mo	79	137	0.44	0.52
Maple, silver (*Acer saccharinum*)	Wis	58	97	0.44	0.50
Maple, sugar (*Acer saccharum*)	Ind., Pa., Vt., Wis	49	67	0.57	0.67
Oak, red (*Quercus sp.*)	Ark., La., Ind., Tenn., N. H	85	...	0.56	0.65
Oak, white[a] (*Quercus sp.*)	Ark., La., Ind	68	...	0.59	0.71
Softwoods:					
Cypress, Southern (*Taxodium distichum*)	La., Miss	120	171	0.42	0.47
Douglas fir (*Pseudotsuga taxifolia*)	Calif., Wash., Ore	40	116	0.45	0.51
Fir, lowland white (*Abies grandis*)	Mont., Ore	90	145	0.37	0.41
Hemlock, Eastern (*Tsuga canadensis*)	Wis., Tenn., N. H., Vt	97	118	0.38	0.42
Hemlock, Western (*Tsuga heterophylla*)	Wash., Ore	93	167	0.38	0.43
Larch, Western (*Larix occidentalis*)	Mont., Idaho	54	124	0.48	0.55
Pine, loblolly (*Pinus taeda*)	Ark., Miss., Tex., La., Ala., Fla	33	105	0.50	0.57
Pine, lodgepole (*Pinus contorta*)	Colo., Mont., Wyo	46	126	0.38	0.43
Pine, longleaf (*Pinus palustris*)	Miss., Tex., La., Ala., Fla	32	108	0.55	0.63
Pine, Norway (*Pinus resinosa*)	Wis	33	132	0.44	0.50
Pine, ponderosa (*Pinus ponderosa*)	Calif., Ore., Idaho, Mont	45	143	0.38	0.42
Pine, shortleaf (*Pinus echinata*)	Miss., Ark., Tex., La., Ala	29	112	0.49	0.56
Pine, sugar (*Pinus lambertiana*)	Calif	104	220	0.35	0.38
Pine, Western white (*Pinus monticola*)	Idaho	75	149	0.36	0.41
Redwood (*Sequoia sempervirens*)	Calif	88	148	0.41	0.44
Spruce, Engelmann (*Picea engelmannii*)	Colo	47	179	0.31	0.35
Spruce, Sitka (*Picea sitchensis*)	Calif., Wash., Ore., Alaska	50	131	0.37	0.42

[a] Commercial, average of six species.

species and is also influenced by temperature, but for practical purposes it may be assumed as about 25 per cent.

The shrinkage[2] of wood occurs largely in the radial and tangential directions—that is, at right angles to the growth rings and parallel to them, respectively—and is generally negligible in the longitudinal direction. Tangential shrinkage is greater than radial. The difference in shrinkage in the two directions varies for different species and even in different boards or timbers of the same species. An average ratio of tangential to radial shrinkage for the various species is about 1.8 to 1. The shrinkage in the volume of wood going from the green to the oven-dry condition is usually about 12 to 18 per cent for the hardwoods and 8 to 14 per cent for the softwoods. If seasoned wood is soaked in water until all the fibers are wet, it will swell until the fiber-saturation point is reached, or in other words, it will regain the dimensions it had when green.

During the process of seasoning there is a moisture gradient from the surface to the interior part of the timber. The interior of a timber may have a moisture content well above the fiber-saturation point while the outer portion is below. The distribution of moisture will depend on variables, such as species, size of timber, seasoning conditions, original moisture content, and whether sapwood or heartwood. On account of these variable factors the moisture content at different distances from the surface may vary considerably from the average value for the whole cross-section, and most timbers, particularly the larger sizes, will not shrink uniformly from the fiber-saturation point. In many cases, however, the average moisture content is of particular interest and is the moisture value most easily found.

Specific Gravity

Specific gravity is the ratio between the density of the material and the density of pure water at 4 C (39.1 F). In the C.G.S. system the numerical values for density and specific gravity are the same. For the English system the density of water is 62.4 lb per cu ft, while its specific gravity is still unity.

[2] Data on the volumetric shrinkage from green to oven-dry condition, average weights per cubic foot when green and air dry, and specific gravities based on the weight of the oven dry wood and volume when green, may be found for various species in U. S. Department of Agriculture Tech. Bull. 174, "The Air Seasoning of Wood."

Since wood normally contains a variable quantity of water, the specific gravity of wood is commonly based on the weight of the water-free wood and the volume at the moisture content under consideration. In other words, the specific gravity is the ratio of the weight when oven-dry of a given volume of wood at the current moisture content to the weight of an equal volume of water at its maximum density.

When the volume of the wood is in cubic centimeters and the weight is in grams, the specific gravity S is equal to the weight of the oven-dry wood per cubic centimeter. If the volume is measured in cubic inches and the weight is in pounds, S is the weight of the oven-dry wood per cubic inch divided by the approximate weight of a cubic inch of water at maximum density, which is 0.0361 lb. Weights of wood are commonly expressed in the United States in pounds per cubic foot, and the specific gravity based on this unit is therefore the weight of the oven-dry wood in pounds per cubic foot divided by 62.4. The value of S will of course be the same whichever system of units is used, since it is the ratio of the weights of unit volumes of wood and water.

Wood is at its maximum volume for any moisture content at or above that corresponding to the fiber-saturation point. The specific gravity based on the weight of the oven-dry wood and volume when green, therefore, will apply for all moisture-content values above the fiber-saturation point regardless of the amount of water present. As wood seasons, however, the weight of the oven-dry wood per unit volume will increase in proportion to the moisture lost below the fiber-saturation point owing to shrinkage and the accompanying increase in the amount of wood substance per unit volume. For this reason the specific gravity of wood will increase as the moisture content decreases below the fiber-saturation point, and the difference in specific gravity for any two moisture values will depend upon the degree of shrinkage. Obviously, then, the term specific gravity when applied to wood has little significance unless the moisture conditions at which the volume was measured are stated.

Since volume changes occur between zero moisture content and the fiber-saturation point, the specific gravity of any timber will vary between that based on the weight of the oven-dry wood and the volume when oven-dry and that based on the weight of the oven-dry wood and the volume when green. Average figures

for these maximum and minimum values are given in Table 1 for the species listed.

Since the specific gravity is computed from the weight of the oven-dry wood, based on unit volume at the current moisture content, the weight of the water in the wood has no bearing whatever on the specific gravity as herein defined. The only effect the water has is its influence on the shrinkage or swelling which occurs between the condition when the wood is oven-dry and that when the fiber-saturation point is reached.

In addition to changes in specific gravity caused by shrinkage or normal moisture changes below the fiber-saturation point there is a condition which sometimes develops during seasoning known as collapse. In this case the cell walls become collapsed to a greater or less extent, thereby causing a reduction in the amount of air space in the lumena or cell cavities. This condition would tend to increase the specific gravity, since there would be a greater amount of wood substance per unit volume.

It must be clearly understood that the specific gravity of wood may vary to a greater or less extent for different timbers of the same species. This is mainly because of differences in growth conditions, but may be due also to variations in the amount of shrinkage if the timbers are partially seasoned. Even samples taken from different parts of the same timber may show more or less variation in specific gravity values. It is therefore evident that specific gravity values, like those of moisture content, are only approximate even when the determinations are made from wood samples taken from the timber under consideration.

Specific Gravity of Wood Substance. In addition to the specific gravity of wood based on the weight of the oven-dry wood and volume at the moisture content under consideration it is of interest in making certain computations to know the specific gravity of the wood substance itself—that is, the specific gravity of the oven-dry wood without any air space or water in it. A given volume of oven-dry wood normally contains a large proportion of air space in the cell cavities and cell walls, and the volume occupied by the wood substance itself is considerably less than the total volume. The specific gravity of wood substance has been found to be fairly constant for all species, and various methods of measurement indicate that it is commonly between 1.50 and 1.58. This indicates that if oven-dry wood could be compressed so that no air space were present, a given volume of the wood substance would weigh something over one and a half times as much as an equal volume of water. An average value of 1.55 for the specific gravity of wood substance is used for computations given in this paper.

The specific gravity of wood substance should not be confused with the specific gravity of wood as commonly known and which is designated as S in the present discussion. In the specific gravity of wood, the wood is assumed to have its normal cellular structure, and as a result a given volume of the oven-dry wood contains both air space and wood substance. The specific gravity S will therefore always be less than the specific gravity of wood substance alone.

Relation of Specific Gravity and Moisture Content. Although it is not always true that the change in specific gravity S with change in the volume of the wood going from a green to an oven-dry condition is directly proportional to the moisture loss below the fiber-saturation point, it is sufficiently close to assume that this proportional relation exists. If then it is desired to find the specific gravity at some moisture content K below the fiber-saturation point, the specific gravity S_k at the moisture content K can be computed, as indicated by formula [10] given in the Appendix. For example, from Table 1 the average specific gravity S_g (based on the weight of the oven-dry wood and volume when green) is shown for loblolly pine to be 0.50, and S_d (the specific gravity based on weight of the oven-dry wood and volume when oven-dry) is 0.57. The computed specific gravity at 10 per cent moisture content would be, $S_{10} = \left[0.57 - (0.57 - 0.50)\frac{10}{25}\right] = 0.542$, or approximately 0.54. The percentage shrinkage in volume to this moisture content would be (formula [9], Appendix) $100\left[\frac{0.54 - 0.50}{0.54}\right]$, or approximately 7.4 per cent.

Fig. 1 has been prepared so that approximate values of S_k can be determined directly for the various species given in Table 1 and is based on the values of S_g and S_d given in the table.

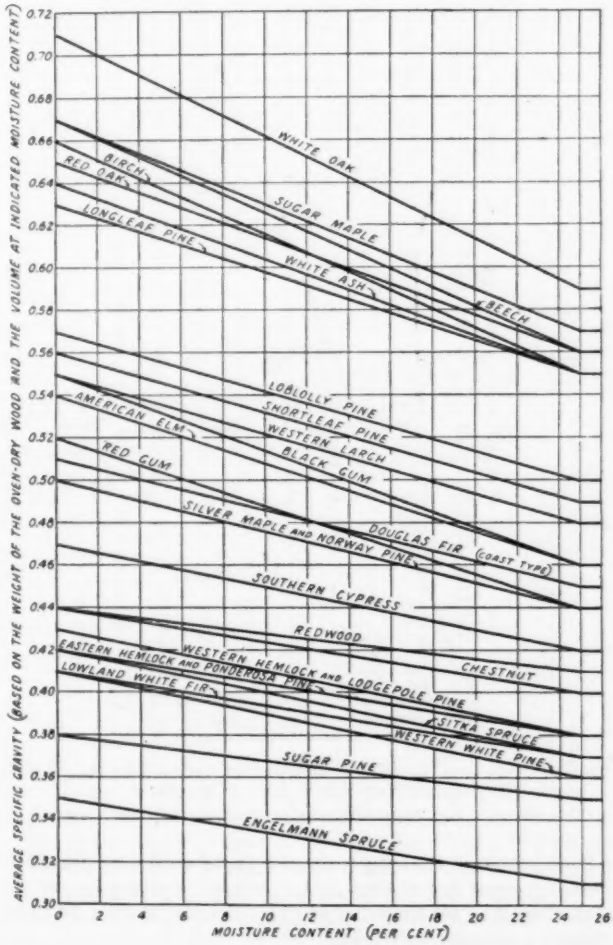


FIG. 1 RELATION OF AVERAGE SPECIFIC GRAVITY AND MOISTURE CONTENT

Specific Gravity Measurements. The specific gravity for any particular timber may be determined within reasonable limits from samples cut from the piece. Where this is not practicable, a hole about 1 in. in diameter and 2 in. deep can be bored to obtain the sample. The volume of wood removed can be computed from the diameter and depth of the hole, and the boring chips can be oven-dried in a paper bag. Samples cut from the timber should not be much over 1 in. in length (in the direction of the grain), as longer pieces require a considerable time for seasoning to the oven-dry condition. The samples should be taken far enough from the ends to avoid the influence of end-seasoning. Samples or borings must be weighed immediately after they are obtained and immediately after oven-drying to avoid unobserved moisture

changes. The moisture content and specific gravity may be determined in the same operation. For example, assume the original weight of a sample with dimensions 3 by 8 by 1 in. is 0.53 lb and the weight when oven-dry is 0.49 lb, the moisture content is then 0.53 minus 0.49 divided by 0.49, or approximately

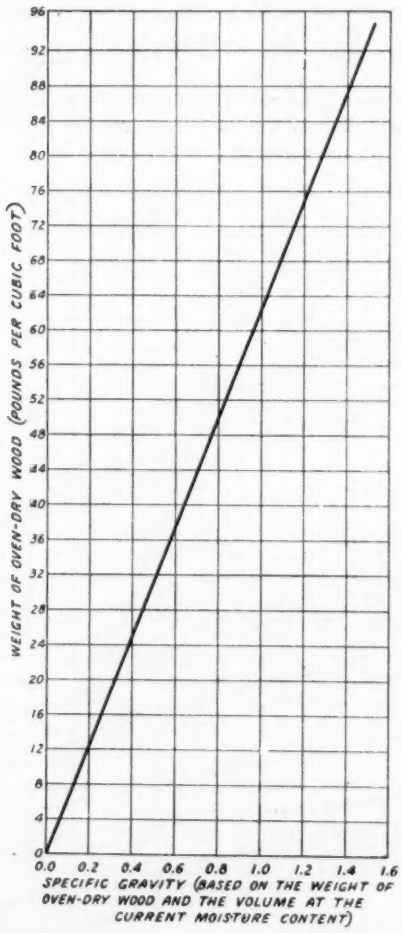


FIG. 2 RELATION BETWEEN WEIGHT OF OVEN-DRY WOOD PER CUBIC FOOT AND SPECIFIC GRAVITY
(The volume in each case being taken at the current moisture content.)

8.2 per cent, and the specific gravity, based on the weight of the oven-dry wood and volume at 8.2 per cent moisture content, is 0.49 divided by the weight of 24 cu in. of water (0.49 ÷ 24 times 0.0361), or approximately 0.57.

If the volume of the specimen is measured after oven-drying, the specific gravity based on the weight when oven-dry and volume when oven-dry can also be determined. For example, if the volume of the specimen, which was 24 cu in. at 8.2 per cent moisture content, is found to be 23 cu in. after oven-drying, the specific gravity based on this volume will be 0.49 divided by 23 multiplied by 0.0361, or approximately 0.59. By substituting 0.57 for S_k, 8.2 for K, and 0.59 for S_d in formula [10] of the Appendix, the specific gravity S_g, based on the weight of the oven-dry wood and the volume when green, is computed as about 0.53.

Relations Between Specific Gravity, Weight of Oven-Dry Wood, and Weight of Water in Wood. Figs. 2 and 3 have been prepared so that the weight of oven-dry wood per cubic foot and the pounds of water per cubic foot can be readily determined for wood having a known percentage of moisture and a given specific gravity. Since both moisture content and specific gravity are only average values, the indicated weights would of course be only approximate.

The maximum weight of water the wet wood can hold per cubic foot and the maximum percentage of moisture possible are found from Fig. 3 by reading the weight and moisture content corresponding to the point where a vertical line from the given specific gravity intersects the maximum moisture line which slopes downward to the right.

To illustrate the use of Figs. 2 and 3 by a concrete example, assume the species under consideration is loblolly pine, for which Table 1 shows an average specific gravity of 0.50 based on the weight of the oven-dry wood and the volume when green. For this specific gravity Fig. 2, or formula [2a] of the Appendix, shows the weight of the oven-dry wood to be 31.2 lb per cu ft of the green volume. At fiber-saturation point (25 per cent moisture) Fig. 3, or formula [5a], Appendix, shows that with the assumed gravity of 0.50 the wood has about 7.8 lb of water per cubic foot, and the maximum amount of water the wood could hold, assuming all the air space filled, would be about 42 lb per

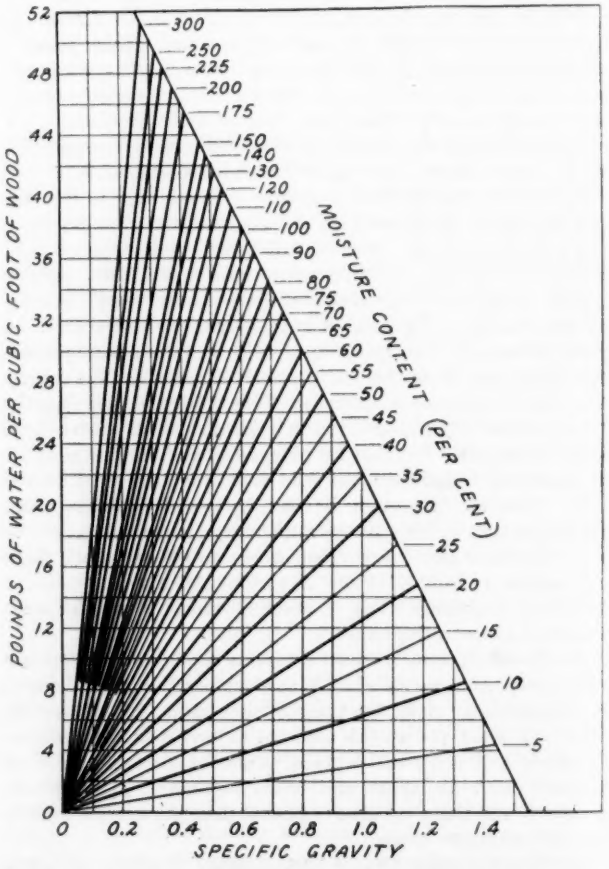


FIG. 3 RELATIONS BETWEEN WEIGHT OF WATER IN WOOD, SPECIFIC GRAVITY, AND PERCENTAGE OF MOISTURE
(Specific gravity based on the weight of the oven-dry wood and volume at current moisture content.)

cu ft. The volume occupied by the water (25 per cent moisture content) is found from formula [6] of the Appendix to be about 12.5 per cent. From Fig. 3, or formula [4] of the Appendix, it is also found that the maximum possible moisture content is about 135 per cent. The total weight of wood and water (formula [7a], Appendix) is the weight of the oven-dry wood per cubic foot found from Fig. 2, plus the weight of water per cubic foot found from Fig. 3.

A previous computation showed that if this species having a specific gravity of 0.50 based on the weight when oven-dry and volume when green were seasoned to 10 per cent moisture the specific gravity would be about 0.54. With this specific gravity and moisture content it is readily found, as in the preceding examples, that the weight of the oven-dry wood (based on the volume at 10 per cent moisture) would be about 33.7 lb per cu ft; the weight of water, 3.4 lb per cu ft; the volume occupied by the water, 5.4 per cent; and the total weight of wood and water, 37.1 lb per cu ft.

If in making moisture-content computations the weight of the oven dry wood is computed from the specific gravity (formula [2] or [2a], Appendix), it is important that the proper specific gravity figure be used. This specific gravity must be based on the weight of the oven-dry wood and the volume of the wood at the current moisture content—that is, the volume at the moisture conditions when the sample is taken.

For example, assume a green timber weighs 65 lb per cu ft, the specific gravity based on the weight when oven-dry and the volume when green is 0.53 and the specific gravity after air seasoning is 0.59. The weight of the oven-dry wood based on the volume when green is then (0.53 × 62.4), or 33.1 lb per cu ft, as can also be found from Fig. 2. From Equation [3a] of the Appendix the moisture content based on the weight of the oven-dry wood is found to be about 96 per cent. This can also be checked very closely from Fig. 3. Subtracting 33.1 from 65 gives 31.9 lb of water per cubic foot. By reading across from 31.9 on the vertical scale (Fig. 3) to the specific gravity 0.53, the chart shows that the moisture-content scale gives approximately 96 per cent.

If, on the other hand, the specific gravity based on the weight of the oven-dry wood and volume when air dry of this timber had been taken, the weight when oven-dry would be 0.59 × 62.4, or 36.8 lb, and the computed moisture content based on this weight would be about 77 per cent. The latter is an incorrect value, as it does not give, as in the first example, the same moisture content that would be shown if a sample were cut from the green timber and the moisture content computed as indicated in formula [3] of the Appendix, where $M = \left(\frac{W}{D} - 1\right) 100$.

Relation of Moisture Content and Total Weight of Wood. Frequently it is inconvenient to take samples for making a computation of moisture content and specific gravity. In such cases a rough estimate of the average moisture content can be made for moisture content values above the fiber-saturation point by weighing representative timbers, finding the weight per unit volume, and computing the average moisture content from the specific gravity data based on the weight of the oven-dry wood and the volume when green given in Table 1 for the species under consideration. For example, if red-oak timbers have an average weight of 50 lb per cu ft, it might be of interest to know about what the average moisture content would be for this weight. It would first be necessary to estimate whether the average moisture content is above the fiber-saturation point or not. This can be done by using the average specific gravity based on green volume (Table 1), assuming the moisture content at fiber-saturation point to be about 25 per cent, and computing the total weight by formula [7a] of the Appendix. With the specific gravity figure given for red oak, the weight at the fiber-saturation point is found to be 0.56 × 62.4 × 1.25, or about 44 lb per cu ft.

With a weight of 50 lb per cu ft, the average moisture content is evidently well above the fiber-saturation point, and is found by means of formula [3a] of the Appendix to be about 43 per cent.

Fig. 4 makes it convenient to determine approximately these moisture percentage values for variations from the fiber-saturation point to the maximum amount of water the wood can hold. The maximum weight line is the greatest weight (in pounds per cubic foot) the wood can have for the specific gravity given.

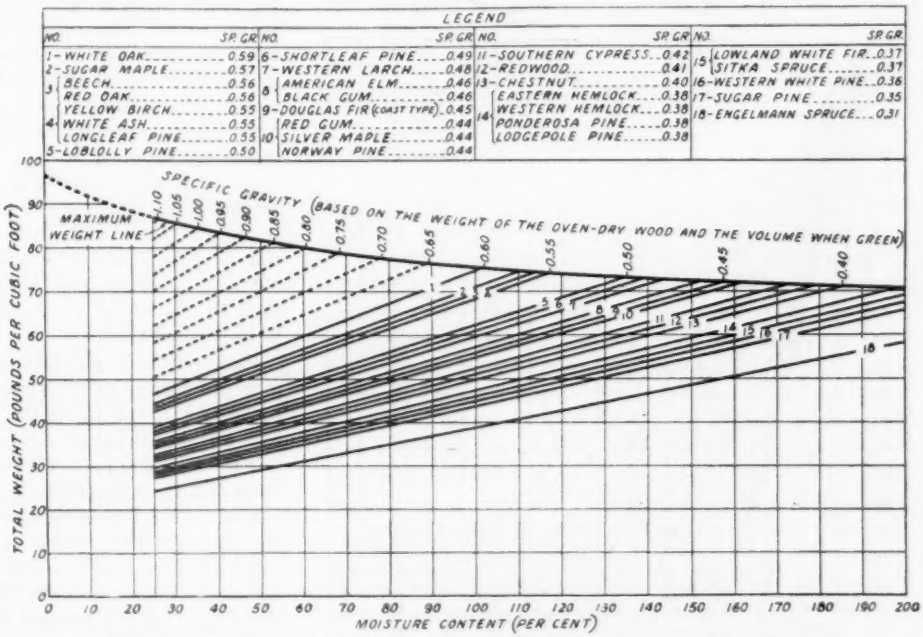


FIG. 4 RELATION OF WEIGHT OF GREEN WOOD AND MOISTURE CONTENT

The specific gravity figures in Fig. 4 are based on the weight of the oven-dry wood and the volume when green, since the wood is assumed at or above the fiber-saturation point.

Fig. 4 shows weights and moisture content values for differences in specific gravity of 0.05 and can be used for species other than those listed by interpolating between the lines connecting the different specific-gravity values shown on the maximum weight line. The dotted lines are for species that may have specific gravities higher than those of the species listed. If, for example, a wood had a specific gravity of 0.63 based on the weight of the oven-dry wood and the volume when green and weighed 60 lb per cu ft, the average moisture content is found to be approximately 54 per cent, a value lying between the lines for species having specific gravities of 0.60 and 0.65, respectively.

If weights of sample timbers are taken at intervals, Fig. 4 will also be found useful for estimating how fast the moisture content is changing during the seasoning period. The moisture content indicated from the weight of the timber will of course be only an approximate average value because of the uncertain influence of such variables as size and shape of the timber and amount of shrinkage at the surface where the moisture content may be below the fiber-saturation point. Furthermore, the specific gravity of the sample timbers may depart more or less from the average for the species.

The change in weight of oven-dry wood per unit volume below the fiber-saturation point offsets to some extent weight variations caused by changes in moisture content. On this account a considerable change in moisture content below the fiber-saturation point may cause only a relatively small variation in the total weight per unit volume.

Air Space in Wood

In a given volume of wood the air space is the proportion of the total volume that is left after subtracting the volume occupied by the wood substance and the water which the wood contains. The volume occupied by the wood substance will be the weight of the oven-dry wood divided by the weight of an equal volume of the wood substance. If, for example, a cubic foot of wood at 15 per cent moisture content weighs 36 lb when the wood is in an oven-dry condition, the volume occupied by the wood substance

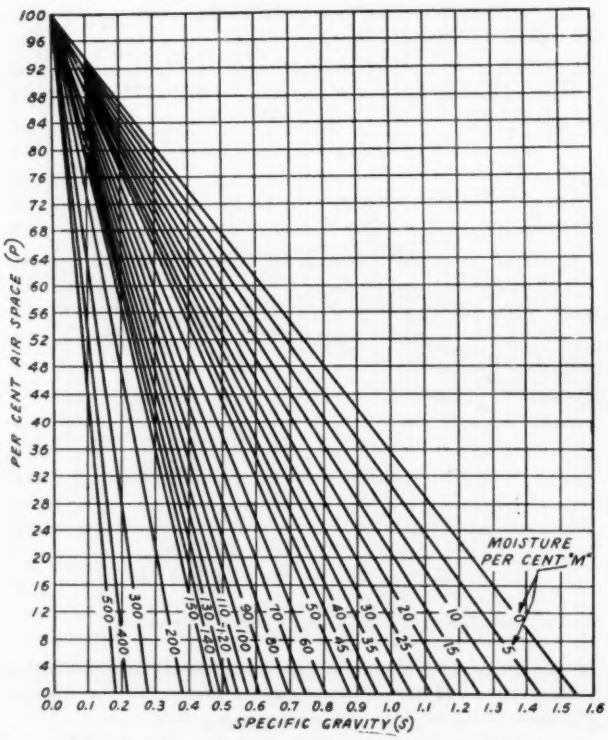


Fig. 5 Maximum Per Cent of Air Space in a Given Volume of Wood Having a Moisture Content M and Specific Gravity S (Specific gravity based on the weight of the oven-dry wood and volume at current moisture content.)

is 36 divided by (62.4 × 1.55), or about 0.372 cu ft. From formula [5] of the Appendix the weight of water equals 36 × 0.15, or 5.40 lb. The volume occupied by water is 5.40 divided by 62.4, or about 0.087 cu ft. The air space is then, by formula [8] of the Appendix, equal to [1 — (0.372 + 0.087)], or about 0.541 cu ft, or 54.1 per cent.

Fig. 5 has been prepared so that the percentage of air space can be found directly for a given moisture content and specific gravity. For example, with the loblolly pine having a specific gravity of 0.50 and moisture content of 25 per cent, the air space is found to be about 55 per cent, while at 10 per cent moisture (specific gravity = 0.54) the wood has about 59 per cent air space. The air space for intermediate moisture values can be found sufficiently close by interpolating.

Fig. 5 can also be used for finding the maximum percentage moisture content the wood can have, which is equal to P/S, where the value of P is the percentage air space, the value of which is found opposite the zero moisture content line, and S is the specific gravity based on the weight of the oven-dry wood and the volume when green. For example, when the specific gravity equals 0.50, the maximum air space is found from Fig. 5 to be 67.5 per cent at zero moisture content. In this case the green volume condition is assumed without any water in the wood. Dividing 67.5 by 0.50 gives 135 per cent moisture, which as previously noted was the moisture content shown in Fig. 3 as the maximum for this specific gravity. It may be of interest to observe that if, in formula [8a] of the Appendix, the moisture percentage M is zero and both sides of the equation are multiplied by $1/S$, the result gives formula [4] for finding the maximum possible moisture content. The point where a given moisture-percentage line intersects the abscissa (Fig. 5) is the average specific-gravity value of the species (based on the weight of the oven-dry wood and the volume when green), which would have the moisture content shown on the intersecting line as a maximum. For example, wood having a specific gravity of 1.0 could not have, at most, a moisture content of more than about 35.5 per cent even if completely saturated. In most growing trees, however, the heartwood has more or less air space, and this in many cases is also true of the sapwood.

In the wood-preserving industry computations based on the amount of air space may be of assistance in determining whether the wood has been sufficiently seasoned to obtain a given absorption of preservative or whether the specified absorption can be obtained, considering the moisture conditions at which the wood must be treated and the proportion of the total volume that can be impregnated.

If, for example, the loblolly pine timber having 10 per cent moisture content (specific gravity = 0.54 at 10 per cent moisture content) were treated with a preservative oil, the moisture content would not change, assuming the preservative does not contain water, and as far as the effect of the preservative is concerned there would be no appreciable change in volume, since the swelling caused by the oil absorbed is very slight. For this reason the absorption of preservative would depend on the amount of air space and how much of this air space was filled in treatment. Assume for illustration that the specific gravity of the preservative oil is 1 at the treating temperature, that all of the wood volume can be impregnated, and that all of the air space can be filled with preservative. When the air space (59 per cent) is completely filled, the maximum absorption that can be obtained is 62.4 × 0.59 = 36.8 lb per cu ft. On the other hand, if the wood at the same moisture content (10 per cent) were treated with a water solution, water would be absorbed in the cell walls as well as in the air space, and the volume would increase up to the fiber-saturation point, while the specific gravity would decrease to that based on the weight of the oven-dry wood and volume when green. The increase in volume would be about proportional to the change in specific gravity; that is, $\frac{0.54 - 0.50}{0.54} = 0.074$, or roughly 7 per cent. Since a cubic foot of the wood at 10 per cent moisture was found from Fig. 3 to have about 3.4 lb. of water, the amount of original water per cubic foot after treatment would be 3.4/1.07, or about 3.2 lb. From Fig. 3 it was found that the maximum quantity of water the wood could hold would be about 42 lb per cu ft. Then 42.0 minus 3.2 gives 38.8 lb of solution that could be absorbed as a maximum when the specific gravity of the solution is 1, or the same as that assumed for the preservative oil.

As another illustration, assume that at the time of treatment the sapwood in a charge of Southern pine poles averages about 60 per cent of the total volume, the average moisture content of the sapwood is 50 per cent, and the average specific gravity based on the volume when green is 0.50. A specification calls for a

net absorption of 18 lb of coal-tar creosote per cubic foot, and it is assumed that only the sapwood is penetrated. The question arises whether it is possible to obtain this absorption without further seasoning.

From Fig. 5 it is found that with a moisture content of 50 per cent and a specific gravity of 0.50, the sapwood has about 43 per cent air space, but since only 60 per cent of the timber is to be treated, the air space in the treated portion is equivalent to 0.6×43, or approximately 26 per cent based on the entire volume. The absorption is in pounds per cubic foot, and the specific gravity of the preservative at the treating temperature will be assumed as 1.03 for the purpose of computation.

The maximum gross absorption possible if the air space were filled would be $1.03 \times 62.4 \times 0.26$, or about 16.7 lb per cu ft. Furthermore, the net absorption is generally less than the gross absorption, even when a so-called full-cell treatment is employed, and under ordinary treating conditions not all of the air space in the treated portion could be completely filled. This is particularly true in the case of preservative oils which penetrate the cell walls only to a very slight extent. With the conditions assumed it is evident that it would be impossible to meet the specification requiring an absorption of 18 lb per cu ft.

If it is assumed that 80 per cent of the air space is filled with preservative, the percentage of air space necessary, based on the total volume, is evidently (18×100) divided by $(1.03 \times 62.4 \times 0.80) = 35$ per cent. Since only the sapwood comprising 60 per cent of the total volume is impregnated, the required air space in the treated portion is 35 divided by 0.6, which equals approximately 58 per cent. Fig. 5 shows that to obtain this amount of air space the timber would require seasoning below the fiber-saturation point if the specific gravity is 0.50. A close estimate of the moisture content and specific gravity giving this air space can be made by using Fig. 1. Assuming the species is loblolly pine and that the moisture content is 15 per cent, Fig. 1 shows the corresponding specific gravity for this moisture content to be nearly 0.53. With this specific gravity and moisture content, Fig. 5 indicates that the air space would be about 58 per cent.

Except in the case of sapwood or easily treated heartwood, only the outer part of the timber is impregnated. A simple calculation like the foregoing, however, shows the limitations of gross absorption which sometimes are not considered in the preparation of treating specifications.

It may also be of interest to know how much of the air space is filled with preservative for a specified treatment. For example, assume that a loblolly pine tie (all sapwood) has an average moisture content of 20 per cent and is to be impregnated by an empty-cell process with a net retention of 6 lb of creosote per cubic foot. At 20 per cent moisture content formula [10] of the Appendix, or Fig. 1, shows that the specific gravity would be about 0.514, or approximately 0.51. Fig. 5 shows that with a specific gravity of 0.51 and a moisture content of 20 per cent, the wood has about 56 per cent air space. If this space were all filled with preservative having a specific gravity of 1.03, the gross absorption would be $62.4 \times 1.03 \times 0.56$, or about 36.0 lb per cu ft. Since the tie is to be impregnated by means of an empty-cell method, there will be a "kickback," that is, a quantity of preservative rejected by the wood after pressure is released. This kickback may vary from about 50 to 65 per cent of the gross absorption. Assuming the kickback to be 60 per cent, the gross absorption would need to be 6 divided by 1 minus 0.6, or 15 lb per cu ft. That is, only 15 divided by 36, or about 42 per cent of the available air space would be filled during the treatment. Similarly only one-sixth, or less than 17 per cent, of the air space would be filled when the treatment was completed.

Computations such as the foregoing, of course, apply whether a full-cell or an empty-cell process is used.

Appendix

FORMULAS FOR COMPUTING RELATION OF MOISTURE CONTENT, SPECIFIC GRAVITY, AND AIR SPACE IN WOOD

The notations used have the following meanings:

- M = percentage moisture
- M_m = maximum percentage moisture when air space is completely filled with water
- S = specific gravity based on the weight of the oven-dry wood and volume at current moisture content M
- W = original weight of moisture specimen
- D = weight of moisture specimen when oven-dried
- W_v = weight per unit volume of wood at moisture content M
- W_m = weight of water in wood at moisture content M (weight per unit volume)
- W_d = weight of oven-dry wood per unit volume at moisture content M
- W_t = total weight of wood and moisture per unit volume at any moisture content M; that is, $W_t = W_v$
- W_w = weight per unit volume of water at maximum density (W_w equals 1 in C. G. S. system, and equals 0.0361 lb per cu in. or 62.4 lb per cu ft in the English system)
- P = percentage air space in wood having a specific gravity S and moisture content M
- 1.55 = specific gravity of wood substance
- P_w = percentage of volume occupied by moisture
- S_g = specific gravity based on the weight of the oven-dry wood and volume when green
- S_d = specific gravity based on the weight of the oven-dry wood and volume when oven-dry
- S_k = specific gravity based on the weight of the oven-dry wood and volume at current moisture content K below fiber-saturation point
- P_s = percentage shrinkage in volume in seasoning from fiber-saturation point to moisture content K.

Specific gravity:

$$S = \frac{W_d}{W_w} \quad [1]$$

When W_d and W_w are in pounds per cubic foot:

$$S = \frac{W_d}{62.4} \quad [1a]$$

Weight of oven-dry wood:

$$W_d = W_v \div \left[1 + \frac{M}{100}\right] = S(W_w) \quad [2]$$

When W_d is in pounds per cubic foot:

$$W_d = S(62.4) \quad [2a]$$

Moisture content in per cent:

$$M = 100\left[\frac{W}{D} - 1\right] = 100\left[\frac{W_v}{S(W_w)} - 1\right] \quad [3]$$

When W_v is in pounds per cubic foot:

$$M = 100\left[\frac{W_v}{S(62.4)} - 1\right] \quad [3a]$$

Maximum moisture content in per cent:

$$M_m = 100\left[\left(W_w - \frac{W_d}{1.55}\right) \div W_d\right] = 100\left[\frac{1}{S} - \frac{1}{1.55}\right] \quad [4]$$

Weight of water per unit volume of wood:

$$W_m = (W_v - W_d) = (W_d)\left(\frac{M}{100}\right) = W_w\left[\frac{MS}{100}\right] \quad [5]$$

When W_m is in pounds per cubic foot:

$$W_m = 62.4\left[\frac{MS}{100}\right] \quad [5a]$$

Percentage of volume occupied by water:

$$P_w = 100\left[\frac{W_v - W_d}{W_w}\right] = MS \quad [6]$$

When W_v and W_d are in pounds per cubic foot:

$$P_w = 100\left[\frac{W_v - W_d}{62.4}\right] \quad [6a]$$

Total weight of wood and water at any moisture content M:

$$W_t = \left[W_d + \frac{MW_d}{100}\right] = (S)\,(W_w)\left[1 + \frac{M}{100}\right] \quad [7]$$

When W_t and W_d are in pounds per cubic foot:

$$W_t = S\,(62.4)\left[1 + \frac{M}{100}\right] \quad [7a]$$

Percentage air space in wood:

$$P = 100\left[1 - \left(\frac{W_d}{W_w(1.55)} + \frac{W_v - W_d}{W_w}\right)\right] \quad [8]$$

$$= 100\left[1 - \left(\frac{S}{1.55} + \frac{MS}{100}\right)\right] \quad [8a]$$

Percentage shrinkage in volume in seasoning from fiber-saturation point to a moisture content K:

$$P_s = 100\left[\frac{S_k - S_g}{S_k}\right] \quad [9]$$

Specific gravity at any moisture content K below the fiber-saturation point:

$$S_k = [S_d - (S_d - S_g) \times (K/25)] \quad [10]$$

WDI-55-8

Modern Developments in Wood-Finishing Materials

By M. J. PEARCE,[1] READING, PA.

Wood-finishing materials as used today are classified according to composition, color, drying time, use, or other properties. The paper details the development work that has produced the modern finishes, and covers the purpose that they are designed to serve and the materials upon which they are used. The many stains, wood fillers, coaters and sealers, lacquers, and varnishes are described, their constituents are revealed, and their method of application is fully covered.



WOOD-FINISHING materials as used today are of many varieties and may be classified in many ways—according to composition, color, drying time, use, or other properties. Interest centers primarily in the practical angle of the use of finishing materials on such wooden objects as furniture, store fixtures, radio cabinets, pianos, and other similar commercial articles rather than in the scientific or theoretical angles of composition and chemical reaction connected with the manufacture and behavior of the individual materials. For the purpose of this paper, therefore, wood-finishing materials will be considered in a general way—first, according to the purpose which they serve, and second, according to the type of material. To further clarify the subject, the discussion will be limited to those materials used in producing a transparent or semi-transparent finish and not solid-covering primers, enamels, and other opaque finishing materials.

Many changes have taken place in finishing materials and methods during the last three decades. It is a far cry from present-day finishing methods to the "French polish" of 30 years ago, when a finish was worked up only after days of laborious rubbing with the shellac pad. The development or perfecting of the spray gun has been paralleled by an improvement in the drying time, toughness, durability, and other properties of finishing materials, so that it is possible to give a piece of furniture in one day a finish far superior in appearance and durability to that produced ten or fifteen years ago in ten days to two weeks.

The finishing of wooden articles has been effected with greater speed, more economy, and improved durability during recent years. The improved practical results are due to new raw materials, new technical research, and modification in methods of use. Through cooperation of the practical and laboratory technician, the finishing industry has improved the formulation of stains, fillers, lacquers, and varnishes, resulting in faster drying, greater durability, and lowered production costs. An examination of what has been accomplished may point the way to still greater expected improvements.

Stains

In the early days of the furniture and woodworking industry, the natural color of the wood was frequently the only color given to the finish. The early American maple, cherry, and walnut furniture illustrates this point. Mahogany and other well-grained woods gave a pleasing color when finisbed without stains.

With toe advent of large factories that turn out thousands of wooden articles in a year, such large quantities of the more desirable woods have been used that the quality of the color and figure has become less perfect. Furthermore, many species of wood which do not give such beautiful results without the use of stain, such as gum, elm, beech, and birch, have come into general use for such parts as posts and rails, and must therefore be made to closely match the color and figure of the more expensive woods used in other parts of the same article. These factors, together with the decided preference of the buying public for richer and deeper shades than those given naturally by many of the woods, have led to the use of stains.

Early in the use of stain, each operator made his own stain according to a special formula, sometimes inherited. Today few make their stains.

All stains may be divided into two general classes—powder stains and liquid stains. The powder stains are called water-soluble, spirit-soluble, or oil-soluble, according to the liquid in which they are dissolved. Liquid stains are divided into acid stains, spirit stains, penetrating oil stains, pigment oil stains, chemical stains, and non-grain-raising stains.

Powder Stains. Powder stains, or stain powders, are the dry coloring matter which gives the color to stains. Many are dyes made from intermediates manufactured from benzol, naphthalene, anthracene, and other coal-tar derivatives. Two or more intermediates must be mixed to produce a dye. Each color is a definite chemical compound with characteristic properties. Some are soluble in water, some in alcohol, and some in benzol, toluol, solvent naphtha, turpentine, lacquer solvent, and other so-called oils. Some are fast to light, while others fade easily. Mixtures usually are necessary to obtain a range of shades. Since not all of the colors can be used together safely, it is necessary to know their characteristics. Only experts in the blending of the powders can insure uniformity.

Of the powder stains, the water-soluble ones are the best and the most widely used. They are aniline dyes derived from coal-tar by-products. Sometimes they are incorrectly called "acid stains" because in the textile industry the colors frequently are mixed with acid materials. In the wood-staining industry acid is seldom added to stains, and when used it is very weak.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

There is no doubt of the merits of water-soluble aniline stains for all fine work. They are fast to light and permanent in color, while oil-soluble and spirit-soluble stains soon fade upon exposure to sunlight. Furthermore, water stains produce more uniform tones, and thus aid in bringing out the beauty of the wood. The water stains are inexpensive, since 4 to 8 oz of powder to 1 gal of water produces most of the popular shades.

The chief drawback of water stains is that they raise the grain and loose fibers of the wood, producing a rough surface, which must be sanded again before finishing. To aid the sanding it is customary to apply a wash coat of shellac made by adding eight to ten parts of denatured alcohol to one part of a 4-lb cut shellac, after the stain has dried. This stiffens the fibers and allows the sandpaper to cut them off smoothly. The water in a water stain has a natural affinity for the fibers of the wood, and from 24 to 48 hr should be allowed after staining the work before the next finishing operation, in order to let this excess moisture evaporate. Water stains are therefore not well suited to fast production schedules. Despite the low initial cost of water stains, they are expensive to use, when all of the costs in connection with a given finish are considered.

Spirit-soluble powders, as a rule, are aniline colors. They are generally conceded to be the least fast to light of all stains, but where extra-quick drying is desired, they are the logical stains to use, especially if not exposed to strong light. Formerly, they were used quite extensively for blending, matching, and unifying the figure or color of one piece of wood to that of another piece in the same article.

Spirit stains always contain some water, and therefore raise the grain of the wood to some extent, and for this reason and also because of their extreme sensitiveness to light, they are rarely used for finishing furniture, other than in overtoning or color patching to even up the color of the work.

Oil-soluble powders are usually aniline dyes and are more frequently purchased in the liquid form than are water stains or spirit stains. From the standpoint of fastness to light, oil stains are somewhat better than spirit stains, but they are not nearly so permanent as water stains. The red, orange, and yellow shades of oil stains are especially fugitive to light. Oil stains do not raise the grain of the wood, and for this reason have received much consideration.

Oil stains are quite penetrating, and consequently do not produce so uniform a color tone as water stains. The oil stain penetrates the soft portions of the wood and gives a much darker color here than over the harder spots. Oil stains are partially soluble in alcohol and are wholly soluble in most of the solvents used in the manufacture of varnish and lacquer. This causes an oil stain to "bleed," or to become partially dissolved, in succeeding coats of finishing materials applied over it. This bleeding tendency frequently spoils the work, not only from the color-change standpoint, but also because any appreciable quantity of oil stain in the succeeding coat of varnish or lacquer will interfere with the drying of that coat and result in a "mushy" or "rotten" finish, which eventually becomes brittle.

This slow-drying and "rottening" effect of oil stains is intensified by the fact that many of the colors are precipitated on a stearic-acid base at the time of manufacture. This stearic acid and other constituents of oil stains interferes with the adhesion and drying of nitrocellulose lacquer, and in some cases goes so far as to precipitate the nitrocellulose in the lacquer, which in turn causes a decided blushing of the lacquer. For these reasons, oil stains, except certain special shades recommended by stain manufacturers, should never be used under lacquers, and they should not be used under varnishes where best results are desired.

Liquid Stains. Liquid stains are made from powders dissolved in the proper liquids, although materials other than powder colors, such as dry pigments and asphaltum, are frequently added to liquid stains. So-called liquid acid stains are little more than water solutions of water-soluble powders, to which small quantities of other materials are sometimes added. Spirit liquid stains are usually solutions of spirit-soluble powders. Penetrating liquid oil stains consist essentially of a solution of oil-soluble powders.

Pigment oil stains are never shipped in the powder form and are so-called "surface stains" in that they do not penetrate deeply into the fibers of the wood. Pigment oil stains are made from dry pigment colors such as umbers, siennas, Van Dyke brown, and rose pink. They are usually mixed with drier, turpentine, naphtha, linseed oil, asphaltum and similar liquids which serve as a suspending medium until the stain is applied and as a binding medium for the pigment particles after the stain is dry.

Pigment oil stains are applied by spraying or dipping, on chairs and cabinet work of the less expensive and less beautifully figured woods. They are inexpensive, fairly permanent in color, do not raise the grain of the wood, and because of the tendency to hide the figure of the wood slightly, they give quite uniform tones, which, however, are not as clear as the colors produced with the other stains. To even up the color tones still further and reduce the tendency of the stain to hide the grain of the wood too much, pigment stains are usually wiped with a dry cloth shortly after being applied.

Chemical stains are liquid stains that depend upon a chemical reaction to produce the color effect. Some of the more common materials are permanganate of potash, iron acetate, tannic acid, and picric acid. These stains require such careful handling that they are seldom used where time is an important factor.

Perhaps the best example of the use of chemical stains is the imitation fumed oak formerly produced with ammonia fumes in an airtight chamber. The modern method is to apply a water solution of tannic acid, which does not materially change the color of the wood, but does even up the quantity of tannic acid in the wood, so that when another chemical solution is applied, the reaction between the tannic acid in the wood and the chemical in the stain produces a fairly uniform fumed-oak color.

Non-Grain-Raising Stains. From the foregoing discussion it will be seen that an ideal stain would be one having the permanency, uniform color tone, and non-bleeding properties of a water stain, coupled with the non-grain-raising and quick-evaporating properties of an oil stain. In other words, if the good qualities of a water stain and the good qualities of an oil stain could be combined, a new stain would be produced which would be fast, durable, and beautiful. After much development work, the modern so-called non-grain-raising stains were produced. This type of stain varies greatly, depending upon the manufacturer, but all have the same general aim—to prevent the raising of the grain of the wood, coupled with fast drying and permanence of color.

While non-grain-raising stains are not quite as fast to light nor of as uniform staining properties as water stains, they are satisfactory for all practical purposes. Because of the cost of the solvent, non-grain-raising stains have a higher initial cost than water stains. The saving in time, labor, and material, due to the elimination of the wash coat of shellac and the sanding operation, often compensates for the slight increased cost of the stain itself, however, and many woodworking companies have found non-grain-raising stains adaptable to their needs. Because of the fast evaporating nature of the solvent, the next finishing operation after the staining operation can proceed within 30 minutes to an hour, although it is wise to allow more time than this.

These non-grain-raising stains may be used under shellac, lacquer, or varnish. It is not necessary to apply a coat of shellac over the stain before varnish or lacquer is applied, as the non-grain-raising stains are non-bleeding. Another advantage is that the color of the stain is not picked up when paste wood filler is applied directly over the stain.

Other Miscellaneous Stains. Uniforming stains, glazing stains, shading stains, and other special stains are often used, either by themselves or with the other stains heretofore discussed. Uniforming stains are intended to unify the color of sap streaks and other defects in veneer or for matching the color of one type of wood to that of another type. Glazing stains or overtoning stains are frequently little more than pigmented oil stains, which are applied over other colors and wiped off to produce a glazed effect. Shading stains of the best quality are made on a lacquer base and by spraying lightly on parts of the work produce a darker color. The shading stains are usually applied after one or more coats of finishing material, but before the final coat. They are used more often with lacquer finishes than with varnish finishes.

Paste Wood Fillers

Practically all of the woods used in furniture and similar articles have open pores which must be filled before a smooth finish can be secured. In the old days, many thin coats of shellac or varnish were applied and rubbed down so that most of the finish was removed except that in the pores. For modern production methods the paste wood fillers have been developed.

In the woodworking industry today, paste wood filler is applied to fill the open pores of the wood and to prevent the succeeding coats of finishing materials from being absorbed. A good paste wood filler is composed essentially of silex, linseed oil, japan drier, and coloring matter. On most work, the filler is applied following the staining operation. It should be the same shade as the stain, or possibly a little darker.

Filler is usually reduced with naphtha, which should contain no grease, water, or sulphur. As a rule 10 to 12 lb of filler is used to 1 gal of naphtha. The more open the pores, the more heavy the filler should be. Also, a little heavier filler coat should be used under lacquer than under varnish.

The reduced filler is usually applied by brushing, although it may be applied by dipping or spraying. By the brushing method there is less likelihood of unfilled pores. After the filler has flattened on the work, it is padded into the pores with a pad of burlap or hemp, using a circular motion. The excess filler is then cleaned off with a cloth, first wiping across the grain and then lightly with the grain, being careful not to pull the filler out of the pores.

Until recently, paste wood fillers required at least 48 hours to dry before the next coat or finishing material was applied. Through development work, the drying time of fillers was reduced 16 to 24 hours, and peak-production demands brought about the fast-drying fillers that are sometimes referred to as "four-hour fillers."

Quick-Drying Fillers. The four-hour fillers were developed primarily for use on conveyor lines where the padding-in and cleaning-off operations can be done immediately after the application of the filler. The use of conveyors in the finishing room is the exception rather than the rule, and these extremely fast-drying fillers are therefore not well adapted for the average plant. Four-hour fillers have all the qualities of 24-hour fillers, but they must be padded in and cleaned up shortly after application or they will set up so hard that this operation is almost impossible.

Several filler manufacturers have revised the four-hour filler somewhat by lengthening the drying time to about 12 hours, without sacrificing the easy working properties of a 24-hour type of filler. This 12-hour filler is fast becoming the standard in woodworking plants because it does dry to recoat in that length of time, but still can be applied and worked in a normal manner.

So-called combination stains and fillers have also been developed to stain and fill the wood in one operation. To the ordinary type of filler an extra quantity of coloring material is added. Obviously, the range of color is quite limited, and so this type of filler is used only where low cost is more important than extremely sharp or clear color tone. High-grade furniture is seldom finished with a combination stain and filler.

Filler Quality Is Important. The filler is the foundation of any finish. Some fillers contain soap, borax, soda ash, weak alkali, and water solutions to partially saponify the oil and help hold the inert pigment in suspension. These materials give the filler a false body and prevent them from filling the pores properly. Another fault is a lack of permanence of color after wood lacquer has been applied. If the filler is not bone dry, a bleeding action frequently takes place, the finish taking on a muddy appearance and the filler turning gray in the pores. If the proportion of color or the character of the color itself is not right, the same grayish trouble develops. Wood lacquer is a natural solvent of oil-soluble or spirit-soluble colors. If these are present in the fillers, the powerful lacquer solvents are likely to bleed them out and leave the gray silex showing in the pores.

Many of the wood fillers which gave good results under varnish are unsatisfactory under lacquer because of the faults mentioned. Several manufacturers have improved their fillers so that they are equally well suited for use under lacquer or varnish.

Nitro-Cellulose Lacquers

Although clear nitro-cellulose lacquers had been used for years on metal articles to produce hard, tough finishes, it was not until about ten years ago that their use became practical for finishing furniture and other wooden articles. The old metal lacquers were viscous, requiring considerable solvent to thin them for spraying. This resulted in a low proportion of solid materials and made it necessary to apply many coats to build up a dried film of sufficient thickness to permit rubbing and polishing operations. Furthermore, they lacked proper adhesive qualities, especially when applied to wood, and could be peeled or stripped from the work quite easily.

Although the first attempts to use these metal lacquers for finishing wood were not successful, they led to the development of so-called "wood lacquers," which at first were little more than bronzing liquids or waterproofed substitute shellacs. They adhered to the wood better than the old metal lacquers, but the solid content was still so low as to give a very thin coating, suitable only for producing "shellac and wax" or other open-pore finishes. This type of lacquer is mentioned only because of the stimulating effect upon further research, which led to the development of more modern wood lacquers.

A little over ten years ago methods were developed for lowering the viscosity of nitro-cellulose solutions, enabling four times as much solid matter to be incorporated in a lacquer. This made it possible to put as much finish on the work in one coat as was formerly put on in four coats. Methods were also developed, at about the same time, for adding a proportion of varnish resins to the lacquer which greatly improved the adhesive qualities without impairing the hard, tough, waterproof features. New, better, and less expensive solvents and other ingredients were also made available at about the same time, thus making it possible to improve the spraying and rubbing properties at lower costs.

While many formulas have been developed since that time,

to meet specific plant requirements and conditions, the early formulation was so well and carefully done that certain clear lacquers which were first commercially available are still in use today, and cannot be improved upon for general use where a medium, solid content is satisfactory. They have such well-balanced working properties that they can be used under most all factory conditions to produce a tough, horny, durable film of medium thickness. Variations from these original clear lacquers are, in reality, only changes which have been made to meet production needs of higher solid content and lower costs. To be sure, in accomplishing these improvements, new raw materials have been utilized as fast as they have been made available.

Wood lacquers today, however, consist essentially of four classes or types of raw materials, namely: (1) Nitro-cellulose, (2) gums or resins, (3) solvents and thinners, and (4) softeners or plasticizers. Broadly speaking, the nitro-cellulose imparts hardness, toughness, and waterproofness; the resins give improved adhesion and greater film-forming and easier rubbing properties; the solvents and thinners dissolve the nitro-cellulose and resins and give the proper spraying, blush-resisting, and drying properties; the plasticizers counteract the brittle effect of the resins, increase the solid content, and prolong the life of the dried film.

While nitro-cellulose today is essentially the same chemical compound that it was a few years ago, many improvements have been made to give greater durability and lower viscosity to the finished product. Development work is continuing in this field and additional improvements may be expected from time to time.

Synthetic resins have been developed which partially or completely replace the older type of natural resins in many modern lacquers. Any manufactured resin can be classed in a sense as a synthetic resin, and the variety of results that can be produced is as unlimited as the number of resins themselves. The use of many of these synthetic resins imparts unusual properties to a lacquer film, such as increased solid content, toughness, hardness, flexibility and resistance to water, alcohol, acids, alkalis, etc. Still greater developments are expected in this very important group of raw materials.

The family of solvents, non-solvents, and thinners has been enlarged greatly during the last decade. Many new products have been developed, many of them from entirely new sources, which do their job better and at lower cost than the old solvents. There are three classes of liquids in this family. First come the nitro-cellulose solvents, which are used to dissolve the nitrated cotton; then there are the solvents for the resins; and finally come those materials which within themselves are poor solvents for either the nitro-cellulose or the resins, but which can be added in certain proportions to combined solutions of nitro-cellulose and resins to reduce their viscosity, dilute their solid content, and lower their cost. New developments in this field are announced almost daily.

Important strides have been made in the plasticizer field, also. Camphor and castor oil were the principal plasticizers in the old days, but a host of new products have been added to this group. Not of the least importance are the new processed vegetable oils, which through partial polymerization and other special treatment can now be used in large amounts to increase the solid content of the lacquer without sacrificing the drying or hardening properties. Some of the new synthetic resins also act to a certain extent as plasticizers, and the family of resins and plasticizers are often closely related to each other.

Research in the plasticizer field has resulted in the production of lacquers with greater toughness, flexibility, and durability. Undoubtedly many other developments in this field can be expected, since new improvements are being discovered from time to time.

The net result of improvements in each of these four types of raw material has been the gradual lowering of the cost of a lacquer finish on the work, through increased solid content, better working properties, and improved methods of use. A few years ago a satisfactory lacquer could not be made to contain more than 15 to 18 per cent total solids, without spraying, drying, and rubbing difficulties. Today, through the combined use of improvements in many raw materials, lacquers are available that contain approximately 25 per cent total solids and yet are low enough in viscosity to be sprayed without reduction. Certain development work is now being done that may result in raising the solid content another 5 or 10 per cent. If these hopes materialize, lacquers will then be available which will approach the long-sought goal of the spreading, film-forming, and low-cost properties of varnish combined with the quick-drying and durable features of lacquer.

Lacquer Sealers or First Coaters. For some time after the advent of lacquer finishing materials, shellac was used as the sealer coat over the stain and filler, or in many instances the sealer coat was eliminated entirely. Since shellac required a minimum drying time of three to five hours before sanding and the clear lacquers dried so hard they could be sanded only with difficulty, there gradually arose a demand for a sealer or first-coat material which would spray smoothly, dry rapidly, and sand easily.

The first developments in this direction resulted in the incorporation of finely ground translucent pigments, such as silica, into an ordinary clear lacquer. While these original sealers were more waterproof, less brittle, and quicker drying than shellac, much was lacking in body on the work, clarity of film, and easy sanding properties.

By replacing these pigments with metallic soaps, such as zinc stearate and zinc palmitate, and by substituting certain synthetic resins for some of the natural resins, it was possible to improve the drying time, sanding properties, clearness of film, body on the work, and durability of the ultimate finish.

Synthetic resins, new solvents, new metallic soaps, and many other raw materials not available a few years ago make it possible to produce sanding lacquer sealers today which contain 25 per cent total solids as compared to a former solid content of only 15 per cent.

These modern sealers spray smoothly, can be sanded in 30 minutes after application without gumming the sandpaper, and hold succeeding coats of lacquer much better than the earlier sealers. They have contributed their part to the reduction of number of coats, quicker finishing schedules, and lower costs of production demanded by the woodworking industry.

Flat Drying Lacquers. At first, after clear furniture lacquers were placed on the market, there were no flat drying lacquers. All of the lacquered work was brought up with clear gloss lacquers which were rubbed with pumice stone and oil to a dull finish. With the production demand for faster and more economical schedules and for lower unit costs, there arose a need for lacquers which dry to a dull (or rubbed effect) luster. These flat lacquers were produced, at first, by incorporating finely ground, transparent pigment particles into a clear lacquer vehicle. Lack of clearness of the dried film, poor water resistance, and low tensile strength were some of the defects of these early flat lacquers.

As time went on, new flattening agents, such as metallic soaps, waxes, and flat drying resins, were developed, and flat lacquers are now obtainable whose dried films are transparent, resistant to water, and practically as durable as clear lacquers. There has been a constant demand for flat lacquers containing more and more total solids, just as in the case of lacquer sealers

and clear lacquers. Flat lacquers are obtainable today which contain approximately 25 per cent total solids, which is from 5 to 10 per cent higher than two or three years ago.

Varnishes

For a time after the advent of clear furniture lacquer, rubbing and polishing varnishes took a back seat because of slow drying, lack of hardness and toughness, and other defects in the old-style varnish film which had been overcome by the nitro-cellulose lacquer. Varnish manufacturers have been constantly experimenting, trying to speed the drying of varnish products so as to approach the working properties of lacquer as nearly as possible and still retain the higher solid content of the old-style oil varnish. From the standpoint of spreading capacity, film-forming properties, and low initial cost per gallon, the oil type varnish is and has always been superior to lacquer, and if the fast drying, hardness, toughness, water resistance, and other good properties of lacquer could be approached without sacrificing the solid content, varnishes would again come into their glory.

Many new synthetic resins and specially treated oils have been developed which give promise of making this goal obtainable, although to date no so-called synthetic varnish is equivalent to lacquer for speed, hardness, toughness, or adhesion, unless the drying and polymerization are hastened by heat. Synthetic varnishes are available, however, which can be recoated within a few hours and satisfactorily rubbed after an overnight drying period.

It must be remembered that in the case of lacquer the hardening or drying of the film is accomplished almost entirely by evaporation, and oxidization or other chemical reactions enter into the drying phase only to a small extent. As soon as the solvents have evaporated sufficiently, the film is hard. In the case of varnish, however, the drying phenomenon is divided into two distinct phases. The solvents evaporate and set the varnish on the work so that it will not run or sag, but the principal hardening action comes about through a very complex chemical reaction, consisting chiefly of oxidization and polymerization of the vegetable oils in the varnish. This explains why it has not been possible to duplicate the fast hardening of lacquer in an air-drying varnish.

The development of suitable synthetic resins, partially polymerized oils, faster evaporating solvents, and more efficient metallic driers has made it possible to produce varnishes, however, which approach the drying speed of lacquers. It has also been possible to combine specially prepared varnishes with nitro-cellulose solutions to produce what might be called varnish-lacquers, although these materials are more often thought of as lacquers rather than as varnishes, because of the similarity of odor to that of lacquer. Development work in this field is going on constantly, and undoubtedly it may be expected that lacquers and varnishes will become nearer and nearer alike as time goes on.

On medium- and low-priced furniture there has been a decided tendency to return to the use of varnish, for economical reasons only. Varnish will spread further, costs less per gallon, and will produce a finish of a given depth with fewer coats than lacquer. Therefore, where the price of the furniture is not sufficiently high to justify the cost of using lacquer, varnish is being used.

Flat Drying Varnishes. For a long while, flat drying varnishes have been manufactured by incorporating finely divided, transparent pigment particles in an ordinary type of rubbing varnish. These flat drying varnishes have had certain defects, such as the gray appearance in the film, poor resistance to water, and brittleness. Much development work has been done recently to improve flat drying varnishes, and as a result many of these defects have been eliminated. A new style of flat varnish that has only recently been developed contains little or no pigment. It therefore does not settle or skin over in the package, it dries much faster than the ordinary type of flat varnish, and it gives much clearer results on the work.

The film of all flat drying varnishes is necessarily not quite as tough nor durable as the same grade of clear rubbing varnish. The flatting agents all have a deteriorating effect upon the varnish, but where a rubbed effect is desired, without the expense of an actual rubbing operation, some very beautiful results can be secured with modern flat drying varnishes.

Conclusion

It will be seen from the foregoing discussion that as a need develops in the wood-finishing industry, the finishing-material manufacturers try to solve the problem, and in many cases with success.

During the last five or six years, the development has primarily been in keeping with the increased pace of production. The cry has constantly been for faster drying, fewer coats, and lower costs. Some woodworking factories have carried this matter so far that today the finish often lasts only a little longer than the time necessary to get the article to the hands of the consumer.

After all, the finish on a wooden article goes a long way toward attracting the customer and keeping him satisfied. It would be far better to spend a little more money on the finish and build a reputation for the future.

Note: Discussion of this paper jointly with that of R. H. McCarthy and L. A. Lefcort by F. L. Browne appears in the discussion following the latter paper.

WDI-55-9

Accelerated Varnish Drying

By R. H. McCARTHY[1] and L. A. LEFCORT,[2] MASPETH, N. Y.



R. H. McCarthy

Varnish drying on wood at a temperature of 150 F, instead of the usual lower limit of 100 to 110 F, has been attained after experimentation and tests by the authors. This finish process retains the desirable factors of appearance, durability, economy, and speed. Varnish finishes at elevated temperatures are shown by tests to be superior to ordinary finishes in durability and resistance to marring. Samples made by this system have undergone accelerated weathering for several months without failure, indicating that those finishes that take advantage of the toughness and hardness resulting from the heat treatment of synthetic and some natural resins are superior to the ordinary run of long-oil varnish finishes. A feature of the system is that it makes feasible the conveyorization of varnish finishes as well as lacquer finishes.



L. A. Lefcort

GIVEN the desired appearance, durability, and cost of material and labor for wood finishes, furniture manufacturers have consistently favored quick-finish processes. In many processes, however, appearance, durability, economy, and rapid drying are not all possible because of inherent limitations of the materials. Thus many slow-drying varnishes produce fine appearance and excellent durability only by sacrificing economy and speed. Some lacquers and varnishes, on the other hand, offer appearance, economy, and speed, but are lacking in durability. Therefore a great amount of time has been spent to secure speed without loss of durability and appearance. The results have been greatly improved materials, particularly the phenol formaldehyde, glyptal, and other synthetic resin varnishes, and radical changes in finishing procedures, such as the use of accelerated drying at relatively high temperatures.

Most high-grade long-oil varnishes ordinarily take from 8 to 48 hours per coat to dry satisfactorily at room temperatures, dependent upon atmospheric temperatures, humidity, and varnish formulations. Acceleration of shellac and varnish drying by the use of ovens operating at 100 to 110 F has been common practice for many years in the woodworking industry. Likewise it has been common practice to bake varnish finishes on metal at much higher temperatures.

The following description of a wood-finishing process employed by the Western Electric Company in the manufacture of telephone booths is illustrative of what the new-type varnishes have made possible in acceleration of drying schedules and in increased durability of finishes. The development proposed to raise the drying temperature as far as the materials involved would permit—i.e., wood, glue, filler-stain, and varnish. As a result the drying time has been reduced to $1^1/_2$ hours per coat regardless of atmospheric conditions, thereby putting varnish into a class with lacquer in respect to speed of production. The investigation was initiated and developed by the authors in connection with the finishing of wood products at the Queensboro Works of the Western Electric Company. Dr. A. E. Schuh and Mr. W. L. VanArnam, of the Bell Telephone Laboratories, cooperated in the investigation and conducted the laboratories' tests to determine durability,[3] which resulted in the selection of two varnishes having synthetic resins and one having natural resins as superior in resistance to abrasion, impact, and accelerated weathering.

A steam-heated oven, Figs. 1 and 2, insulated with a 2 in. thickness of 85 per cent magnesia and equipped with an automatic recording temperature regulator, was installed. Fresh air is drawn from outdoors through a filter box, past a regulating damper, through a fan and steam coil, and forced into the oven proper through adjustable louvers. From another set of louvers in the opposite wall of the oven the air is drawn back to the regulating damper, which is set to discharge a portion outdoors and to recirculate the remainder with the incoming fresh air. The amount of recirculation was fixed so that the fresh-air supply was sufficient to avoid any possibility of explosive mixtures. No attempt was made to humidify the air, since it was thought that the moisture change of wood during the relatively short time in the oven would not be serious. Subsequent tests (shown in Fig. 4) verified this belief.

TABLE 1 TESTS OF SHEAR BLOCKS

(All tests made immediately after heating)

	Load at failure, lb					
	Heated 3 hours			Heated 6 hours		
	Walnut	Mahogany	Oak	Walnut	Mahogany	Oak
Room temp, average of 5 samples	2668	1588	2394	2668	1588	2394
140 F, average of 5 samples	1948	1426	1828	1840	1466	1640
150 F, average of 5 samples	1952	1286	1834	2094	1258	2098
160 F, average of 5 samples	1890	1170	1758	2176	1362	2138

NOTE: All samples when fully cooled showed same strength as before heating.

An investigation was planned to furnish the following information:

1 Proper operating temperature with respect to (*a*) the wood and (*b*) the glue joints

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society

[3] "The Evaluation of Industrial Finishes," by A. E. Schuh, *Ind. & Eng. Chem.*, vol. 23, Dec., 1931, pp. 1346–1352.

2 Minimum drying time for various coats
3 Most advantageous sequence of operations
4 Method of varnish application.

It should be stated that the system described herein was applied to solid and veneered exposed mahogany, oak, and walnut telephone-booth parts; to end assemblies, doors, door frames, backs, seats, and shelves, as shown in Fig. 3; and to a few other

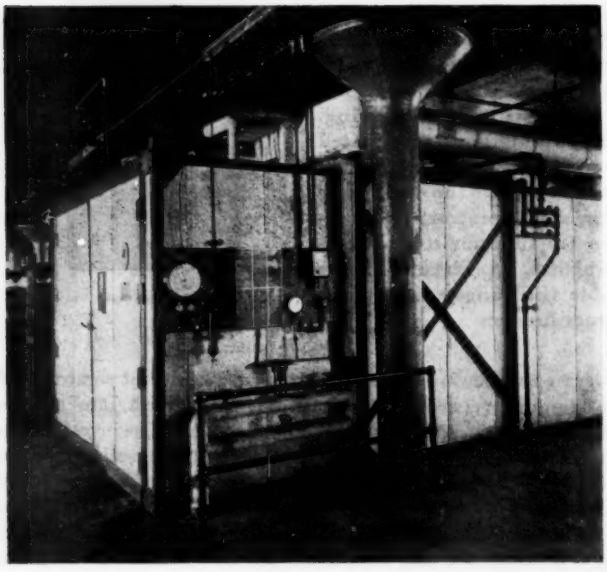


FIG. 1 EXTERIOR OF STEAM-HEATED VARNISH-DRYING OVEN
(Showing automatic controls.)

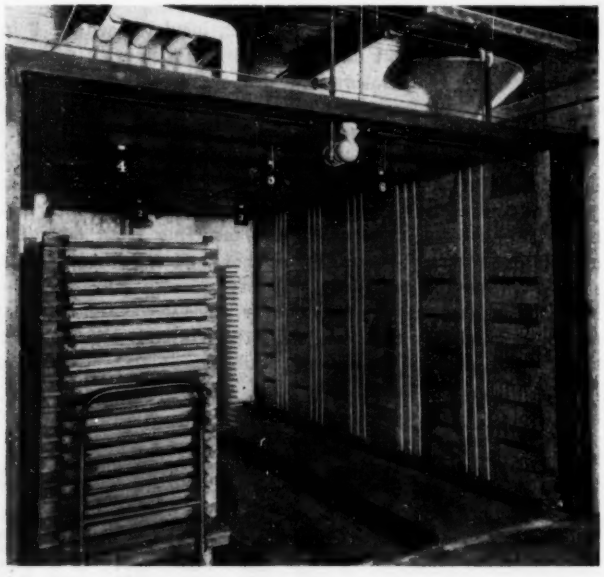


FIG. 2 INTERIOR OF STEAM-HEATED VARNISH-DRYING OVEN
(Showing adjustable louvers for cross-ventilation.)

items of miscellaneous furniture. Prior to the investigation an all-varnish system had been adopted wherein the use of non-bleeding filler-stains eliminated the use of sealers other than the varnish itself.

OPERATING TEMPERATURES

Guided by a series of preliminary tests that had indicated a temperature range from 140 to 160 F for more detailed investigation, several complete unfinished unit assemblies were subjected to oven temperatures of 160, 150, and 140 F for 5-hour periods to determine what defects, if any, would result from the heat treatment. The assemblies were carefully inspected for warpage, surface checks, shrinkage, and defective glue joints. At 160 F the mahogany and walnut showed no visible damage, but quartered white oak exhibited small surface checks and open glue joints in the faces of veneered panels. Upon lowering the temperature to 150 F, there were no indications of surface checks and very little opening of face-veneer joints. Similar results were obtained at 140 F. A further examination proved that in every instance of open veneer the joints had been improperly jointed or glued before the tests were made.

At the outset of the test a fear was expressed that unhumidified oven treatment would create serious changes of the wood moisture content, with resulting warps, hairlines due to inset-panel shrinkage, and deterioration of the finish. For this reason a series of tests were made to show the successive moisture changes during each step of the finishing process. Eight adjacent test sections were sawed from the middle of a five-ply $^{7}/_{16}$-in. oak-plywood panel. Two of these sections, Nos. 1 and 8, were tested at the

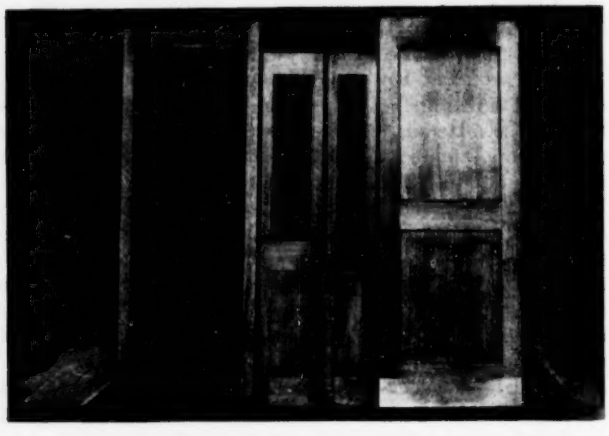


FIG. 3 VARNISHED TELEPHONE-BOOTH PARTS
(Showing seat, door frame, folding door, end panel, and shelf.)

outset for moisture content and taken to be representative of the initial condition in other sections. The remaining six sections then received a coat of filler stain, were sealed back into the large panel, and air-dried for 24 hours at room temperature. When dry, section No. 2 was taken out, the filler stain was sanded off, and the section was tested for moisture content. The same procedure was followed for the two succeeding varnish coats after their respective oven-drying periods.

The results as given in Fig. 4 show clearly that the fluctuations of moisture content are not large, since the stock when ready for rubbing was little different in condition from that in the ordinary air-dried system. Undoubtedly these results were approximate, because thinners and oils from the finishing materials not removed by sanding and process-drying appeared as moisture content. Since the tests were made, the air-drying of filler-stains has been replaced by oven-drying, which would reduce the high indicated moisture content shown in the graph at this stage of the cycle (see sample 2, Fig. 4).

In order to determine the effect of temperature on glue joints, tests were made on oak, mahogany, and walnut blocks that had been heated at 140, 150, and 160 F for periods of 3 and 6 hours. For this purpose shear test blocks constructed of three thicknesses of lumber were made according to accepted practice and were

tested to destruction. The results are given in Table 1. It should be noticed that they were tested immediately upon withdrawal from the oven; hence the results showed, as might be expected, decreased joint strength while the wood was warm. But after the blocks had cooled for several hours, the glue joints returned to their original strength, thus indicating no permanent deterioration. The net results of these tests pointed to surface checking and opening of veneer joints in oak as the limiting condition, and therefore 150 F was chosen as the proper operating temperature for further work.

Filler-Stain Drying Time

Five combination filler-stains are used in the standard variety of product: light red, dark red, and walnut brown applied on mahogany solid and veneered construction, a dark walnut on black walnut, and a medium golden oak on quartered white oak. All these filler-stains except the dark mahogany were readily adapted to the accelerated schedule, drying satisfactorily in 1 1/2 hours at 150 F. After extended trials with dark-mahogany stains in which a deep-red color had to be obtained without bleeding through varnish, yet had to be permanent and adaptable to forced drying and suitable for application in combination with a filler, a non-grain-raising water-soluble stain appeared to be acceptable. It does not dry so hard as the other filler-stains, but after the additional drying that it receives during varnish-coat drying, the finished product proves satisfactory.

The brush application of filler-stains is followed after a 15-min hardening period by padding and wiping off. The stained and filled articles are immediately placed in the oven and remain for 1 1/2 hours at 150 F. They are then ready for varnishing.

Varnish-Drying Time

The problem of accelerated drying had been greatly simplified by the results of considerable experimentation at the Kearny Works of the Western Electric Company and the Bell Telephone Laboratories, in which it became evident that for durability and appearance a finish using varnish directly over the filler-stain compared very favorably with other methods of sealing. This simplified the problem, because it left only the filler-stains and varnish to deal with, eliminating sealers as a third variable.

Various combinations of drying times and temperatures for the first and second varnish coats were tried. Some of these combinations are given in Table 2.

TABLE 2 VARNISH-DRYING TESTS

	Combination				
	A	B	C	D	E
Temperature, F	140	140	150	150	150
First coat, drying time, hours	1	1.5	1	1.5	1
Second coat, drying time, hours	1	1.5	1	1.5	2

Combination A gave incomplete drying over air-dried fillers. Combination B produced a film dry and hard to the touch, but required (dependent upon atmospheric conditions) 5 to 16 hours' additional air-drying to put it into proper condition for sanding. Combination C produced incomplete drying when using ordinary filler-stains, but was satisfactory over filler-stains especially prepared for oven treatment. Combinations D and E, with the exception of one varnish that showed incomplete drying of the first coat in 1 hour, were both satisfactory. Materials dried in this way were ready to sand when cooled at ordinary room temperatures in about 3 hours, except during the summer hot weather and high humidity, when the drying time at room temperature before sanding was extended to overnight. For operating simplicity combination D was chosen, because it gave a uniform drying period for filler-stains and varnishes. The overnight drying period preliminary to rubbing can be reduced to less than 1 hour by lengthening the oven-drying periods 1/2 hour each and employing forced cooling. Similar results can be obtained by forced air circulation for 1 1/2 hours at room temperature preliminary to rubbing. Again, for uniformity of operation and adaptability to a conveyorized sequence, the 1 1/2-hour drying with forced circulation will be adopted. If the system were not to be conveyorized, the overnight drying would be more economical in view of the fact that the shop works only 8 hours out of the 24

Sequence of Operations

To summarize, the following sequence of operations has been adopted:

1 Apply filler-stain, pad, and wipe
2 Dry 1.5 hours at 150 F
3 Apply first varnish coat and allow to dry under room conditions at least 5 min before placing in oven
4 Oven dry at 150 F for 1 1/2 hours
5 Apply second varnish coat immediately after the first oven-drying period without cooling or sanding
6 Oven dry at 150 F for 1 1/2 hours

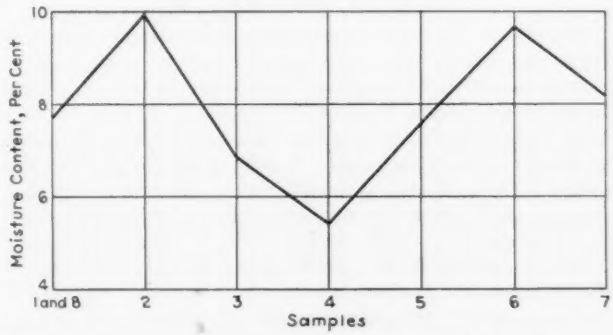


Fig. 4 Change in Moisture Content of 7/16-In. Oak-Plywood Panel During Finishing Cycle

7 Cool to approximately 75 F. (As has been stated, this cooling period varies with atmospheric conditions. It can be greatly reduced by forced cooling with fans. On the other hand, when atmospheric temperature and humidity are high, stock should be allowed to stand overnight.)
8 Sand with 6/0 aluminum-oxide paper. (Since there is no sanding between coats, particular attention must be paid to securing a thorough sanding job at this stage. The greater toughness of varnish as compared with ordinary lacquer sealer or shellac sealer makes it more difficult to sand.)
9 Rub with pumice and oil.

Method of Varnish Application

Several precautions must be observed during the finishing process to insure satisfactory results. All lumber should be uniformly dried to a moisture content that will bring it to the finishing department at about 7 per cent. The filler-stain must be thoroughly dried or the varnish drying will be greatly retarded. Of the greatest importance, however, is the thickness of the varnish film. This is necessarily the resultant of two variables: the percentage of solids in the varnish and the amount of material sprayed per square foot. For purposes of this development varnishes with approximately 54 per cent solids content and a viscosity of E on the Gardner-Holdt bubble-tube viscometer were used.

Many difficulties will be experienced if the proper thickness of coating is not applied. If it is too thin, the surface will not be protected adequately, and it will appear starved or lacking in

depth. On the other hand, if the coating is too heavy, the varnish will blister or wrinkle. Blistering is more likely to occur than wrinkling. It is caused by the presence of volatile thinners imprisoned within a slightly heavier-than-medium coat of varnish. When the parts are placed in the oven at 150 F, the surface of the film immediately begins to harden, and at the same time the thinners expand and attempt to escape. Due to the increasing toughness of the varnish, the expanding thinners stretch the film that is just forming and make blisters that harden on the surface. In some particularly large blisters the pressure generated by the thinners is sufficient to rupture the film while still plastic and leave circular impressions upon the surface. With the materials used in this investigation, wrinkling is caused by the presence of an unusually heavy coating, which is puckered or drawn together by the action of the high temperature. The dried varnish film averages 0.0024 in. in thickness, corresponding to an average coverage of about 350 sq ft per gal on large areas of flat stock. It will be readily understood that the importance of refraining from heavy varnish coats emphasizes the need for good filling work.

Quality of Finished Product

The system proposed has now been in satisfactory operation for a year. From laboratory tests we are convinced that the varnish finishes dried at elevated temperatures as made during these trials are distinctly superior to ordinary finishes in durability and resistance to marring. This conviction is readily verified by ordinary shop tests. A series of samples made by this system for tests of each of the different woods were produced in the shop and have undergone accelerated weathering cycles for a period of several months without failure. The results have indicated conclusively that those finishes that take advantage of the toughness and hardness resulting from the heat treatment of the resins used are distinctly superior to the ordinary run of long-oil varnish finishes.

For simplicity the system described leaves little to be desired in reduction of numbers of raw materials and cost for application of materials. The outstanding merit of the process is that it makes feasible the conveyorization of varnish finishes as well as lacquer finishes.

Discussion

Paul Kennedy.[4] The paper is an interesting addition to the all-too-meager fund of authentic research work on wood finishing. That this system has proved satisfactory in production for a year automatically removes it from the "theory" or "small-scale laboratory" class. Nevertheless, it is an apparent contradiction of certain accepted knowledge. It evidently denies the necessity of maintaining a fairly constant relative-humidity ratio when accelerating varnish drying. It also employs varnishes of synthetic-resin types, which are conceded to be faster setting than natural resin types of varnish and hence more sensitive to case hardening.

It has been an accepted theory that the kiln drying of varnish requires both heat and humidity for safest results. Most chemical reactions are accelerated by heat, and the oxidation of varnish is so stimulated. Humidity is conceded to be a retarder of varnish drying proportionate to its relative percentage in the atmosphere.

In the past there have been several types of kilns employing temperatures usually from 100 to 125, with no humidity control, but moisture was usually artificially introduced into them. This was supplied in the cruder outfits from pans or troughs of water, open steam jets, etc. In the more scientific designs, the air was heated and humidified in a mechanical air conditioner, located outside the kiln, and this conditioned air was automatically regulated for relative humidity and air change. The conclusion was that the humidity kept the varnish film "open," while the heat accelerated the oxidization of the varnish. The humidity would act as a sufficient retarder to make the film dry from the "bottom up," whereas minus this counter-action of humidity the impulse of the heat would be to oxidize and harden at the points of least resistance—namely, the "top of the film."

"Case-hardened" film drying was particularly serious in piano finishing—for example, where practice demanded the application of a number of heavy coats of varnish, and subsequently reflected itself in "tearing," "pulling out" when rubbed, "scaling," "sweating," and "printing." Worst of all, if the first coat was not thoroughly dried, the hardening of the second coat would be much less than that of the first, even with the same film thickness and heat, and this condition became further abnormal with subsequent coats.

These actual production facts are recounted because there is apparently here a paradox, inasmuch as the integrity of the paper under discussion is not to be questioned. There must be some explanation.

The baking of varnishes has long been standard practice, and no field is more conspicuous than the Jamestown metal furniture and trim industry. These companies employ an average varnish-baking temperature of about 175 F, occasionally higher, and sometimes as low as 150 F. It has long been felt that at temperatures from 150 F up, a unique reaction occurs in the hardening of varnish films, which might be termed a "shock action." When varnish films are suddenly exposed to these higher temperatures, a rearrangement of molecules probably takes place.

Further corroboration of this is found in the similar behavior on baking of certain strictly synthetic-resin solutions and the fact that the same varnish when baked (at 150 to 200 F) will have far greater resistance than if air-dried normally or stimulated by the comparatively gentle acceleration of ordinary kiln drying (100 to 120 F). Dr. A. E. Schuh has noted this same phenomenon in his examination of the film produced by the process under discussion. This would seemingly confirm that this "shock action" had occurred with his varnishes. As before stated, 175 F is generally accepted as the best average baking temperature, but synthetic varnishes unquestionably would make a reduction in temperature feasible and even safer practice. These synthetic-resin varnishes outstand for their quick "setting out of dust" even with air dry. This is because they have the faculty of expelling the volatile solvents from the varnish film far more rapidly than natural resins. This feature is entirely distinct from any action they may have in conjunction with the oil of the varnish in the hardening of the film, and is doubly to be noted if the application of the varnish is by spray gun.

When varnish is atomized in the course of spraying, the minutely divided particles dissipate most of their solvent into the air when traveling from the point of the gun to the surface of the work. Hence, the varnish as applied on the work contains only a small percentage of the solvent originally present in it. With synthetic resins this dissipation of solvents is even higher than with old-type varnishes. To the writer's mind this factor is one of the vital drawbacks, in the present exposition, when considering its general adaptability. The authors of the paper have shown their appreciation of this in stressing the point of "proper film thickness of coating to be applied" and their comments on "blistering" and "wrinkling." The wrinkling is a particular factor to be avoided, and the conditions just outlined make synthetic varnishes extremely sensitive to this difficulty.

Much work has already been done in the baking field with this type of varnish, but unless radically modified, the results have

[4] Vice-President, Murphy Varnish Company, Newark, N. J. Mem. A.S.M.E.

been rather discouraging, particularly from the standpoint of wrinkling. On the other hand, this type of varnish has proved extremely successful in the tin-decorating and metal-package field, mainly because the application is by coating machine, and a thin and uniform film can be applied to the tin plate. Of course, this method of application is ideal, because it is mechanical, but even so, carelessness sometimes creeps in, resulting in wrinkled finishes or ones with indifferent adhesion to the tin plate. We have the human element solely to consider in the varnishing of woodwork, and also the factor of a wide range of designs (especially on fancy furniture) which are conducive to the collection of excessive quantities of varnish at certain spots. These sections would most certainly wrinkle on drying.

The authors comment upon the economic superiority to lacquer. It is granted that from the standpoint of material cost and film building the varnish finish has a decided advantage. On the other hand, there are the factors of clean spraying and the "hospital" work that usually involve an expensive repair department. If dirt or specks collect in the varnish, "pull outs" are almost certain when rubbed, and the varnish naturally attracts and retains such foreign matter. There is rarely any difficulty of "pull outs" with lacquer.

Of even greater importance is the refinishing of damaged panels etc. There is usually no requirement for a special section to repair a lacquer film, as only a few minutes are needed to patch up damage, and it can be done on the line. To repair varnish-finish damage, it is often necessary to repeat the original finish system. How these factors offset each other is an important determination of the relative economies.

The writer has purposely not commented upon the points of glue joints, air changes, recirculation, etc., feeling that other discussers may be better qualified and will develop such possibilities of further fact and interest. To sum up, he would call attention to the following points:

1 Employing the temperature of 150 F does not make for a true comparison with the lower temperatures of general kiln-drying practice, because a "shock action" is involved that produces a different type of hardened film, as confirmed by the improvement in durability and resistance noted by Dr. Schuh.

2 The practice admittedly requires close supervision, particularly from the standpoint of uniform application and film thickness.

3 The human element is a critical factor, particularly from the standpoint of spray application, cleanliness, and handling.

4 In the instance cited, the same general type of work is constantly being handled by the human element and might be likened to mass production. With low turnover in help, the spray operators can logically be expected to acquire almost mechanical accuracy in their work.

5 Considering the factors of the human element and the chemically active varnishes employed, it would be the writer's conclusion that this process would have definite possibilities for adoption in such classes of production as sash and trim and certain fixtures where the parts to be coated are generally uniform. He questions, however, whether it would prove economical for general furniture production and especially for high-grade cabinet work where the designs change very rapidly and repair or "hospital" expense is important.

6 Emphasis is stressed on maintaining a wood-moisture content of 7 per cent. Offhand, it would seem that this could be better controlled in mass production than where a number of different designs are employed, or where for sound reasons work proceeds through the factory rather slowly and the wood must be exposed for considerable periods "in the white."

7 The writer believes that the practice should be termed "baking" rather than "accelerated drying," as these two terms have distinctively different meanings, just as they produce films of different character, even with the same varnishes. A typical example of this is the long-service records of baked varnish on steel cars of the Pennsylvania Railroad. The same varnish, air dried, does not approach by many months the durability it has when baked.

8 The thought is suggested of further research work in better adapting such varnish for this particular practice of baking on wood. This would involve the introduction of some slow evaporating solvent into the varnish, which would not be dissipated in spraying, but require some time and heat to expel it. Meanwhile it might act to temporarily give a greater margin of safety against the two factors of danger, wrinkling and case hardening.

H. F. Payne.[5] The authors have carried to a successful conclusion, in a highly practical way, a problem in wood finishing which many said was impossible. The writer was able to show some time ago in a theoretical and small-scale experimental way that, given the wood in correct condition, the right oven facilities, and most important of all, some modern finishing materials, furniture could be finished in one day ready for rubbing the next day.

The finishing materials referred to dry more by polymerization than by oxidation. Heat is a well-known accelerator of polymerization, and polymerized films dry through, thereby eliminating the danger of surface hardening, with its detrimental after-effects of "sweating out." A nicely proportioned solvent is necessary to keep the film open long enough for the heat to penetrate the entire film; the coefficient of heat conductivity of a varnish film is quite low. The authors' statements regarding the necessity of maintaining a limited film thickness illustrates this point. In a thick film the top dries before the heat has reacted on the bottom of the film. The final penetration of heat and the drying of the bottom cause strains to be set up in the film, with consequent wrinkling. In this connection the type of oven is quite a factor. If the satisfactory results obtained by the authors could be attained in a box-type oven, with almost sudden application of the full heat, then, given a conveyor-type oven with gradual application of the heat, it has been the writer's experience that the process is more "foolproof," with less danger of wrinkling and particularly blistering. The authors' finding that blistering "is caused by the presence of volatile thinners imprisoned within a slightly heavier than medium coat of varnish" would suggest that the imprisoned thinners were not sufficiently good solvents for the film. The final evaporating thinner must not be just a "high boiler," but a true solvent, thereby keeping the film open.

The varnishes tested by Dr. Schuh were designed for baking and not air drying. The drying of these varnishes could have been accelerated in a number of ways, such as the manipulation of the cooking process, addition or change of catalysts or driers, and subjection of the drying film to the rays of a carbon-arc or mercury-vapor lamp. The use of the term "baking" would therefore have been more definite than "acceleration."

F. L. Browne.[6] Of the two papers dealing with wood finishing, the one by M. J. Pearce presents the point of view of the maker and seller of finishing materials, while the one by R. H. McCarthy and L. A. Lefcort presents that of a large user of such materials. The user in this case differs from most industrial purchasers of wood-finishing materials in one important respect. The Western Electric Company sells its telephone booths to an affiliated corporation that retains possession of them, and there-

[5] John L. Armitage & Co., Newark, N. J.

[6] Senior Chemist, Forest Products Laboratory, Forest Service, U. S. Department of Agriculture, maintained at Madison, Wis., in cooperation with the University of Wisconsin.

fore thinks of the durability of the finish in terms of maximum years of service at minimum expense for maintenance rather than in terms of initial appearance and sales arguments to impress purchasers. For that reason the conclusions reached by McCarthy and Lefcort deserve unusually serious consideration.

A reading of these two papers in sequence inspires comments from a point of view not specifically considered in either one of them. Pearce's paper surveys the great variety of finishing materials that are made available by their manufacturers and shows how their numbers have multiplied in recent years. It might reasonably be supposed that a new material distinctly better than an old one would force the old one from the market, while a new material not distinctly superior would make little headway. Experience shows, however, that new finishing materials often win a place for themselves, but rarely displace the old ones. In explanation of this curious tendency it is sometimes suggested that new materials are developed to meet an increasing diversity in requirements of the users. To the writer, at least, the argument is unconvincing; it seems more likely that the chief urge toward multiplication lies rather in the desire of raw-materials manufacturers to find new outlets for such basic products as nitro-cellulose, synthetic resins, and organic solvents and in the eagerness of makers of finishing materials to cater to the caprices of a market in which standards of value are not yet well defined.

To the woodworking industries this multiplication of finishing materials may be more of a menace than an opportunity. New materials usually offer certain advantages, but so far the improvements have nearly always been accompanied by serious disadvantages that must be taken carefully into consideration. Not only must the merits of the new material itself be determined, but its compatibility with the other finishing materials with which it is to be used must be established. At best, wood finishing is a complicated operation beset with many pitfalls. When the field of choice is broadened, the problem of wise selection becomes still more complex and the chance of an unwise choice is greater.

In revamping a finishing process the woodworker faces a major difficulty in the fact that a most important characteristic, the durability of the finish, has not heretofore been measurable by accepted laboratory or practical tests and must usually be determined by experience. In other words, the woodworker's customers often make his durability tests for him, and not infrequently they are also the durability tests for the maker of the finishing materials. Such testing by the ultimate consumer is economical enough as long as the results are satisfactory, but gives rise to a heavy charge against good will when a test turns out unfavorably. Since each change in a finishing process involves the hazard of a serious loss of reputation, the woodworker must be very conservative about making changes, and consequently has good reason to view the rapid development of new materials with some alarm.

The woodworker traditionally has relied upon craftsmanship for wood-finishing operations. Craftsmanship, however, has been breaking down rapidly under modern economic conditions and is being replaced only slowly by engineering control. For technical information about wood finishing the woodworker has turned more and more to the makers of finishing materials, so much so that the selling of finishing materials tends to become incidental to the furnishing of engineering service, with the cost of the service carried as overhead in the price of the materials. To the materials maker this arrangement is desirable as long as it remains profitable to him, because it makes the woodworker more dependent upon him and gives the large materials maker a decided advantage in competition with his smaller rival, who is less generously supplied with engineers and chemists. In the writer's opinion the woodworking industries will not gain the full benefit of technical service in dealing with their wood-finishing problems until they employ their own engineers more generally than is now the case.

Both Pearce and McCarthy and Lefcort show clearly that the dominant trend in wood-finishing development has been toward faster production schedules. Apparently durability has sometimes been sacrified to attain that end. The materials maker has sought to hasten finishing operations chiefly by developing products that dry more rapidly at ordinary room temperatures, which has led to the wood lacquers that yield a finish inferior to varnish in certain important respects. It is natural that the materials maker should approach the problem from that direction because he is interested primarily in the sale of materials rather than in redesigning equipment and processes that would be equally suitable for some other maker's materials.

McCarthy and Lefcort, however, show how the application of engineering knowledge from the woodworker's point of view achieves rapid production and simplification of procedure without resort to lacquer by designing equipment to take advantage of a property of some modern varnishes. The finishing process is accelerated so greatly that the production schedule can be set in accordance with convenience in plant operation rather than by the time required for coatings to dry.

Development of the new process involved three steps that required engineering knowledge rather than craftsmanship:

1 Establishment of laboratory tests to determine the relative serviceableness and durability of wood finishes
2 Design of a drying oven and determination of temperatures necessary to achieve sufficient acceleration of drying
3 Proof that the heat treatment is not injurious to the wood or to glued joints.

Varnish-drying kilns are by no means new in wood finishing, of course, but the temperatures and degree of acceleration attained are distinctly new, at least for articles as large and complicated as telephone booths. Furthermore, in varnish kilns of the past, control of humidity has always been considered necessary, usually in the belief that the rate of drying of varnish depends upon humidity as well as upon temperature. The writer has suspected, however, that the principal reason for controlling humidity was to hold the moisture content of the wood constant. The new process requires so little time that humidity control is not necessary to maintain the moisture content of the wood and is evidently unnecessary to secure thorough drying of varnishes of the type for which it is designed.

Speed in production is attained by McCarthy and Lefcort, not only by hastening varnish drying, but also by simplification of the finishing process. Staining and filling are done in one operation, two coats of varnish are applied, the surface is sanded, and finally rubbed with pumice and oil. When the number of operations are minimized, it is of course essential that each one of them be performed in exactly the right way, and therefore closer control of material, equipment, and operation is necessary than under a more complicated finishing schedule.

When engineering skill is applied more generally by the woodworking industries in dealing with their wood-finishing problems, the multiplicity of materials and complexity of processes will, in the writer's opinion, become greatly simplified. The notion that each woodworker's finishing problems are unique will give way to a realization that there are relatively few basic operations, the various combinations of which produce the different finishes in use. Means will be developed to determine unambiguously the relative merits and costs of the available finishes, so that the less desirable ones can be eliminated from practical consideration. New materials must then demonstrate their right to consideration, and if successful will usually displace the less satisfactory

ones. Changes in process rather than in materials may then play a more important part in future developments. The present uncomfortable plethora of materials may be reduced with advantage to the woodworker, the materials maker, and to the ultimate consumer.

Authors' Closure

By Mr. McCarthy. Mr. Kennedy has emphasized the use of synthetic resins in forced drying. As was stated, one of the three satisfactory varnishes had a natural resin base. While determining the amount of solvents liberated in the oven in order to serve as a basis for avoiding any possibility of combustible mixtures, the authors incidentally secured the following data concerning the mechanism of film drying and hardening. Test metal panels were spun and sprayed with a varnish of known solids and thinner content and were dried at room temperature for 30 min, and then placed in a laboratory oven for 60 min and weighed periodically. It was expected that the panels would at first decrease in weight due to loss of solvents, and then increase in weight due to oxygen absorption. Morrell[7] has reported that drying oils of the type used in varnishes increase their weight from 13 to 17 per cent upon exposure to air at room temperatures.

TABLE 3

	Film spun		Film sprayed	
Time, min.	Weight	Per cent	Weight	Per cent
0	0.71	100.0	1.21	100.0
5	0.54	80.3	1.08	89.3
10	0.50	70.4	1.01	83.5
15 (at 78 F)	0.45	63.3	0.94	77.7
20	0.42	59.2	0.89	73.6
25	0.39	54.9	0.84	69.4
30	0.37	52.1	0.81	67.0
60 (at 150 F)	0.31	43.7	0.53	43.8
90 (at 150 F)	0.31	43.7	0.53	43.8

Varnish analysis: Solids content, 43 per cent; thinner content, 57 per cent.

The data in Table 3 show that the varnish decreased in weight as expected until about 2 per cent higher than the value indicated by the initial analysis for solids. Apparently then the film had become sufficiently hard for sanding operations when only about one-third of the oxygen absorption that might be expected from a linseed-oil vehicle had taken place. The data indicate that completion of the oxidation process is not important to the hardening of the film, from the production standpoint.

The same data shed light on the much-discussed question of what happens when spraying varnish. Rather than retaining "only a small percentage of the solvents," practically all of it is retained in the passage between the spray gun and the panel, since the loss of weight of the film on the panel corresponds practically with the original solvent content shown in the varnish analysis.

Mr. Kennedy is quite correct in concluding that the accelerated process is not foolproof. Nevertheless it has been used successfully on a rather wide variety of special, high-grade woodwork in addition to the telephone booth, work requiring three and four coats of varnish and highly polished surfaces.

As Mr. Payne has pointed out, the word "accelerated" as used in this article is perhaps misleading, since it seems to have a technical meaning in connection with the formulation of the varnish. The authors intended it to describe a process which had been made faster.

The authors concur heartily with the opinions advanced by Dr. Browne. Wood finishing, at best, seems bound to be a highly complicated process because of the extreme variations or complexity of the materials involved—i.e., wood variability in structure, condition, surface preparation, color and variability of the natural ingredients of stains, sealers, and varnishes. A given finished appearance may frequently be secured by a hundred different procedures and formulations, some producing good results, some otherwise. Each of the different procedures is defended persistently, although many of the results are never examined critically by the producers and purchasers or the defects which develop are taken for granted. Thus there is thrown about the art a veil of complexity and deep mystery which even the large number of variables does not justify.

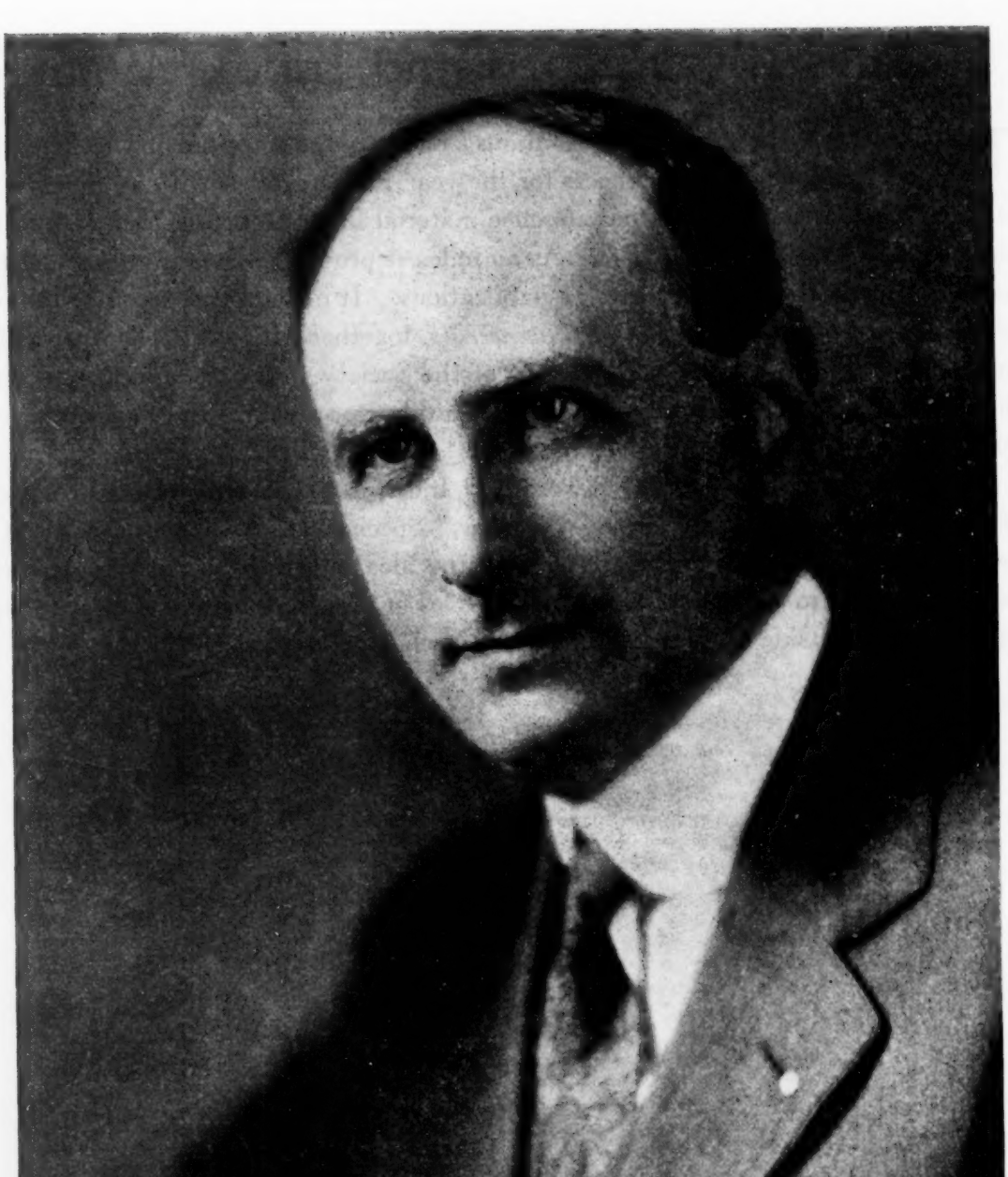
[7] "Varnishes and Their Components," R. S. Morrell, p. 80.

A.S.M.E. Record and Index, 1933

THIS section of the Transactions contains a record of the activities of The American Society of Mechanical Engineers for the year 1933, together with an index of its publications. It includes all the reference material likely to be of permanent value and of interest to future searchers. As an index it provides a means of locating the technical information in the Society's publications. It includes the separate indexes of the Transactions and *Mechanical Engineering*, together with a miscellaneous index to reports or other technical publications of the Society that have appeared during the year and a list of papers issued in mimeographed or similar form for meetings in 1933, but not published.

A considerable portion of the Record and Index is devoted to memorial notices of members who have died during the year. Special care has been given to the preparation of these brief biographies with the idea in mind that fitting attention should be given to significant events in the lives of deceased members, even though they may fall outside the immediate field of mechanical engineering.

COMMITTEE ON PUBLICATIONS



Harris & Ewing

ANDREY A. POTTER

PRESIDENT OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
1932–1933

Andrey A. Potter

ANDREY A. POTTER, dean of the Schools of Engineering of Purdue University, Lafayette, Ind., served as president of The American Society of Mechanical Engineers for the term 1932–1933.

Dean Potter was born in Vilna, Russia, on August 5, 1882, the son of Gregor and Rivza Potter. He came to the United States in 1897 and in 1903 was graduated from the Massachusetts Institute of Technology with the degree of B.S. He received his doctor's degree in engineering from Kansas State College in 1925.

Upon his graduation from M.I.T. Dean Potter entered the employ of the General Electric Company, at Schenectady, N. Y., during the early stages of the development of the steam turbine. He resigned in 1905 to become a teacher of engineering with the title of assistant professor of mechanical engineering at Kansas State College, where he was advanced to a full professorship in 1910 and to the position of dean and director of the engineering experiment station of that institution in 1913. Since 1920 he has been associated with Purdue University as dean of the Schools of Engineering, director of the engineering experiment station, and professor of power engineering.

From 1924 to 1931 Dean Potter also was advisory editor to Ginn & Co., and from 1928 to 1930 consulting expert to the Bureau of Education of the United States Department of the Interior. He was an associate member of the United States Naval Consulting Board from 1917 to 1919 and district educational director for the Committee on Education and Special Training of the War Department during 1918 and 1919. He is a pioneer in personnel work as applied to engineering colleges. Since 1910 he has served as a consulting engineer for utilities and industries in the field of power engineering.

Dean Potter became a member of the A.S.M.E. in 1912. His services to the Society include membership for five years on the Committee on Relations With Colleges; membership for three years on the Main Research Committee; chairmanship of the Indianapolis Section of the Society; representative on the Science Advisory Committee for the Century of Progress Exposition, Chicago, Ill., 1933; and representative for three years on the American Engineering Council.

In 1924–1925 Dean Potter was president of the Society for the Promotion of Engineering Education; in 1919–1920 president of the Kansas Engineering Society; in 1922–1923 president of the Indiana Engineering Society; in 1921–1922 vice-president and acting president of the Association of Land Grant Colleges; and in 1929–1930 chairman of the Engineering Section of the Association of Land Grant Colleges and Universities. From 1920 to 1933 he was a member at large of the Prime Movers Committee of the National Electric Light Association.

Dean Potter is an honorary member of the American Association of University Professors, a fellow of the American Association for the Advancement of Science, and a member of the honorary societies of Tau Beta Pi, Sigma Tau, Phi Kappa Phi, Sigma Xi, and Scabbard and Blade. He is the author of "Farm Motors," "Elements of Steam and Gas Power," "Engineering Thermodynamics," as well as of numerous articles which have been published in engineering and educational magazines.

Whither Engineering?

By A. A. POTTER

THINKING INDIVIDUALS and progressive social groups must reflect on the past and plan for the future, must take stock of their objectives and seek improved goals. The engineer must analyze his work, study trends, and adjust himself to changing conditions. He has in the past said little about his work and has infrequently concerned himself with public questions. It is timely for us to inquire, whither engineering? What are the thoughts, plans, and attitudes affecting our profession? If we know what is going on about us and what to do, let us consider the means by which we may best apply our knowledge and experience for the public good.

Our times are distinctive, owing to the introduction of science and engineering in the everyday affairs of everyday people. Whether civilized man likes it or not, he is destined to live and to work in an environment affected to an increasing extent by science and technology. While some blame the engineer for the sufferings of our times, the fact is that scientists and engineers should really be credited with the delivery of humanity from the so-called Malthusian doom. Whereas in past civilizations man had difficulty in producing enough to maintain himself above the level of mere existence, the engineer, by applying science to practical uses, assures an abundance of the world's goods for all, given only an effective mechanism of exchange. Present ills are not due to a lack of material wealth to satisfy human needs, but to our inability to distribute it rationally. However, our present problems, serious though they be, should not prove as difficult as those resulting from scarcity and want.

Also, it should be recognized that while the work of the engineer in improving methods of production has reduced employment in some specific cases, the engineer is primarily a creator both of wealth and of opportunities. By proper application of science he has created, during the past 65 years, where nothing was before, such giant industries as those which manufacture automobiles, typewriters, radios, talking machines, airplanes, and telephones, as well as new utilities which are concerned with electric communication, electric transportation, and electric light and power. These are creations, not developments. These are not displacing labor, but are adding new opportunities for profitable employment and happy careers for millions of people. It is reasonable to expect the engineer to develop new industries in the future as he has in the past, creating new opportunities, new jobs, and new careers. He knows what developments are needed and is able to direct the scientific researches and the manufacturing methods to produce results of value.

The engineer has contributed to the general welfare by reducing drudgery, by providing new entertainment, by saving time, and by increasing material possessions. His contributions are being fully appreciated, as to an increasing extent it is realized that America's predominant industrial position is due largely to the engineer. There is a general feeling, however, that he can make additional contributions by taking a greater interest in social and economic problems. Not that the engineer is a miracle worker who can bring about, through his own efforts, a perfect social order, but it is felt that, dealing as he does with facts rather than opinions, he should be able to help with the social problems affecting us. He is skilled in the proper use of science and machinery and is in a position to forestall movements likely to hinder invention and discovery or retard the introduction of improved methods. He may even be able to contribute, by his straight thinking, to a better solution of the social and economic problems which at present confront us.

The engineer has substituted knowledge for guesswork in dealing with technological problems. Has he the wisdom to apply the same straight thinking in his dealings with human relations, social welfare, economics, and finance? It is to be hoped that we are now standing on the verge of a new and better epoch, a period during which the actions of engineers as individuals and as members of a great profession will be influenced still more by clear thinking and by true measurements, and less by irrational opinions, prejudices, traditions, and precedents.

Even though the engineer may not be able to solve, offhand, the problems of human organization, where supposed social experts have apparently failed, it is high time that he seriously consider them. Having developed the necessary technique for almost unlimited production of wealth, he can no longer afford to stand aside while financiers, promoters, and politicians so tragically bungle the problem of its distribution. Primarily, engineering must deal with the technological problems of research, development, design, production, testing, and operation, yet the engineer is becoming more and more interested in finance and is to an increasing extent the directing head of industry and of public works; his success in financial, administrative, and executive posts may be attributed to his talent for dealing with the human phases of industry. He is beginning to interest himself in and to appreciate social and economic trends and is already taking his place among the leaders of our times. Engineers must think more, write more, and talk more on social and economic questions in order to assist in the solution of the major problems confronting humanity.

At the dedication of the Engineering Societies Building in 1907, Dr. Hadley, president of Yale University, said: "Let a man or group of men arise who can add to their technical knowledge a readiness to use that knowledge in the public service, and the people will be ready to put them in charge of affairs and follow their lead." If our profession is to maintain public leadership, we must conscientiously dedicate ourselves to public service and take our rightful places in the affairs of men. This we can do by giving major attention to the following:

Active Interest in Government and Politics

The engineer must actively interest himself in government and politics, realizing that his contributions to society may be lost without stable government. He must work for good government and for obedience to law, and must apply clear thinking and courageous action to public questions. Furthermore, members of our profession must stand for elective public offices and campaign on well-thought-out policies of action, because a democratic state depends upon participation by the ablest of its citizens in political affairs. Perhaps the greatest contribution of engineers, politically, will come through their familiarity with the technological problems which arise in our civilization, since the government of our cities, states, and nation is very largely concerned with engineering matters.

In these days when unrest is too prevalent to be ignored, the greatest need of this or any democratic government is men and women in political office who think logically, conclude honestly, and act courageously. There is a grave danger that in the present

Presidential Address delivered at the Annual Meeting, New York, N. Y., Dec. 4 to 8, 1933, of The American Society of Mechanical Engineers.

popular and indiscriminating demand for economy and for reduced taxes our cultural advance, our educational system, and our social institutions may be seriously crippled. When we realize that the total cost of maintaining activities indispensable to national welfare, such as education, public health, national defense, good roads, police protection, and scientific research, is less than that which we spend for amusements, cosmetics, and tobacco, we gain perspective in social values. In many localities education, which is our hope for the future and our main bulwark against unwise changes, is being sacrificed to extravagances in governmental activities. This can be remedied only by insisting upon a higher type of political office holder. Education cannot be postponed or its effectiveness reduced without great penalties in the future. It should also be recognized that the consumer demand of two and a half million Americans is fixed every year, and the schools influence the upper limit of nearly half of these; to close or to impoverish our schools is to dry up our markets for a generation at least, a market which can be developed through schools—more schools—better schools. Industry, jobs, and prosperity depend upon customers and the will to have; the determination to be a customer depends upon education. Similarly, reduction of the essentials for national defense, police protection, public health, and research will prove costly in the days to come. The engineers owe it to themselves and their country to utilize their habits of analytical thought in the solution of public questions. In this they must maintain the same regard for truth and fairness as when they deal with their professional problems.

Contributions to Social Welfare and Economic Readjustment

Before the engineer can take his rightful place in the affairs of the nation and contribute most effectively to social welfare and economic readjustment, we, as a professional body, must take an active and leading part in engineering education, insisting on definite educational standards as a qualification for membership. As engineers we have taken little active part in shaping or in directing engineering education, and our engineering schools have not had the serious backing of the engineering profession. What aid has been given to engineering education by engineers has been largely individual and unofficial, and has not represented the concerted thought or the unified action of the profession. It may be expected that the Engineers' Council for Professional Development, representing as it does the leading engineering societies and with its program definitely focused upon the educational preparation of the engineer, will take a definite stand which will insure that the engineer of tomorrow will be fully prepared to take advantage of the new opportunities and responsibilities which are bound to be his. The Engineers' Council for Professional Development provides an opportunity for The American Society of Mechanical Engineers, and for the other major engineering societies as well, to take a leading part in improving the educational preparation of the engineer. However, it must be realized that reports and recommendations of this Council will not improve the engineering profession without the unified cooperative action of the engineering societies, engineering colleges, and licensing agencies of the various states. The American Society of Mechanical Engineers will, I trust, lead in backing the reports of the Engineers' Council for Professional Development. In the end we may expect that recognition of professional attainment on the basis of education and actual achievement, as tested by examinations, will be substituted for the present haphazard methods of certifying to Society membership. Then we shall find that membership in a society such as ours will unqualifiedly admit to good standing in the great profession of engineering.

At the annual meeting of this Society exactly forty years ago, E. B. Coxe, in his presidential address, decried the tendency of engineering educators to magnify the differences between civil, electrical, mechanical, mining, and other branches of the engineering profession. Unfortunately, this condition still persists, although in practice the work of the engineer is organized largely along the functional lines of research, design, production, operation, and sales, rather than the fields indicated by our engineering societies. Undoubtedly there is still a definite need to revise the engineering programs of study so that the young engineer may have a keener interest in the engineering profession as a whole.

Engineering education is now confronted with two radically different requirements. There is, on the one hand, a very definite demand for engineers particularly well prepared in science, mathematics, and technology. At the same time, engineering colleges are urged to give more attention to social sciences. The demand for a more thorough scientific preparation comes from a realization that the major achievements of our profession can only result when the engineer has sufficient preparation to make full use of the foundations laid by pure science, mathematics, and engineering. The pressure brought to bear upon engineering colleges for more social science comes from an appreciation of the fact that the technical graduate lands flat-footed in a world of men with whom he must cope and cooperate, and sadly needs such knowledge as is available regarding the organization of men into society.

In considering the preparation of the engineer, or, in fact, any educational problem, we must keep two things clearly in mind: In the first place, education is an opportunity to gain knowledge, resulting, we like to think, in greater wisdom. But education is not a guarantee of happiness or wealth or position or security or anything else. In the second place, no amount of education or special training will make two people of unequal talent equal. Leaders are not developed from people naturally inclined to follow. The most any educational program can do is to assist the students to attain the highest level which their natural endowment allows. Mentally and temperamentally, men differ far more than in their physical make-ups, and these differences, being inborn, cannot be overcome by education. When by a fortunate combination of natural ability, environment, and industry a technical graduate proves a leader, a driving force in our profession, we are all apt to class as failures all those less fortunately equipped who suffer by comparison, and are inclined to ascribe the inability of some to reach their goal, or the position desired for them by relatives or friends, to the educational system. Straight-thinking people will not make this error, but will remember that not every freshman entering college is even potentially endowed with the leadership of a Napoleon, the technology of a Steinmetz, the scientific prowess of an Einstein, the inventive genius of an Edison, or the wisdom of a Solomon; most people must derive such consolation as may be had from the reflection that society needs a goodly supply of earnest, conscientious, industrious, painstaking followers who, after all, assume the burden of the world's work. These qualities, together with worthy ideals of character and conduct, we may properly expect educators to stress in all their contacts with students. We may also expect such an environment in college that the student will be imbued with a keen interest in knowledge and a continuing desire for its achievement.

Aside from preparation for a career, engineering education is the modern type of general liberal education, acquainting the learner, as it does, with the processes, devices, and methods which make our civilization distinctive. Cultured people are those who understand their environment—the world in which they live—and no type of education so directly assists the individual to understand his surroundings as engineering educa-

tion, which familiarizes one with the machines and the techniques of the modern world. So we of the engineering profession as well as the public at large are interested in making engineering education more effective as a preparation for happy living. As Herbert Spencer stated, "To prepare us for complete living is the function which education has to discharge." This applies to engineering as well as to other types of education.

Some of the graduates of engineering colleges should be given encouragement to continue their preparation for engineering as a career, but a much larger number should be assisted in entering non-engineering vocations with engineering education and possibly even with some engineering practice as a foundation. There is a definite need for clear-thinking and fact-finding technique in the professional, economic, business, and political fields.

A.S.M.E. Stand on Present-Day Problems

Present conditions make new demands upon a professional group. Our own society must assume leadership in the field of mechanical engineering, taking a definite stand in connection with the problems which are confronting our country and our profession. To be specific:

(*a*) The American Society of Mechanical Engineers must discourage the present movement to replace machine processes by hand labor. During recent months an effort has been made to decry capital expenditures, to urge the substitution of human and animal labor for mechanical power and machinery, and to hinder economical methods of production. The present will not be improved and the future will be greatly impoverished if this type of thinking prevails. Surely, we cannot have too many or too effective devices for relieving men from drudgery or for supplying human beings with more comforts. It may be recalled that during the years of greatest prosperity millions of people, even in this most favored land, lacked many of the material comforts which are essential for well-being and happiness. We are far from the saturation point in our need to utilize science and engineering.

(*b*) Our Society must take the lead in the stabilization of the so-called durable-goods or capital-goods industries. In 1928 more than half of the industrial workers of the country were engaged in these industries, which comprise tool making, machine making, power production, equipment manufacture, construction, and the development of the heavy industries such as iron and steel, coal, copper, and oil. Activity in these industries is absolutely essential to insure the welfare of a highly industrialized land such as ours. At the same time, however, care must be taken not to have industry over-tooled for the production of the market demand for consumers' goods, and it is clearly within the province of a Society such as ours to aid in the preparation and dissemination of accurate statistics regarding production capacity and consumer demand.

(*c*) The American Society of Mechanical Engineers must lead in a policy of rational distribution of the leisure dividends of quantity production. To this end it should encourage investigations of the shorter work week, not only for the manual workers but for the clerical, engineering, and executive staffs as well. Perhaps consumer demand may be fully satisfied with fewer man-hours of work and the resulting leisure more equitably divided than is now the case. Also we should cooperate individually and through our Sections with the adult-education programs in the different parts of our land and should definitely become exponents of cultural advance.

(*d*) Our Society should give even greater attention than it does now, in its publications and in its meetings, both national and local, to discussions and papers which are definitely concerned with the scientific, technological, social, and economic problems of the present industrial age. Among the most important of these are the art of industrial management, correlation of agriculture and industry, re-estimating fundamental social values, and aesthetic objectives in engineering design. The engineering society must be constantly contributing to public welfare and must be working to widen the horizon and to enlarge the usefulness of the engineer, so that he may be better prepared to render broad and effective service.

(*e*) The American Society of Mechanical Engineers must keep in close touch with science, as the great engineering achievements of the future will definitely depend upon more scientific knowledge. It must aid in scientific research, in the development of rational designs, in national and international standardization, and in the perfection of labor-liberating machinery to ease the burdens of everyday living.

Unifying the Engineering Profession

Said Sir Francis Bacon more than 300 years ago: "I hold every man a debtor to his profession; from the which as men of course do seek to receive countenance and profit, so ought they of duty to endeavor themselves by way of amends to be a help and ornament thereto." Can we today better define the relation of the individual to his profession? Can we as a society hold to a higher objective than the fostering of this attitude?

It is gratifying that our own society has always reviewed its own accomplishments with a critical eye and has promptly moved forward in the line of progress. Among the recent advances may be mentioned its leadership in coordinating the joint activities of the engineering profession, its interest in human and economic problems, evidenced by the Society publications, its activity in the stimulation of the capital-goods industries, and its initiative in organizing the Engineers' Council for Professional Development.

The engineering profession must be unified in its viewpoint, thought, and action. Engineers must assume professional status individually and collectively. An individual becomes a professional man when he regards himself as a public servant first and as a private practitioner second. A society like The American Society of Mechanical Engineers becomes a professional body when, and only when, the conscious object of public service underlies all of its actions. Codes of professional conduct and concerted action in matters of general interest will help to focus the attention of the members of a professional group upon the public aspects of their calling; perhaps the individualistic attitude of some persons may even be changed by the effective public opinion of their peers, thus unifying the engineering profession in its efforts for the public weal.

Until recently the engineer has been an individualist who surveyed, designed, constructed, tested, and operated public works, machines, and tools, and gave little attention to the wider significance of his work or to his social responsibilities. He has worked within the confines of his own field and has had no sense of professional responsibility; neither has he appreciated the strength potential in cooperation among engineers. He has usually joined a professional society when it appeared that his immediate interest would be served, and this in turn has compelled the engineering societies to devote first attention to serving the member's immediate needs, that is, to provide for his technological advance. Gradually this attitude has changed—has given way to the larger conception of an engineering profession; and it only remains for us to consolidate the gains, realizing in formal unified organization the progress already achieved, and to prepare for a rapid further advance.

The work of the engineer is not of recent origin; his contributions date back to the earlier periods of human history, and antedate some callings which have long enjoyed professional

standing. Members of the professions of religion, law, and medicine received public recognition when among themselves they realized the public character of their work and developed principles of action or codes of ethics. Engineers must think, act, and appear as a unified and well-coordinated group; a well-knit and well-integrated solid body, not as a collection of individuals, each going his separate way; nor can the profession be properly represented by several distinct societies going their separate ways. This is the time for definite action to perfect the unity and solidarity of our profession, and I am glad that the Founder Engineering Societies are now working on this important problem. We may expect that their efforts will result in some form of central agency, competent to deal with the joint activities of all engineering societies and to pass upon questions of general import without, however, interfering with the technical field or with the autonomy of the various engineering groups. Engineers have been credited with outstanding success in organizing industrial projects of great magnitude; let us now utilize some of the ability displayed in this work to unify and coordinate our professional advance.

Unifying the engineering profession will enable us to work more effectively on public questions of truly national scope. A central national agency should soon be supplemented by regional or state joint committees to assist governmental authorities in the solution of economic, social, political, and other policy-planning programs. Such groups, composed of engineers, informed and alert, can do the things needful to help governmental agencies in dealing with public questions and particularly with those which have technological aspects.

Whither Engineering?

Whither Engineering? Inevitably toward public service—toward the ideal that the engineer is a public official, charged first with a public function, as a lawyer is an officer of the court, responsible for its dignity and honor. Inevitably toward a wider responsibility—toward the ideal that he is responsible for maintaining that balance in the social structure which modern conditions have sometimes so violently upset. Inevitably toward active participation in political affairs where clear thinking and courageous action are badly needed. Inevitably toward a higher type of education—toward the ideal that men shall appreciate their time and understand its trend. Inevitably toward a more closely knit professional organization and a more pronounced professional consciousness—toward the ideal that the wider duties and greater responsibilities of our calling may be met promptly and effectively.

The American Society of Mechanical Engineers

THE personnel records and other information presented in this section of the Record and Index refer, except where otherwise noted, as in the case of Local Section and Student Branch chairmen, to the official year beginning in December, 1932. Such information is purely of historical nature and is included here as a permanent record. Further information may be secured from the Office of the Society.

Officers and Council[1]

Dates in parentheses denote expiration of terms

PRESIDENT

A. A. POTTER

PAST-PRESIDENTS

Terms expire in December

WILLIAM L. ABBOTT (1933)
CHARLES M. SCHWAB (1934)
CHARLES PIEZ (1935)[2]
ROY V. WRIGHT (1936)
CONRAD N. LAUER (1937)

VICE-PRESIDENTS

Terms expire December, 1933

CHARLES M. ALLEN
FREDERICK H. DORNER
ROBERT M. GATES
WILLIAM B. GREGORY

Terms expire December, 1934

HAROLD V. COES
JAMES D. CUNNINGHAM
C. F. HIRSHFELD

MANAGERS

Terms expire December, 1933

WILLIAM L. BATT
HAROLD L. DOOLITTLE
HERBERT L. WHITTEMORE

Terms expire December, 1934

ALEXANDER J. DICKIE
EUGENE W. O'BRIEN
HARRY R. WESTCOTT

Terms expire December, 1935

ALEX D. BAILEY
JOHN A. HUNTER
ROBERT L. SACKETT

TREASURER

ERIK OBERG

SECRETARY

CALVIN W. RICE

EXECUTIVE COMMITTEE OF COUNCIL

A. A. POTTER, *Chairman*
ROBERT M. GATES
HAROLD V. COES
WILLIAM L. BATT
HARRY R. WESTCOTT
CALVIN W. RICE, *Secretary*
ROY V. WRIGHT, *President's Representative*

Advisory Members: Chairmen of the Finance Committee, the Local Sections Committee, and the Professional Divisions Committee.

CHAIRMEN OF STANDING COMMITTEES

Representatives on Council but without vote

Finance, K. L. MARTIN, *Chairman*
Meetings and Program, J. W. PARKER
Publications, L. C. MORROW
Membership, HOSEA WEBSTER
Professional Divisions, P. T. SOWDEN
Local Sections, J. M. TODD
Constitution and By-Laws, T. C. MCBRIDE
Awards, K. H. CONDIT
Relations With Colleges, D. B. PRENTICE
Education and Training for Industries, J. T. FAIG
Library, W. M. KEENAN
Standardization, W. S. MONROE
Research, W. H. FULWEILER
Power Test Codes, F. R. LOW
Safety, H. W. MOWERY
Professional Conduct, A. G. CHRISTIE

[1] Special Committees of Council, page 11.
[2] Deceased, October 2, 1933. Term completed by William F. Durand.

EXECUTIVE SECRETARY

C. E. DAVIES

EDITOR

G. A. STETSON

ASSISTANT SECRETARIES

ERNEST HARTFORD
C. B. LEPAGE

Standing Committees

Dates in parentheses denote expiration of terms

FINANCE

K. L. MARTIN, *Chairman and Representative on Council* (1933)
J. H. LAWRENCE, *Vice-Chairman* (1934)
E. A. MULLER (1935)
WALTER RAUTENSTRAUCH (1936)
D. ROBERT YARNALL (1937)

Council Representatives { W. L. BATT, H. V. COES

MEETINGS AND PROGRAM

J. W. PARKER, *Chairman and Representative on Council* (1933)
C. P. BLISS, *Vice-Chairman* (1934)
E. C. HUTCHINSON (1936)
R. I. REES (1935)
H. N. DAVIS (1937)

PUBLICATIONS

L. C. MORROW, *Chairman and Representative on Council* (1933)
S. F. VOORHEES, *Vice-Chairman* (1934)
W. F. RYAN (1936)
S. W. DUDLEY (1935)
M. H. ROBERTS (1937)

A. J. DICKIE, E. B. NORRIS, E. L. OHLE } *Advisory Members*

(*Personnel of Special Committee, p. 11*)

MEMBERSHIP

HOSEA WEBSTER, *Chairman and Representative on Council* (1933)
O. E. GOLDSCHMIDT (1934)
R. H. MCLAIN (1936)
H. A. LARDNER (1935)
C. L. DAVIDSON (1937)
H. W. BUTLER, *Advisory Member*

PROFESSIONAL DIVISIONS

P. T. SOWDEN, *Chairman and Representative on Council* (1933)
C. B. PECK, *Vice-Chairman* (1934)
WALTER SAMANS (1936)[3]
W. A. SHOUDY (1935)
G. B. PEGRAM (1937)

(*Chairmen of Professional Divisions' Executive Committees and Personnel of Special Committees, p. 11*)

LOCAL SECTIONS

J. M. TODD, *Chairman and Representative on Council* (1933)
J. W. HANEY (1934)
W. W. MACON (1936)
W. L. DUDLEY (1935)
R. E. W. HARRISON (1937)

(*Chairmen of Local Sections' Executive Committees, p. 11*)

CONSTITUTION AND BY-LAWS

T. C. MCBRIDE, *Chairman and Representative on Council* (1933)
R. S. NEAL (1934)
P. R. FAYMONVILLE (1936)
H. H. SNELLING (1935)
J. A. HALL (1937)

AWARDS

K. H. CONDIT, *Chairman and Representative on Council* (1933)
F. L. EIDMANN (1934)
D. B. RUSHMORE (1936)[4]
W. L. BATT (1935)
R. C. H. HECK (1937)

[3] Resigned. K. H. Condit appointed, September, 1933, to complete term.
[4] Resigned. D. S. Kimball appointed, July, 1933, to complete term.

RELATIONS WITH COLLEGES

D. B. PRENTICE, *Chairman and Representative on Council* (1933)
E. F. CHURCH, JR. (1934) — E. W. O'BRIEN (1936)
W. L. ABBOTT (1935) — R. V. WRIGHT (1937)

(*Student Branches and Honorary Chairmen, p. 12*)

EDUCATION AND TRAINING FOR THE INDUSTRIES

J. T. FAIG, *Chairman and Representative on Council* (1933)
H. S. FALK (1934) — S. S. EDMANDS (1936)
C. F. BAILEY (1935) — R. H. SPAHR (1937)

LIBRARY

W. M. KEENAN, *Chairman and Representative on Council* (1934)
G. F. BATEMAN (1933) — G. F. FELKER (1936)
E. P. WORDEN (1935) — THE SECRETARY, CALVIN W. RICE, *Ex-Officio*

RESEARCH

W. H. FULWEILER, *Chairman and Representative on Council* (1933)
A. D. BAILEY (1934) — D. B. BULLARD (1936)
G. M. EATON (1935) — C. R. RICHARDS (1937)

STANDARDIZATION

W. S. MONROE, *Chairman and Representative on Council* (1933)
EARLE BUCKINGHAM (1934) — ALFRED IDDLES (1936)
C. W. SPICER (1935) — L. A. CORNELIUS (1937)

POWER TEST CODES

F. R. LOW, *Chairman and Representative on Council* (1935)

Term expires 1933

E. H. BROWN — L. S. MARKS
I. E. MOULTROP — E. N. TRUMP
G. A. HORNE

Term expires 1934

C. HAROLD BERRY
FRANCIS HODGKINSON
D. S. JACOBUS
L. F. MOODY
E. B. RICKETTS

Term expires 1935

F. R. LOW
A. T. BROWN
R. H. FERNALD
C. F. HIRSHFELD
R. J. S. PIGOTT

Term expires 1936

A. G. CHRISTIE
E. C. HUTCHINSON
PAUL DISERENS
G. A. ORROK
WM. MONROE WHITE

Term expires 1937

HARTE COOKE
E. R. FISH
O. P. HOOD
H. B. OATLEY
W. J. WOHLENBERG

SAFETY

H. W. MOWERY, *Chairman and Representative on Council* (1933)
T. A. WALSH, JR. (1934) — W. M. GRAFF (1936)
M. H. CHRISTOPHERSON (1935) — H. H. JUDSON (1937)

PROFESSIONAL CONDUCT

A. G. CHRISTIE, *Chairman and Representative on Council* (1933)
H. S. PHILBRICK (1934) — E. R. FISH (1936)
C. G. SPENCER (1935) — J. H. HERRON (1937)

Special Administrative Committees

REGULAR NOMINATING COMMITTEE

GROUP	REPRESENTATIVE	ALTERNATE
I	P. N. KISTLER	W. E. FREELAND
II	W. M. KEENAN	C. P. BLISS
III	V. M. PALMER	P. F. NYDEGGER
IV	W. R. WOOLRICH, *Secretary*	J. H. POUND
V	H. K. KUGEL	E. R. GNADE
VI	T. L. WILKINSON, *Chairman*	JAMES BROWER
VII	R. H. G. EDMONDS	F. A. RODGERS
		C. P. MANN

LOCAL SECTIONS IN NOMINATING COMMITTEE GROUPS

GROUP I

BOSTON — NEW HAVEN
BRIDGEPORT — NORWICH
GREEN MOUNTAIN — PROVIDENCE
HARTFORD — WATERBURY
MERIDEN — WESTERN MASSACHUSETTS
NEW BRITAIN — WORCESTER

GROUP II

METROPOLITAN (N. Y.) AND FOREIGN MEMBERS

GROUP III

ANTHRACITE-LEHIGH VALLEY — ROCHESTER
BALTIMORE — SCHENECTADY
CENTRAL PENNSYLVANIA — SUSQUEHANNA
ONTARIO — SYRACUSE
PHILADELPHIA — UTICA
PLAINFIELD — WASHINGTON, D. C.

GROUP IV

ATLANTA — KNOXVILLE
BIRMINGHAM — MEMPHIS
CHARLOTTE — NEW ORLEANS
CHATTANOOGA — NORTH TEXAS
FLORIDA — RALEIGH
GREENVILLE — SAVANNAH
HOUSTON — VIRGINIA

GROUP V

AKRON-CANTON — INDIANAPOLIS
BUFFALO — LOUISVILLE
CINCINNATI — PENINSULA
CLEVELAND — PITTSBURGH
COLUMBUS — TOLEDO
DAYTON — WEST VIRGINIA
DETROIT — YOUNGSTOWN
ERIE

GROUP VI

CHICAGO — ROCK RIVER VALLEY
KANSAS CITY — ST. JOSEPH VALLEY
MID-CONTINENT — ST. LOUIS
MILWAUKEE — ST. PAUL
MINNEAPOLIS — TRI-CITIES
NEBRASKA

GROUP VII

COLORADO — SAN FRANCISCO
INLAND EMPIRE — UTAH
LOS ANGELES — WESTERN WASHINGTON
OREGON

TELLERS OF ELECTION FOR OFFICERS

C. A. HESCHELES — V. M. ZAFFARANO
RICHARD KUTZLEB, JR.

Special Technical Committee

BOILER CODE

F. R. LOW, *Chairman* — C. E. GORTON
D. S. JACOBUS, *Vice-Chairman* — A. M. GREENE, JR.
C. W. OBERT, *Honorary Secretary* — F. B. HOWELL
M. JURIST, *Acting Secretary* — J. O. LEECH
H. E. ALDRICH — M. F. MOORE
W. H. BOEHM — I. E. MOULTROP
R. E. CECIL — C. O. MYERS
F. S. CLARE — H. B. OATLEY
W. F. DURAND — JAMES PARTINGTON
E. R. FISH — C. L. WARWICK
V. M. FROST — A. C. WEIGEL

H. LEROY WHITNEY

Honorary Members

T. E. DURBAN — W. F. KIESEL, JR.
C. L. HUSTON — H. H. VAUGHAN

Special Council Committees

ECONOMIC STATUS OF THE ENGINEER

C. F. Hirshfeld, *Chairman*
D. S. Kimball
C. N. Lauer
H. B. Oatley
H. L. Whittemore
W. E. Wickenden

J. M. Todd, *Chairman, Committee on Local Sections*
D. B. Prentice, *Chairman, Committee on Relations With Colleges* } *Ex-Officio*

FREEMAN SCHOLARSHIP COMMITTEE

Clarke Freeman
E. C. Hutchinson
C. T. Main

POLICIES AND BUDGET[6]

H. R. Westcott, *Chairman*
B. M. Brigman
H. M. Burke
W. H. Carrier
A. C. Jewett
R. G. Macy
A. L. Maillard
M. C. Maxwell
W. H. Winterrowd

J. H. Lawrence, *Finance Committee*
Erik Oberg, *Treasurer* } *Ex-Officio*

REGISTRATION OF ENGINEERS

J. H. Lawrence, *Chairman*
D. S. Kimball
Paul Doty
C. F. Hirshfeld
V. M. Palmer
J. M. Todd

RELIEF AND EMPLOYMENT

A. A. Potter
W. W. Macon
Calvin W. Rice

REVENUES[6]

W. L. Batt, *Chairman*
J. D. Cunningham
D. S. Kimball
R. E. Flanders
C. N. Lauer
Erik Oberg
J. L. Walsh

Special Committees of Standing Committees

Publications Committee

BIOGRAPHY ADVISORY COMMITTEE

R. V. Wright, *Chairman*
L. P. Alford
R. E. Flanders
F. R. Low
G. A. Orrok
J. W. Roe
W. H. Winterrowd

Professional Divisions Committees

PROCESS INDUSTRIES COMMITTEE

C. E. Harrington, *Chairman*
Victor Wichum, *Vice-Chairman*
T. R. Olive, *Secretary*
Crosby Field
H. D. Munson
C. W. Thomas

PURE AIR COMMITTEE

(*Dates in parentheses denote expiration of terms*)

E. C. Hutchinson, *Chairman* (1936)
E. H. Whitlock, *Secretary* (1934)
G. W. Bach (1927)
J. W. Armour (1933)
O. P. Hood (1935)

Professional Divisions

Division	Executive Committee Chairmen
Aeronautic	W. B. Mayo
Applied Mechanics	J. M. Lessells
Fuels	R. A. Sherman
Hydraulic	D. J. McCormack
Iron and Steel	A. J. Boynton
Machine Shop Practice	B. P. Graves
Management	F. E. Raymond
Materials Handling	J. B. Webb
National Defense	Calvin W. Rice (*Secretary*)
Oil and Gas Power	L. R. Ford
Petroleum	W. G. Heltzel
Power	C. S. Gladden
Printing Industries	F. E. Wilder[7]
Railroad	L. K. Sillcox
Textile	P. A. Merriam
Wood Industries	A. D. Smith, Jr.

[6] Formerly Revenues Committee.
[7] Reorganized as Special Committee on Policies and Budget, February 17, 1933.

Local Sections

Section	Executive Committee Chairmen[8]
Akron-Canton	K. W. Pike
Anthracite-Lehigh Valley	B. F. Rogers
Atlanta	A. J. Kroog
Baltimore	J. R. Baker
Birmingham	S. D. Moxley
Boston	C. Harold Berry
Bridgeport	D. B. Bullard[9]
Buffalo	H. D. Munson
Central Pennsylvania	F. G. Hechler
Charlotte	Asa Hosmer
Chattanooga	Newell Sanders
Chicago	D. H. Skeen
Cincinnati	C. L. Koehler
Cleveland	Theodore Maynz
Colorado	Arthur Halliwell
Columbus	H. M. Bush
Dayton	H. G. Weinland
Detroit	A. N. Goddard
Erie	L. P. DeVau
Florida	P. O. Yeaton
Green Mountain	J. A. McPherson
Greenville	C. Q. Garey
Hartford	H. P. Cox, Jr.
Houston	Oscar Wolf
Indianapolis	Homer Rupard
Inland Empire	A. C. Abell[10]
Kansas City	N. W. Downes
Knoxville	W. W. Carson, Jr.
Los Angeles	J. A. Garrett
Louisville	B. M. Brigman
Memphis	J. T. Roberts
Meriden	E. H. Kammerer
Metropolitan	G. B. Pegram
Mid-Continent	H. F. Brindel
Milwaukee	W. D. Bliss
Minneapolis	J. V. Martenis
Nebraska	W. L. DeBaufre
New Britain	H. L. Spaunburg
New Haven	L. E. Seeley
New Orleans	Charles Wiggins, Jr.
North Texas	R. R. Slaymaker
Norwich	L. E. Whiton
Ontario	S. L. Fear
Oregon	G. F. McDougall
Peninsula	G. H. Waring
Philadelphia	E. P. Kiehl
Pittsburgh	D. L. McNulty
Plainfield	E. M. Getzoff[11]
Providence	G. T. Huxford
Raleigh	L. L. Vaughan
Rochester	A. H. Candee
Rock River Valley	P. H. Tigwell
St. Joseph Valley	C. C. Wilcox
St. Louis	J. R. Jackson
St. Paul	G. F. Endicott
San Francisco	W. M. Moody
Savannah	W. H. Artley
Schenectady	E. H. Hull

[7] Deceased August 12, 1933. Term completed by John Clyde Oswald
[8] Term of office, July, 1932–July, 1933.
[9] Resigned December 26, 1932. T. H. Beard appointed to complete term.
[10] Deceased January 1, 1933. U. B. Hough appointed to complete term.
[11] Resigned February 1, 1933. R. S. Hoffman appointed to complete term.

Section	Executive Committee Chairmen
Susquehanna	D. J. McCormack
Syracuse	J. W. Linford
Toledo	R. B. Haynes[12]
Tri-Cities	J. H. Ploehn
Utah	E. H. Beckstrand
Utica	R. T. Kent
Virginia	A. F. Macconochie
Washington, D. C.	Frank Cushman
Waterbury	J. H. Roberts
Western Massachusetts	V. P. Marran
Western Washington	C. P. Mann
West Virginia	C. E. Bliven
Worcester	K. G. Merriam
Youngstown	M. H. Mawhinney

Student Branches

The establishment of a Student Branch of the Ontario Section of the Society was approved by the Council at the Annual Meeting in December, 1933. The honorary chairman has not yet been appointed.

Branch	Honorary Chairmen[13]
Akron, Univ. of, Akron, Ohio	F. S. Griffin
Alabama Polytechnic Inst., Auburn, Ala.	V. C. Finch
Alabama, Univ. of, University, Ala.	J. M. Gallalee
Arkansas, Univ. of, Fayetteville, Ark.	R. G. Paddock
Armour Inst. of Technology, Chicago, Ill.	J. C. Peebles
Brooklyn, Polytechnic Inst. of, Brooklyn	Gerado Immediato
Brown Univ., Providence, R. I.	J. A. Hall
Bucknell Univ., Lewisburg, Pa.	F. E. Burpee
California Inst. of Technology, Pasadena	R. L. Daugherty
California, Univ. of, Berkeley, Calif.	H. B. Langille
Carnegie Inst. of Tech., Pittsburgh, Pa.	S. B. Ely
Case School of Applied Science, Cleveland	E. S. Ault
Catholic Univ. of America, Washington	G. A. Weschler
Cincinnati, Univ. of, Cincinnati, Ohio	J. W. Bunting
Clarkson College of Tech., Potsdam, N. Y.	J. A. Ross, Jr.
Clemson College, Clemson College, S. C.	J. H. Sams
Colorado Agricultural College, Fort Collins	Joseph Pinsky
Colorado, Univ. of, Boulder, Colo.	W. F. Mallory
Columbia Univ., New York, N. Y.	L. R. Ford
Cooper Union, New York, N. Y.	H. F. Roemmele
Cornell Univ., Ithaca, N. Y.	D. S. Kimball, Jr.
Delaware, Univ. of, Newark, Del.	W. F. Lindell
Detroit, Univ. of, Detroit, Mich.	F. J. Linsenmeyer
Drexel Inst., Philadelphia, Pa.	Dawson Dowell
Florida, Univ. of, Gainesville, Fla.	C. H. Janes
George Washington Univ., Washington, D.C.	M. A. Lett
Georgia School of Technology, Atlanta, Ga.	N. C. Ebaugh
Harvard Univ., Cambridge, Mass.	L. S. Marks
Idaho, Univ. of, Moscow, Idaho	H. F. Gauss
Illinois, Univ. of, Urbana, Ill.	C. H. Casberg
Iowa State College, Ames, Ia.	R. A. Norman
Iowa, State Univ. of, Iowa City, Ia.	R. M. Barnes
Johns Hopkins Univ., Baltimore, Md.	A. G. Christie
Kan. State Agricultural College, Manhattan	A. J. Mack
Kansas, Univ. of, Lawrence, Kan.	J. A. King
Kentucky, Univ. of, Lexington, Ky.	C. C. Jett
Lafayette College, Easton, Pa.	P. B. Eaton
Lehigh Univ., Bethlehem, Pa.	A. W. Luce
Lewis Inst. of Technology, Chicago	J. S. Kozacka
Louisiana State Univ., Baton Rouge, La.	Hamilton Johnson
Louisville, Univ. of, Louisville, Ky.	H. H. Fenwick
Lowell Textile Inst., Lowell, Mass.	H. J. Ball
Maine, Univ. of, Orono, Me.	I. H. Prageman
Marquette Univ., Milwaukee, Wis.	J. E. Schoen
Mass. Inst. of Technology, Cambridge	James Holt
Michigan College of Mining and Technology, Houghton, Mich.	R. R. Seeber
Michigan State College, East Lansing, Mich.	H. B. Dirks
Michigan, Univ. of, Ann Arbor, Mich.	O. W. Boston
Minnesota, Univ. of, Minneapolis, Minn.	B. J. Robertson
Mississippi State College, State College	R. C. Carpenter
Missouri School of Mines and Metallurgy, Rolla, Mo.	R. O. Jackson
Missouri, Univ. of, Columbia, Mo.	R. W. Selvidge
Montana State College, Bozeman, Mont.	Eric Therklesen
Nebraska, Univ. of, Lincoln, Neb.	C. A. Sjogren
Nevada, Univ. of, Reno, Nev.	F. H. Sibley
Newark College of Engineering, Newark, N. J.	R. B. Rice
New Hampshire, Univ. of, Durham, N. H.	T. J. Laton
New York, College of City of, New York	E. B. Smith
New York Univ., New York, N. Y.	Carlos de Zafra
North Carolina State College, Raleigh	F. B. Turner
North Carolina, Univ. of, Chapel Hill	N. P. Bailey
North Dakota Agricultural College, Fargo	J. R. Van Dyke
North Dakota, Univ. of, Grand Forks	A. J. Diakoff
Northeastern Univ., Boston, Mass.	J. W. Zeller
Notre Dame, Univ. of, Notre Dame, Ind.	C. C. Wilcox
Ohio Northern Univ., Ada, Ohio	J. A. Needy
Ohio State Univ., Columbus, Ohio	F. W. Marquis
Oklahoma A.&M. College, Stillwater, Okla.	V. L. Malleev
Oklahoma, Univ. of, Norman, Okla.	H. V. Beck
Oregon State Agricultural College, Corvallis	J. C. Othus
Pennsylvania State College, State College	C. L. Allen
Pennsylvania, Univ. of, Philadelphia, Pa.	W. A. Sloan
Pittsburgh, Univ. of, Pittsburgh, Pa.	J. A. Dent
Porto Rico, Univ. of, Mayaguez, P. R.	Luis Stefani
Pratt Inst., Brooklyn, N. Y.	J. W. Hunter
Princeton Univ., Princeton, N. J.	E. P. Culver
Purdue Univ., W. Lafayette, Ind.	F. C. Hockema
Rensselaer Polytechnic Inst., Troy, N. Y.	Thomas Fitzgerald
Rhode Island State College, Kingston, R. I.	Carroll Billmeyer
Rice Inst., Houston, Tex.	J. H. Pound
Rose Polytechnic Inst., Terre Haute, Ind.	H. C. Gray
Rutgers Univ., New Brunswick, N. J.	U. C. Holland
Santa Clara, Univ. of, Santa Clara, Calif.	G. L. Sullivan
South Dakota, Univ. of, Vermilion, S. D.	M. W. Davidson
Southern California, Univ. of, Los Angeles	S. F. Duncan
Southern Methodist Univ., Dallas, Tex.	R. R. Slaymaker
Stanford Univ., Stanford University, Calif.	Lawrence Washington
Stevens Inst. of Technology, Hoboken, N. J.	R. F. Deimel
Swarthmore College, Swarthmore, Pa.	G. A. Bourdelais
Syracuse Univ., Syracuse, N. Y.	S. T. Hart
Tennessee, Univ. of, Knoxville, Tenn.	Duncan White
Texas, A.&M. College of, College Station	J. A. Trail
Texas Technological College, Lubbock, Tex.	V. L. Doughtie
Texas, Univ. of, Austin, Tex.	J. L. Bruns
Tufts College, Tufts College, Mass.	E. E. Leavitt
Tulane Univ. of Louisiana, New Orleans	W. B. Gregory
U. S. Naval Academy, Post Graduate School, Annapolis	P. J. Kiefer
Utah, Univ. of, Salt Lake City, Utah	M. B. Hogan
Vanderbilt Univ., Nashville, Tenn.	J. E. Boynton
Vermont, Univ. of, Burlington, Vt.	E. L. Sussdorff
Villanova College, Villanova, Pa.	J. S. Morehouse
Virginia Polytechnic Inst., Blacksburg, Va.	N. W. Conner
Virginia, Univ. of, University, Va.	H. C. Hesse
Washington, State College of, Pullman, Wash.	F. W. Candee
Washington Univ., St. Louis, Mo.	E. H. Sager
Washington, Univ. of, Seattle, Wash.	B. T. McMinn
West Virginia Univ., Morgantown, W. Va.	L. D. Hayes
Wisconsin, Univ. of, Madison, Wis.	B. G. Elliott
Worcester Polytechnic Inst., Worcester	M. L. Price
Wyoming, Univ. of, Laramie, Wyo.	R. S. Sink
Yale Univ., New Haven, Conn.	S. W. Dudley

[12] Resigned August 22, 1932. E. G. DeCoriolis appointed to complete term.

[13] In office December 31, 1933.

Awards

The following paragraphs deal with the awards, scholarships, and loan funds which come within the jurisdiction of the A.S.M.E. The Society also participates with other engineering societies in a number of joint awards, a brief statement regarding which will be found on page 49.

Honorary Membership, to which persons of acknowledged professional eminence are elected by unanimous vote of Council under the provisions of the By-Laws and Rules. A list of honorary members is given on page 16.

Life Membership, which may be conferred by the Council for distinguished service to the Society; or secured by a member by payment for an annuity in accordance with the provisions of the By-Laws.

A.S.M.E. Medal, established by the Society in 1920 to be presented for distinguished service in engineering and science. May be awarded

for general service in science having possible application in engineering.

Holley Medal, instituted and endowed in 1924 by George I. Rockwood, Past Vice-President of the Society, to be bestowed for some great and unique act of genius of engineering nature that has accomplished a great and timely public benefit.

Worcester Reed Warner Medal, provision for which was made in the will of Worcester Reed Warner, Honorary Member of the Society, is a gold medal to be bestowed on the author of the most worthy paper received, dealing with progressive ideas in mechanical engineering or efficiency in management.

Melville Medal, established in 1914 by the bequest of Rear-Admiral George W. Melville, Honorary Member and Past-President of the Society, to be presented for an original paper or thesis of exceptional merit, presented to the Society for discussion and publication, to encourage excellence in papers. The medal may be presented annually.

Spirit of Saint Louis Medal, endowed by members of the Society and citizens residing in St. Louis, Mo., to be awarded for meritorious service in the field of aeronautical engineering. This medal will be awarded at the discretion of the Council of the Society at approximately three-year periods upon the recommendation of a Spirit of Saint Louis Medal Board of Award made up of six members, each appointed for a term of nine years and the terms of two members expiring at each three-year period. The St. Louis Section and the Aeronautic Division will each be responsible for the nomination of three members.

Junior Award, annual cash award of $50, established in 1914, from a fund created by Henry Hess, Past Vice-President of the Society, to be presented, together with an engraved certificate, for the best paper or thesis submitted by a Junior Member.

Student Awards, two annual cash awards of $25 each, established in 1914, from a fund created by Henry Hess, Past Vice-President of the Society, to be presented, together with engraved certificates, for the best papers or theses submitted by members of Student Branches.

Charles T. Main Award, annual cash award of $150, established in 1919 from a fund created by Charles T. Main, Past-President of the Society, to be awarded to a student of engineering, preferably a member of a Student Branch of the Society, for the best paper within the general subject of the "Influence of the Profession Upon Public Life." The exact subject is assigned by the Committee on Awards, subject to the approval of the Council, and is announced each year through the Honorary Chairmen of the Student Branches.

SCHOLARSHIPS AND LOAN FUNDS

Max Toltz: Loan Fund of $15,000 established by Major Max Toltz, former member of the Council of the Society, the income to be used for assistance to students.

John R. Freeman: Fund of $25,000 established in 1926 by John R. Freeman, Past-President of the Society, the income to be used for travel scholarships and research.

Woman's Auxiliary: Scholarship or Fellowship offered by the Woman's Auxiliary to the Society to assist sons and daughters of members or worthy students of mechanical engineering.

RECIPIENTS OF AWARDS

The names of the recipients of the different awards to date are given in the following lists, together with the dates of presentation, and the services or papers for which the awards were made. There were no awards for the years not listed.

A.S.M.E. Medal

1921 Hjalmar Gotfried Carlson, in recognition of the services rendered the Government because of his invention and part in the production of 20,000,000 Mark III drawn steel booster casings used principally as a component of 75-mm high-explosive shells, but also used extensively in gas shells and bombs

1923 Frederick Arthur Halsey, for his paper describing the premium system of wage payments presented before the Society at the Providence Meeting in 1891, as the adoption of the methods there proposed has had a profound effect toward harmonizing the relations of worker and employer

1923 John Ripley Freeman, for his eminent service in engineering and manufacturing by his meritorious work in fire prevention and the preservation of property

1926 R. A. Millikan, in recognition of his contributions to science and engineering

1927 Wilfred Lewis, for his contributions to the design and construction of gear teeth

1928 Julian Kennedy, for his services and contributions to the iron and steel industry

1929 William LeRoy Emmet, for his contributions in the development of the steam turbine, electric propulsion of ships, and other power-generating apparatus

1931 Albert Kingsbury, for his research and development work in the field of lubrication

1933 Ambrose Swasey, for his contributions to the advancement of the engineering profession and for his part in the development of the turret lathe and the astronomical telescope.

Holley Medal

1924 Hjalmar Gotfried Carlson, for his inventions and processes which made possible the timely production of drawn steel booster casings for artillery ammunition, thereby aiding victory in the World War

1928 Elmer Ambrose Sperry, for achievements and inventions that have advanced the naval arts, including the gyroscope that has freed navigation from the dangers of the fluctuating magnetic compass

1929 Baron Chuzaburo Shiba, for his contributions to knowledge through fundamental research, including the field of aerodynamics, by the development of ultra-rapid kinematographic methods.

Worcester Reed Warner Medal

1933 Dexter S. Kimball, for his contributions to efficiency in management as exemplified by his recently revised "Principles of Industrial Organization" and by his many articles, engineering society papers, and public addresses.

Melville Medal

1927 Leon P. Alford, "Laws of Manufacturing Management"

1929 Joseph Wickham Roe, "Principles of Jig and Fixture Practice"

1930 Herman Diederichs and William D. Pomeroy, "The Occurrence and Elimination of Surge or Oscillating Pressure in Discharge Lines From Reciprocating Pumps"

1931 Arthur E. Grunert, "Comparative Performance of a Pulverized-Coal-Fired Boiler Using Bin System and Unit System of Firing"

1932 Alexey J. Stepanoff, "Leakage Loss and Axial Thrust in Centrifugal Pumps"

1933 William E. Caldwell, "Characteristics of Large Hell Gate Direct-Fired Boiler Units."

Spirit of Saint Louis Medal

1929 Daniel Guggenheim, founder of the Guggenheim Fund for the Promotion of Aeronautics

1932 Paul Litchfield, for his work in encouraging and sponsoring airship design and construction in this country.

Junior Award

1915 Ernest O. Hickstein, "Flow of Air Through Thin Plate Orifices"

1916 L. B. McMillan, "The Heat Insulating Properties of Commercial Steam-Pipe Coverings"

1919 E. D. Whalen, "Properties of Airplane Fabrics"

1921 S. Logan Kerr, "Moody Ejector Turbine"

1922 R. H. Heilman, "Heat Losses From Bare and Covered Wrought-Iron Pipe at Temperatures up to 800 Degrees Fahrenheit"

F. L. Kallam, "Preliminary Report on the Investigation of the Thermal Conductivity of Liquids"

1923 S. S. Sanford and S. Crocker, "The Elasticity of Pipe Bends"

1924 R. H. Heilman, "Heat Losses Through Insulating Material"

1925 Gilbert S. Schaller, "An Investigation of Seattle as a Location for a Synthetic Foundry Industry"

1927 Wm. M. Frame, "Stresses Occurring in the Walls of an Elliptical Tank Subjected to Low Internal Pressure"

1928 M. D. Aisenstein, "A New Method of Separating the Hydraulic Losses in a Centrifugal Pump"

1929 Arthur M. Wahl, "Stresses in Heavy Closely Coiled Helical Springs"

1930 Ed Sinclair Smith, Jr., "Quantity-Rate Fluid Meters"

1931 M. K. Drewry, "Radiant-Superheater Developments"

1932 Edmond M. Wagner, "Frictional Resistance of a Cylinder Rotating in a Viscous Fluid Within a Coaxial Cylinder"

1933 Townsend Tinker, "Surface Condenser Design and Operating Characteristics."

STUDENT AWARD

1916 BOYNTON M. GREEN, Stanford University, "Bearing Lubrication"
HOWARD STEVENS, Rensselaer Polytechnic Institute, "An Investigation of the Dynamic Pressure on Submerged Flat Plates"
M. ADAM, Louisiana State University, "The Adaptability of the Internal Combustion Engine to Sugar Factories and Estates"

1917 H. R. HAMMOND and C. W. HOLMBERG, Pennsylvania State College, "Study of Surface Resistance With Glass as the Transmission Medium"

1919 C. F. LEH and F. G. HAMPTON, Stanford University, "An Experimental Investigation of Steel Belting"
W. E. HELMICK, Stanford University, "An Experimental Investigation of Steel Belting"

1920 HOWARD G. ALLEN, Cornell University, "Wire Stitching Through Paper"

1921 KARL H. WHITE, University of Kansas, "Forces in Rotary Motors"
RICHARD H. MORRIS and ALBERT J. R. HOUSTON, University of California, "A Report Upon an Investigation of the Herschel Type of Improved Weir"

1923 CHARLES F. OLMSTEAD, University of Minnesota, "Oil Burning for Domestic Heating"
H. E. DOOLITTLE, University of California, "The Integrating Gate: A Device for Gaging in Open Channels"

1924 GEORGE STUART CLARK, Stanford University, "Two Methods Used for the Determination of the Gasoline Content of Absorption Oils in Absorption Plants"
L. J. FRANKLIN and CHARLES H. SMITH, Stanford University, "The Effect of Inaccuracy of Spacing on the Strength of Gear Teeth"

1925 HARRY PEASE COX, JR., Rensselaer Polytechnic Institute, "A Study of the Effect of End Shape on the Towing Resistance of a Barge Model"
W. S. MONTGOMERY, JR., and E. RAY ENDERS, JR., Pennsylvania State College, "Some Attempts to Measure the Drawing Properties of Metals"

1926 R. E. PETERSON, University of Illinois, "An Investigation of Stress Concentration by Means of Plaster of Paris Specimens"
CECIL G. HEARD, University of Toronto, "Pressure Distribution over U. S. A. 27 Aerofoil With Square Wing Tips Model Tests"

1927 ALFRED H. MARSHALL, Princeton University, "Evaporative Cooling"
ROGER IRWIN EBY, University of Washington, "Measurement of the Angular Displacement of Flywheels"

1928 CLARENCE C. FRANCK, Johns Hopkins University, "Condition Curves and Reheat Factors for Steam Turbines"

1929 FRANK VERNON BISTROM, University of Washington, "An Investigation of a Rotary Pump"
WILLIAM WALLACE WHITE, University of Washington, "An Investigation of a Rotary Pump"

1930 GERARD EDEN CLAUSSEN, Polytechnic Institute of Brooklyn, "High-Temperature Oxidation of Steel"
HAROLD L. ADAMS and RICHARD L. STITH, University of Washington, "A Wind Tunnel for Undergraduate Laboratory Experiments"

1931 JULES PODNOSSOFF, Polytechnic Institute of Brooklyn, "Pressure and Energy Distribution in Multi-Stage Steam Turbines Operating Under Varying Conditions"

1932 H. E. FOSTER, JR., University of Tennessee, "Factors Affecting Spray Pond Design"
WILLIAM A. MASON, Stanford University, "An Experimental Investigation of the Flame Propagation in Internal-Combustion Engines"

1933 HUGO V. CORDIANO, Polytechnic Institute of Brooklyn, "Thermal Analysis of the Lithium-Magnesium System of Alloys"
JAMES A. OSTRAND, JR., Princeton University, "Sudden Enlargement in the Open Channel."

CHARLES T. MAIN AWARD

1925 CLEMENT R. BROWN, Catholic University of America. Subject: "The Influence of the Locomotive on the Unity of the United States"

1926 W. C. SAYLOR, Johns Hopkins University. Subject: "The Effect of the Cotton Gin Upon the History of the United States During Its First Seventy Years"

1927 No award. Subject: "Scientific Management and Its Effect Upon the Industries"

1928 ROBERT M. MEYER, Newark College of Engineering. Subject: "Scientific Management and Its Effect Upon Manufacturing"

1930 JULES PODNOSSOFF, Polytechnic Institute of Brooklyn. Subject: "The Value of the Safety Movement in the Industries"

1931 ROBERT E. KLISE, University of Michigan. Subject: "Interchangeability—Its Development and Significance in Industry"

1932 MARSHALL ANDERSON, University of Michigan. Subject: "Apprenticeship and Vocational Training"

1933 GEORGE D. WILKINSON, JR., Newark College of Engineering. Subject: "Progress in the Prevention of Smoke and Atmospheric Pollution."

FREEMAN TRAVEL SCHOLARSHIP

1927 HERBERT N. EATON and BLAKE R. VAN LEER
1929 ROBERT T. KNAPP
1931 REGINALD WHITAKER
1932 G. ROSS LORD.

Past-Presidents

A list of past vice-presidents, managers, treasurers, and secretaries will be found in the 1930 Record and Index, pages 10–12. Dates in parentheses denote year of death.

ALEXANDER LYMAN HOLLEY, *Chairman of the Preliminary Meeting for Organization of The American Society of Mechanical Engineers* (1882)

1880–1882 ROBERT HENRY THURSTON (1903)
1883 ERASMUS DARWIN LEAVITT (1916)
1884 JOHN EDSON SWEET (1916)
1885 JOSEPHUS FLAVIUS HOLLOWAY (1896)
1886 COLEMAN SELLERS (1907)
1887 GEORGE H. BABCOCK (1893)
1888 HORACE SEE (1909)
1889 HENRY ROBINSON TOWNE (1924)
1890 OBERLIN SMITH (1926)
1891 ROBERT WOOLSTON HUNT (1923)
1892 CHARLES HARDING LORING (1907)
1893–1894 ECKLEY BRIXTON COXE (1895)
1895 EDWARD F. C. DAVIS (1895)
1895 CHARLES ETHAN BILLINGS (1920)
1896 JOHN FRITZ (1913)
1897 WORCESTER REED WARNER (1929)
1898 CHARLES WALLACE HUNT (1911)
1899 GEORGE WALLACE MELVILLE (1912)
1900 CHARLES HILL MORGAN (1911)
1901 SAMUEL T. WELLMAN (1919)
1902 EDWIN REYNOLDS (1909)
1903 JAMES MAPES DODGE (1915)
1904 AMBROSE SWASEY
1905 JOHN RIPLEY FREEMAN (1932)
1906 FREDERICK WINSLOW TAYLOR (1915)
1907 FREDERICK REMSEN HUTTON (1918)
1908 MINARD LAFEVER HOLMAN (1925)
1909 JESSE MERRICK SMITH (1927)
1910 GEORGE WESTINGHOUSE (1914)
1911 EDWARD DANIEL MEIER (1914)
1912 ALEXANDER CROMBIE HUMPHREYS (1927)
1913 WILLIAM FREEMAN MYRICK GOSS (1928)
1914 JAMES HARTNESS
1915 JOHN ALFRED BRASHEAR (1920)
1916 DAVID SCHENCK JACOBUS
1917 IRA NELSON HOLLIS (1930)
1918 CHARLES THOMAS MAIN
1919 MORTIMER ELWYN COOLEY
1920 FRED J. MILLER
1921 EDWIN S. CARMAN
1922 DEXTER SIMPSON KIMBALL
1923 JOHN LYLE HARRINGTON
1924 FREDERICK ROLLINS LOW
1925 WILLIAM FREDERICK DURAND
1926 WILLIAM LAMONT ABBOTT
1927 CHARLES M. SCHWAB
1928 ALEX DOW
1929 ELMER AMBROSE SPERRY (1930)
1930 CHARLES PIEZ (1933)
1931 ROY V. WRIGHT
1932 CONRAD N. LAUER
1933 A. A. POTTER

A.S.M.E. Representatives on Other Activities

The A.S.M.E. is also represented on a number of special boards and on many committees of other organizations. Dates in parenthesis denote expiration of terms.

AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE

SECTION M, ENGINEERING

W. S. Monroe (1933) Ralph Earle (1933)

Representatives appointed annually from locality in which meeting is to be held.

AMERICAN ENGINEERING COUNCIL

Terms expire January 1, 1934	*Terms expire January 1, 1935*
L. P. Alford	B. M. Brigman
Paul Doty	W. S. Conant
C. E. Ferris	Hugo Diemer
R. E. Flanders	W. A. Hanley
E. C. Hutchinson	Ernest Kreher
C. N. Lauer	G. E. Pellissier
J. H. Lawrence	J. W. Roe
R. C. Marshall, Jr.	B. R. Van Leer
A. A. Potter, *Chairman*	T. R. Weymouth
W. H. Trask, Jr.	D. Robert Yarnall, *Vice-Chairman*

AMERICAN SOCIETY OF SAFETY ENGINEERS

ENGINEERING SECTION, NATIONAL SAFETY COUNCIL

H. W. Mowery Ernest Hartford

AMERICAN STANDARDS ASSOCIATION

C. M. Chapman (1933)
K. H. Condit (1934)
C. B. LePage, C. W. Spicer, V. R. Willoughby (*Alternates (Serve one year)*)

THE ENGINEERING FOUNDATION

D. Robert Yarnall (1934) E. R. Fish (1935)[14]
A. E. White (1936)

ENGINEERING SOCIETIES EMPLOYMENT SERVICE

NATIONAL BOARD

Calvin W. Rice, *Chairman*

ENGINEERS' COUNCIL FOR PROFESSIONAL DEVELOPMENT

C. F. Hirshfeld J. H. Lawrence
W. E. Wickenden

NATIONAL RESEARCH COUNCIL

DIVISION OF ENGINEERING AND INDUSTRIAL RESEARCH

D. S. Jacobus (1936)[15] Herman Diederichs (1933)
W. D. Ennis (1935)[16] Albert Kingsbury (1934)
G. W. Lewis (1934)[16] W. R. Webster (1935)
Calvin W. Rice, *Ex-Officio*

UNITED ENGINEERING TRUSTEES, INC.

E. R. Fish (1934) W. L. Batt (1935)
H. V. Coes (1936)

U.E.T. COORDINATION COMMITTEE OF ENGINEERING SOCIETIES

D. S. Kimball R. I. Rees
John Lyle Harrington

[14] A.S.M.E. representative from Board of United Engineering Trustees, Inc.
[15] Special member appointed June, 1933.
[16] Transferred from membership at large of Division of Engineering and Industrial Research, upon its reorganization in April, 1933.

Exchange of Courtesies

Special arrangements have been made between the A.S.M.E. and the following organizations whereby their members will be mutually accorded such courtesies as the use of reading and writing rooms, assistance in securing general information and in arranging to visit industrial plants, receipt of mail, etc.

OUTSIDE THE UNITED STATES

Austria—Vienna: Osterreichischer Ingenieur und Architekten Verein;
Belgium—Bruxelles: Federation des Associations Belges d'Ingenieurs. Société Belge des Ingenieurs et des Industriels.
Canada—Montreal: The Engineering Institute of Canada.
Cuba—Havana: Sociedad Cubana de Ingenieros.
Czechoslovakia—Prague: Association of Czechoslovakian Engineers and Architects in Praha; Masarykova Akademie Prace; Spolek Ceskoslovenskych Inzenyru.
Denmark—Copenhagen: Dansk Ingeniorforening.
England—London: British Association for the Advancement of Science; British Engineering Standards Association; British Engineers Association; Chemical Society; The Institute of Fuel; The Institute of Marine Engineers; The Institute of Metals; The Institution of Automobile Engineers; The Institution of Civil Engineers; The Institution of Electrical Engineers; The Institution of Engineering Inspection; The Institution of Mechanical Engineers; The Institution of Mining Engineers; Institution of Naval Architects; The Institution of Structural Engineers; International Electrotechnical Commission; The Iron and Steel Institute; Junior Institution of Engineers; The Royal Aeronautical Society; The Society of Engineers.
Newcastle-on-Tyne: North East Coast Institution of Engineers and Shipbuilders; North of England Institute of Mining and Mechanical Engineers.
South Kensington: Newcomen Society.
France—Paris: l'Association Technique Maritime et Aeronautique; Société des Ingènieurs Civils de France; Société Francaise des Electriciens.
Germany—Berlin: Verein deutscher Ingenierue.
Italy—Rome: Sindacato Nazionale Fascista delgi Ingegneri.
Japan—Tokyo: Kogakkai.
Mexico—Mexico City: Asociacion de Ingenieros y Arquitectos de Mexico.
Netherlands—The Hague: Koninklijk Instituut van Ingenieurs.
Norway—Oslo: Den Norske Ingeniorforening.
Scotland—Glasgow: Institution of Engineers and Shipbuilders in Scotland.
South Africa—Johannesburg: South African Institution of Engineers.
South America—Colombia: Sociedad Colombiana de Ingenieros.
Sweden—Stockholm: Svenska Teknologforeningen.
Switzerland—Winterthur: Verein Schweizerischer Maschinen-Industrieller.

UNITED STATES

California—Los Angeles: Engineering and Architects Association.
District of Columbia—Washington: Society of American Military Engineers.
Florida—Gainesville: Florida Engineering Society.
Illinois—Chicago: Western Society of Engineers.
Louisiana—New Orleans: Louisiana Engineering Society.
Maryland—Baltimore: Baltimore Engineers Club.
Massachusetts—Boston: Boston Society of Civil Engineers.
Michigan—Detroit: Detroit Engineering Society.
Missouri—St. Louis: Engineers Club of St. Louis.
New York—New York: American Gas Association; American Institute of Electrical Engineers; American Institute of Mining and Metallurgical Engineers; American Society of Civil Engineers; American Society of Refrigerating Engineers; American Society of Safety Engineers; Chemists Club Library; Columbia University Library; New York Railroad Club.
Rochester: Rochester Engineering Society.
Syracuse: Technology Club of Syracuse.
Utica: Mohawk Valley Engineers' Club.
Ohio—Cleveland: Cleveland Engineering Society.
Dayton: Engineers' Club of Dayton.
Pennsylvania—Philadelphia: Engineers Club of Philadelphia.
Pittsburgh: Engineers Society of Western Pennsylvania.
Scranton: Engineers Society of Northeastern Pennsylvania; Scranton Engineers Club.
Rhode Island—Providence: Providence Engineering Society.
Tennessee—Nashville: Engineering Association of Nashville.
Washington—Seattle: Pacific Northwest Society of Engineers.
Spokane: Associated Engineers of Spokane.

Honorary Members

HONORARY MEMBERS IN PERPETUITY

ALEXANDER LYMAN HOLLEY, Founder of the Society. Died 1882.

JOHN EDSON SWEET, Founder of the Society. Died 1916.

HENRY ROSSITER WORTHINGTON, Founder of the Society. Died 1880.

DECEASED HONORARY MEMBERS

	ELECTED	DIED
HORATIO ALLEN	1880	1889
SIR WILLIAM ARROL	1905	1913
SIR BENJAMIN BAKER	1886	1907
JOHANN BAUSCHINGER	1884	1893
SIR HENRY BESSEMER	1891	1898
SIR FREDERICK JOSEPH BRAMWELL	1884	1903
JOHN ALFRED BRASHEAR	1908	1920
GUSTAVE CANET	1900	1908
ANDREW CARNEGIE	1907	1919
DANIEL KINNEAR CLARK	1882	1896
RUDOLPH JULIUS EMMANUEL CLAUSIUS	1882	1888
SIR JOHN COODE	1889	1892
PETER COOPER	1882	1883
CARL GUSTAF PATRIK DE LAVAL	1912	1913
RUDOLPH DIESEL	1912	1913
JAMES DREDGE	1886	1906
VICTOR DWELSHAUVERS-DERY	1886	1913
THOMAS ALVA EDISON	1904	1931
ALEXANDRE GUSTAVE EIFFEL	1889	1923
MARSHAL FERDINAND FOCH	1921	1929
SIR CHARLES DOUGLAS FOX	1900	1921
JOHN RIPLEY FREEMAN	1932	1932
JOHN FRITZ	1900	1913
MAJOR - GENERAL GEORGE WASHINGTON GOETHALS	1917	1928
FRANZ GRASHOF	1884	1893
REAR-ADMIRAL ROBERT STANISLAUS GRIFFIN	1920	1933
OTTO HALLAUER	1882	1883
CHARLES HAYNES HASWELL	1905	1907
FRIEDRICH GUSTAV HERRMANN	1884	1907
GUSTAV ADOLPH HIRN	1882	1890
JOSEPH HIRSCH	1889	1901
IRA N. HOLLIS	1928	1930
ROBERT WOOLSTON HUNT	1920	1923
BENJAMIN FRANKLIN ISHERWOOD	1894	1915
HENRI LÉAUTÉ	1891	1916
ERASMUS DARWIN LEAVITT	1915	1916
ANATOLE MALLET	1912	1919
CHARLES H. MANNING	1913	1919
REAR-ADMIRAL GEORGE WALLACE MELVILLE	1910	1912
THE HONORABLE SIR CHARLES ALGERNON PARSONS	1920	1931
CHARLES TALBOT PORTER	1890	1910
AUGUSTE C. E. RATEAU	1919	1930
SIR EDWARD J. REED	1882	1906
FRANZ REULEAUX	1882	1905
HENRI ADOLPHE - EUGENE SCHNEIDER	1882	1898
C. WILLIAM SIEMANS	1882	1883
VISCOUNT EI-ICHI SHIBUSAWA	1929	1931
HENRY ROBINSON TOWNE	1921	1924
HENRI TRESCA	1882	1885
WILLIAM CAWTHORNE UNWIN	1898	1933
FRANCIS A. WALKER	1886	1897
WORCESTER REED WARNER	1925	1929
SIR WILLIAM HENRY WHITE	1900	1913
GEORGE WESTINGHOUSE	1897	1914
SIR ALFRED FERNANDEZ YARROW	1914	1932

LIVING HONORARY MEMBERS

	ELECTED
SIR JOHN AUDLEY FREDERICK ASPINALL	1911
WILLIAM WALLACE ATTERBURY	1925
MORTIMER ELWYN COOLEY	1928
CHARLES DE FRÉMINVILLE	1919
NATHANAEL GREENE HERRESHOFF	1921
HERBERT CLARK HOOVER	1925
MASAWO KAMO	1929
HENRI LE CHATELIER	1927
GRANDE UFFICIALE ING. PIO PERRONE	1920
CALVIN WINSOR RICE	1931
PALMER C. RICKETTS	1931
CHARLES M. SCHWAB	1918
AMBROSE SWASEY	1916
ELIHU THOMSON	1930
SAMUEL MATTHEWS VAUCLAIN	1920
OSKAR VON MILLER	1912
RIGHT HONORABLE LORD WEIR	1920
ORVILLE WRIGHT	1918

Report of the Council

During the year, two important phases of our Society's affairs have had the principal attention of the Council. These are matters of direct interest and value to our members.

First, the response to the necessity for the most prudent operation of the Society in the face of a constantly decreasing income was prompt and effective. Far-reaching economies have been made in every activity. Policies have been adopted which will be of inestimable value to the Society in the years to come as well as the present. As a result, the Society is on a sound financial basis and with the advancement in general business conditions, further improvement in the affairs of the Society can be looked for. While there is still need for proceeding providently, the outlook is such as to cause confidence in the future.

Second, greater thought than ever before has been given to the interests and views of our members. Every effort has been made to promote these interests in so far as it has been within the power of the Society to do so. The utmost consideration has been given to members' views.

Constructive action has been initiated for the benefit of mechanical engineers in the capital-goods industries. The code of fair competition for professional engineers has been endorsed. Relief and unemployment and the problems of the junior engineer have had appropriate attention. The foundation has been laid for the development and unification of the profession. These are activities which are of direct and personal value to all members, particularly in these times.

The detailed reports submitted to the Council by the Standing and Special Committees of the Society and by its representatives on joint activities reveal an amount of constructive activity which is phenomenal in view of the reduced income and the many serious problems that faced the individual members of the Society as well as the Society as a whole. A greater degree of interest in the welfare of the Society than ever before evidenced by the members, indicates a loyalty strong enough to sustain the Society through the period of necessarily reduced activity still ahead.

The low ebb of the industries employing engineers, particularly the mechanical, has brought a serious condition and a challenge to our Society. Recognizing the fact that activity in the capital-goods industries is essential to the welfare of engineers, the Council has appointed a Committee on the Capital-Goods Industries to study the facts involved. This committee will attempt to gage the attitude of industry and of the Federal Government regarding the purchase of capital goods at this time, with a view to the development of a constructive program for bringing about increased expenditures for machinery and materials in the manufacture of which engineers are largely engaged. It is suggested that such a program might be carried out with the cooperation of the American Engineering Council through the Local Sections and Professional Divisions of the Society and other influential groups of engineers.

The Status of the Engineer

Notwithstanding the pressure of many new problems, members of the Council have been continuously thoughtful of measures to improve the status of the engineer as a professional man. With this object in mind the Council approved the Code of Fair Competition for the Professional Engineer Division of the Construction Industry covering relations between professional engineers, clients or employers, employees, and others in respect to construction work, including the investigation, design, and selection of mechanical components of structures. This Division of the Code is of direct concern to mechanical engineers, and the Society has therefore assumed the responsibility of participating in the Board of Control which will administer its provisions. The Council has also authorized the formation of a committee to prepare a manual of engineering practice which will clarify the problems of the consulting engineer who is retained to aid in the design, construction, and installation of mechanical equipment.

Related to this is the suggestion of the Committee on Professional Conduct that the major engineering societies join in the preparation of a uniform Code of Ethics.

The Society is participating actively in the Engineers' Council for Professional Development which is engaged in a program for the accrediting of engineering schools and the development of an accurate definition of an engineer. It is hoped that this will also result in some recommendation looking to a standardization of membership grades in the various engineering societies.

During the year the Council of the Society approved the model law for the registration of professional engineers and land surveyors as a basis for the framing of new registration laws and the amendment of existing laws with a view to attaining a uniform high standard throughout the United States. The Council also authorized the appointment of a representative of the Society as an advisory member to the National Bureau of Registration, which is operated under the auspices of the National Council of State Boards of Engineering Examiners. This body has been established primarily to minimize the effort and expense of securing registration, especially by engineers practising in more than one state.

Unifying the Profession

For the past three years leaders of the engineering profession have been concerned with the general problem of improving the effectiveness of the profession by bringing together in a coordinated way the activities of the several joint bodies sponsored by engineering societies. This question is in the hands of the Coordination Committee of Engineering Societies, which has before it a plan for a central administrative agency that will provide the necessary coordination, administration, and representation. Some of these joint bodies are the United Engineering Trustees, Inc. (the holder of the Engineering Societies Building in New York and trustee for the funds of The Engineering Foundation), The Engineering Foundation, the Ameri-

can Engineering Council, the Engineering Societies Library, the American Standards Association, and the newly formed Engineers' Council for Professional Development.

Relief and Unemployment

The Employment Service, another of the joint activities, with offices in New York, Chicago, and San Francisco, in spite of reduced financial support has carried on in an increasingly helpful way, securing every possible opportunity for the employment of engineers.

Furthermore, in several cities, special campaigns have been instituted by the Local Sections to provide relief for engineers who are in straitened circumstances. Wives of members have cooperated by visiting the families of unemployed engineers and rendering aid urgently necessary.

Special recognition should be given to the work of the Professional Engineers Committee on Unemployment operated by the New York Sections of the four Founder Societies, which has collected and distributed large sums of money and has been successful in securing the cooperation of various other municipal committees. This committee points out that in the period before the depression there were more than enough jobs for engineers, with the result that the engineer was not obliged to concern himself with the technique of securing a job. The P.E.C.U. stresses the need for a concerted attack on the problem of instructing the engineer how to acquire skill in job hunting and even in job making.

Engineers and the Federal Government

The present times have brought added responsibilities to the representatives of the Society on the American Engineering Council and to the Washington office of that body. The report of the Society representatives on the American Engineering Council points out that the centering of unusual national activities in Washington during recent months has given the A.E.C. an opportunity to render a greater and more direct service to the profession than ever before.

The Society itself has supported the proposal made through The Engineering Foundation that sixteen million dollars be appropriated from the Public Works Administration for a six-year program of research in the natural sciences and their applications. This support was given because scientific research is an essential element of any far-sighted and comprehensive program for planning and improving the national prosperity.

Junior Participation

The importance of Junior Members in the Society and the need for providing organized participation by them in the Society's work are increasingly evident. Several Sections, notably the Los Angeles Section, have successfully brought Junior Members into Society work. The Committee on Meetings and Program has had a Junior advisor during the past year. Junior Members are taking increasing responsibility in the work of the technical committees. In order to record this progress and to explore all possibilities, the Council authorized the appointment of a special Committee on Junior Participation, made up of seven Junior Members, which will present a report to the Council at its meeting in December, 1933.

Policies and Program for 1933–34

By far the most constructive task performed by the Council during the past year was the reduction of current operating expenses and the curtailment of plans for expenditures for 1933–34. It should be realized, however, that the reduction in Society expenditures began in 1930. In 1929–30 the total operating expense was $893,126.64; in 1930–31, $840,681.67; in 1931–32, $644,588.52; and in 1932–33, $432,848.99.

The principal difficulty that the Society faces during a period of dropping income is due to the fact that its budget of expenses must be made on the basis of income estimated four months before the opening of the fiscal year. Plans involving commitments for many large items must be made several months before the fiscal year begins, but not until six or seven months later can an accurate check of the dues income be obtained. Therefore, if the income drops seriously below the amount estimated, the Society must reduce expenditures long after the commitments for those expenditures have been made.

This difficulty became acute during 1932–33. As early as November, 1932, only a month after the opening of the fiscal year, it became evident that many of our members were finding increasing difficulty in paying dues. (See statistics in the Finance Report, page 21.) In January of 1933, sharp reductions in advertising income made necessary a reduction of the budget by 14 per cent, or about $78,000. At the Council Meeting in December, 1932, therefore, the Special Committee on Revenues was called into service. This Committee, subsequently organized as the Special Committee on Policies and Budget, was given a twofold task. First, to study the situation that faced our Society because of the depression and devise measures to improve that situation, particularly as regards the immediate future, and to report recommendations to the Executive Committee as a guide for shaping the 1933–34 budget. Second, to determine the future aims and objectives of our Society and propose long-range policies for attaining them, and to make recommendations as a guide for the future conduct of the Society.

This Committee presented an exhaustive report at the meeting of the Council in Chicago in June. In preparing this report, the Committee had studied the activities of the Society for the past ten years, made an analysis of the general situation, studied the detailed statement of income and expenditures and the balance sheet, investigated curtailment of membership activities already made, presented an exhaustive analysis of membership growth, pointed out that the criticism and feeling of the membership had resulted in deterioration of Society morale, and made a careful study of the status of other engineering societies. It reviewed the detailed cost of all of the activities of the Society and classified them into their relative importance in serving the individual members of the Society; made certain suggestions as to reductions in the operating costs which were reflected in the budget for the next year; canvassed new sources of income; and reviewed the activities and obligations of the Society to other organizations.

The final report on the first half of the Committee's work, a mimeographed document of more than one hundred pages of typewritten and diagram matter, was the basis for establishing the budget for 1933–34 and also the basis for policies and a program which the Council adopted and which are summarized in the following paragraphs.

Policies

Dues. The Committee on Policies and Budget made a painstaking analysis of the question of dues reduction. It considered the wishes of many members and earnestly endeavored to find feasible means to reduce the rate of dues. After careful study of all the factors involved, however, the Committee presented as its findings that at the present time a reduction in dues would not make it possible for a sufficiently greater number of members to pay their dues to maintain an adequate income but instead would result in a considerable decrease in income to the Society. These findings were concurred in by a special committee appointed at the conference of Local Section delegates at the 1932 Annual Meeting. The Council, therefore, adopted the policy of taking no action looking to the reduction of dues at this time.

Reductions in Expenses. As economy measures, the following

procedures were initiated for 1933–34. Transactions of only two Divisions are to be distributed per member; the *A.S.M.E. News*, suspended in 1933, and the Membership List, suspended in 1931, are not to be printed in 1934; only two meetings of the Society are to be held; inactive Local Sections and Professional Divisions and certain committees are to be temporarily suspended; traveling allowances are to be reduced; Council and Committee meetings for which mileage and expenses are paid by the Society are to be reduced; the Local Sections Delegates Conference is to be made up of one representative of each of the seven geographical areas of the country, each geographical group to have had its own conference prior to November 1, 1933; the new Student Branch plan is to be installed in only one additional area in 1933–34; no appropriation was made for campaigning for new members; representation on the ASA was reduced from two to one; and the work of the technical committees is to be curtailed.

American Engineering Council. The American Engineering Council is carrying on a highly worth-while work, especially in view of this time of depression. However, the Society requested that its representatives on American Engineering Council urge the adoption of a policy whereby the expenditures of the Council can be reduced to a minimum by a simplification in activities and a substantial reduction in the number of representatives of member organizations.

Library. The Standing Committee on Library and the Society's representatives on The Engineering Foundation were authorized to request The Engineering Foundation to furnish as much as practicable of the A.S.M.E. appropriation to the Library for the next fiscal year with the understanding that the Society waive its request for research aid for new projects.

Employment Service. The Council adopted a policy of urging the Employment Service to approach commercial rates and become self-supporting in normal times. Under present conditions it is the opinion that the rates should be slightly increased.

Engineering Index. The Engineering Index is to be discontinued on January 1, 1934, unless sufficient outside aid is received to remove the possibility of loss to the Society.

Balance Sheet. Every effort is to be made to increase the Society's cash, decrease inventory, reduce borrowings, and write off unpaid accounts and obsolete inventory. The schedule of income and expenses for next year appears in the Finance Committee's report.

Fiscal Policy. The Council also adopted eighteen detailed statements of policy for the guidance of the Finance Committee in the formulation of the budget and for the aid of the Council in administering the fiscal operations of the Society.

Program

(1) Steps are to be taken to upbuild the morale of our Society and to win the enthusiastic support of its members. To that end the aims, objectives, and ideals of our Society will be defined and broad policies for achieving them, enunciated. The Committee on Policies and Budget has this in process.

(2) A definite plan will be adopted to provide regular visits of Officers and Members of Council to Local Sections and Student Branches as funds permit.

(3) The new Student Branch plan is to be extended and further developed as funds permit.

(4) The plan to serve better the interests of our Junior Members is to be further developed to bridge the gap between the Student Member and full membership.

(5) A plan is to be devised to strengthen the Local Sections by bringing into their Executive Committees the outstanding members in each Section.

(6) The Local Sections Committee, assisted by the Meetings and Program Committee, is requested to devise a plan to elevate the character of the meetings of the Local Sections by sending prominent speakers to discuss timely subjects and thus increase interest in the Sections and the Society. To that end, part of the funds used for this should be considered as a part of the appropriations for the Sections visited.

(7) The Committee on Professional Divisions is asked to devise a plan to upbuild and strengthen the Professional Divisions as a means of binding our Society more closely together, increasing our Society's value to its members and increasing their interest in it. During 1933–34 new Divisions should not be launched. The present ones, however, should be encouraged to be more active, and inactive ones be temporarily suspended.

(8) The Local Sections Committee and the Professional Divisions Committee are requested to devise jointly a plan for promoting closer cooperation between these activities for the purpose of increasing the effectiveness and usefulness of each, separately and jointly.

(9) The Publications Committee is asked to give continuing attention that the reader interest in *Mechanical Engineering* be increased to augment its advertising value.

(10) A critical examination is to be made of all activities to determine the advisability of continuing them.

(11) Critical examination is to be made to determine if the activities of other societies and associations are being duplicated and, if so, that those activities be eliminated in our Society if possible. In case elimination does not at first appear feasible, steps should be taken to induce the other societies and associations to conduct those activities so as to permit elimination in our Society.

(12) The Committee organizations are to be reviewed for the purpose of simplifying them, and committees engaged in activities remote from mechanical engineering are to be discontinued.

(13) Steps are to be considered to make membership in the Society mean more to members by raising the professional status of our Society; consideration also is to be given to raising membership to a higher level by more strict entrance requirements.

(14) The Committee on Local Sections is requested to conduct a careful study of the reasons for the large number of members dropped for non-payment of dues.

(15) The committees responsible for Standardization and Safety are requested to give further consideration to the trade association movement with the view toward having the projects carried on by trade associations when feasible.

(16) The Finance Committee is to be charged with the specific duty of acting as a conservator of our Society's funds for the express purpose of preserving the soundness of our Society's financial structure, guarding against undue spending, and insuring the practice of rigid economy.

(17) The Council is requested to establish a Committee on Funds and Endowments, this Committee to be responsible for collecting such Funds.

(18) The American Engineering Council is to be interpreted to our members so that they may better understand its purpose and appreciate its value.

(19) Our Society's organization, management, and procedure is to be made more responsive to and governed more by changes in general business conditions.

(20) The Finance Committee is requested to consider the desirability of entering our investments on the balance sheet at market value instead of at cost.

(21) The Finance Committee is requested to consider the desirability of keeping accounts on a cash basis instead of on accrual basis, as at present.

Summary of Operations in 1932–33 as Shown by Committee Reports

Notwithstanding several drastic curtailments in activities and staff, the year just passed shows a deficit in Society operation. This deficit is largely due to the fact that dues from the members fell far below the most conservative estimate. The Finance Report which follows gives the matter in detail. It should have the careful study of every member.

Despite reductions in income for the current year, the general activities of the Society have been carried on with devotion and enthusiasm. The Annual Meeting in 1932 was a splendid success and the meeting in Chicago, bringing together as it did all of the engineering societies, has not been equaled by a meeting outside New York in many years. The number and quality of meetings held by the Sections, despite reduced appropriations, is highly creditable, and the large number of excellent papers that were secured as a result of the efforts of the Professional Divisions indicates that the Society is very much alive as a technical body.

The Library was obliged to function with a proportionate reduction in appropriation and yet had to meet a demand from seventeen per cent more readers and from some eighteen thousand inquirers by mail.

The excellent program for the development of Student Branches which was started in one area of the country during the previous year was extended to three additional areas (the New England, North Atlantic, and North Central States) during

the past year and will be extended still further to an additional area during the coming year.

The publications of the Society were necessarily curtailed. The Membership List was not published, the *A.S.M.E. News* was suspended, the number of papers that were published was reduced, and savings were made in the publication of the Record and Index by sending it to only those members who requested it. Further savings were made by issuing offset printed papers where the material could not be included in the reduced volume of Transactions and the provision of thousand-word abstracts instead of preprints for the Chicago meeting.

As an evidence of the virility of the Society, during the year the Membership Committee recommended 1670 new candidates for membership and 174 transfers to a higher grade. The comparative figures for the year before were 1123 and 179, and for the year before that were 1396 and 252. It will be apparent that even without concerted effort, the number of new members recommended for election in the Society has increased materially. While the major portion of this increase comes from Junior Members who transfer from student membership under the new plan, this is indeed an excellent showing. The accompanying table shows the actual changes in membership recorded up to September 30, 1933.

CHANGES IN MEMBERSHIP—OCTOBER 1, 1932, TO SEPTEMBER 30, 1933

	Membership		Increases			Decreases				Total Changes		
	Oct. 1, 1933	Oct. 1, 1932	Transferred to	Elected	Reinstated	Transferred from	Resigned	Dropped	Died	Increases	Decreases	Net change
Honorary Members	22	25	0	0	0	0	0	0	3	0	3	−3
Life Members	66	68	1	0	0	0	0	0	3	1	3	−2
Members	8,657	8,878	66	66	2	2	234	0	119	134	355	−221
Associates	515	564	1	4	0	1	45	0	8	5	54	−49
Associate-Members	4,123	4,150	70	48	0	22	96	0	27	118	145	−27
Juniors (20)	1,875	2,076	0	0	0	74	110	16	1	0	201	−201
Juniors (10)	4,942	4,318	0	688	4	24	39	0	5	692	68	+624
	20,200	20,079	138	806	6	123	524	16	166	950	829	+121

The technical committee activities have progressed also in spite of reduced financial support. Two new standards projects were initiated as were two new research projects. All of the Committees report well-attended, enthusiastic meetings in spite of the depression.

Other actions of an important administrative nature include the transfer of the supervision of the Max Toltz Loan Fund from the Awards Committee to the Committee on Relations With Colleges and the decision to defer temporarily the proposed revision of the Constitution and By-Laws to bring them up to date with current procedure. This was also deferred because of the work of the Committee on Policies and Budget which is reexamining the purposes and functions of the Society with the view to clarifying its aims.

The Individual Member

The Council and officers are aware of the responsibility of the Society to its individual members, particularly in these trying times. Always in the foreground of thought is consideration of what can be done for them. Their point-of-view is paramount and as the coming year unfolds, more and more attention will be given to it. This is regarded as the first duty of the Council and the officers.

Reports of Committees

Complete reports of committees follow. They are worthy of careful and discerning study by the members of the Society. In addition, this year there are appended reports on Joint Activities and on the Office of the Society.

Calvin W. Rice, *Secretary.*

Reports of Standing and Special Committees

FINANCE

The Society has passed through a year of strenuous effort in meeting the financial problems brought about by the general business depression. The situation was complicated by an unexpected shrinkage in income to an amount greatly below the conservative estimate against which the budget of expenditures was balanced. In spite of the heavy retrenchment in expenditures that had previously been made, a further drastic reduction during the year was demanded. To give the members the usual high quality of service under such conditions without an appreciable curtailment in activities taxed the resourcefulness of all spending agencies.

In order that the membership may be fully informed regarding the financial operations of the Society during this critical period, the annual financial statements for the past fiscal year ended September 30, 1933, are presented in much greater detail than has been customary in recent years. In addition to the usual statement of income and expenses, and balance sheet, there are given detailed statements of the direct expenses of Standing Committees, the appropriations to joint agencies, the expenses incurred in printing and distributing publications, the expenses of the office of the Society classified both by items and by departments, expenses of The Engineering Index, and the direct and indirect cost of all activities. A statement is also given of the changes in the surplus that were authorized by the Council. By presenting the financial transactions of the Society for the past year in such complete form, the Finance Committee has endeavored to answer all reasonable questions which may arise.

The books of account were audited at the close of the fiscal year by Haskins and Sells, a firm of certified public accountants of nationally recognized standing, which has for several years audited the accounts of the United Engineering Trustees, Inc., and other national engineering societies. The services of this firm were engaged owing to the regrettable death of William J. Struss, who had served the Society in this capacity for many years. The financial statements in this report were prepared from the annual report of the auditors.

Budget Reductions

The balanced budget, adopted by the Council at Bigwin in June, 1932, had to be revised downward twice during the year, due to the unanticipated shrinkage in income. The original budget of $543,500 was 16 per cent lower than the actual expenditures of $644,588.52 during the preceding fiscal year 1931–32. After four months' operation the Executive Committee was compelled to revise the budget in January, 1933, by authorizing a further reduction of 14 per cent, which brought it down to $465,500. The second revision was made by the Council at its meeting in Chicago in June, 1933, when an additional reduction of 8 per cent, to $429,400, was authorized. It was recognized at that time that a further reduction in expenditures would seriously affect major activities during the remainder of the year; furthermore, a large part of the appropriations for many activities had already been spent. An operating deficit at the close of the year was therefore anticipated.

Operating Income

The activities of the Society, and its service to the membership, are dependent upon two main sources of income—advertising and membership dues. Income from advertising is beyond the control of both the management and the membership. The payment of dues is, however, the responsibility of the membership. Both of these sources of income were seriously affected by the general business conditions.

Income from advertising for 1932–33 was 48 per cent lower than for the preceding year, as shown by the comparative statement of

income in Exhibit A. Even though this tremendous reduction in advertising income occurred, the Society would have enjoyed a surplus of approximately $70,000, instead of a deficit of $44,347.57, if all members had paid their 1933 dues. If, in addition, every member in arrears at the beginning of the year had paid the full amount of back dues owing to the Society, the total income would have exceeded the total expenses by approximately $181,000.

Early in 1932, the Council adopted the liberal policy of not dropping members who desired to retain their membership but were unable to pay their dues. The total number of members on the rolls of the Society October 1, 1933, was 20,200 as shown in the table of membership changes on page 20. At the beginning of this year, on October 1, 1932, 5,737 members owed the Society $110,867.36, for dues, of which amount $55,000.00 was set aside as a probable loss. The following tabulation shows the status of the membership on September 30, 1933:

Honorary Members		22
Members of More Than 35 Years' Standing		482
Members Paid in Full		10,445
Members With 1933 Dues Temporarily Suspended		760*
Members in Arrears:		
For 1933 Dues Only	3,112**	
For 1933 and Prior Years' Dues	5,379*	8,491
Total Membership, September 30, 1933		20,200

Upon recommendation of the Finance Committee, the Council considered it advisable to set up a reserve of $115,000.00 for the uncollected amount of 1933 dues, as shown in the income statement, Exhibit A, and to provide for the uncollected amount of prior years' dues by making a deduction of $49,317.89 from surplus, as shown in the surplus statement, Exhibit B. This action resulted in a very small amount of uncollected dues being carried over as a current asset on the balance sheet. As shown in Exhibit C, the asset of dues receivable amounts to only $12,101.56, which is considerably less than the amount which should be collectable.

Almost all of the Society's investments promptly paid the regular rate of interest when due. The total amount of interest and discount received was $14,546.85, which was $6,113.98 in excess of the interest and discount paid.

Operating Expenses

The operating expenses for all activities of the Society were reduced to a greater extent than the income, the percentage of reduction being 33 per cent from a total expense of $644,588.52 in 1931–32 to $432,848.99 in 1932–33.

In the four major groups of expense the percentages of reductions as shown by Exhibit A were: Committee Expense, 21 per cent; Publications Printing and Distributing Expense, 42 per cent; Office Expense, 28 per cent; and Engineering Index Expense, 47 per cent.

For the convenience of those who wish to examine the expenditures in greater detail, the operating expenses are itemized in Schedules 1 to 7, inclusive. Direct appropriations to the joint activities and to Standing Committees are given in Schedule 1. The printing and distribution expenses of each publication of the Society are shown in Schedule 2. The expense of the office of the Society is classified in Schedule 3 according to departments with salaries separated from other expense items; and is itemized in Schedule 4 according to the purposes for which the expenditures were made. The total expense of the Engineering Index is itemized in Schedule 5. The direct and indirect cost of activities is shown in Schedule 6; and the total cost of activities is compared with the budget for 1933–34 in Schedule 7.

Deficit

Although the total operating expenses for the year exceeded the total operating income by $44,347.57, the net deduction from working surplus required for the current year's operations was $31,552.13, inasmuch as the income of $12,795.44 from initiation fees was not considered as a part of operating income but was credited directly to surplus. The primary purpose of building up the surplus during the period from 1924 to 1929 was to take care of such a situation as the Society has been in during this depression.

* Did not receive *Mechanical Engineering* and Transactions.
** Did not receive Transactions.

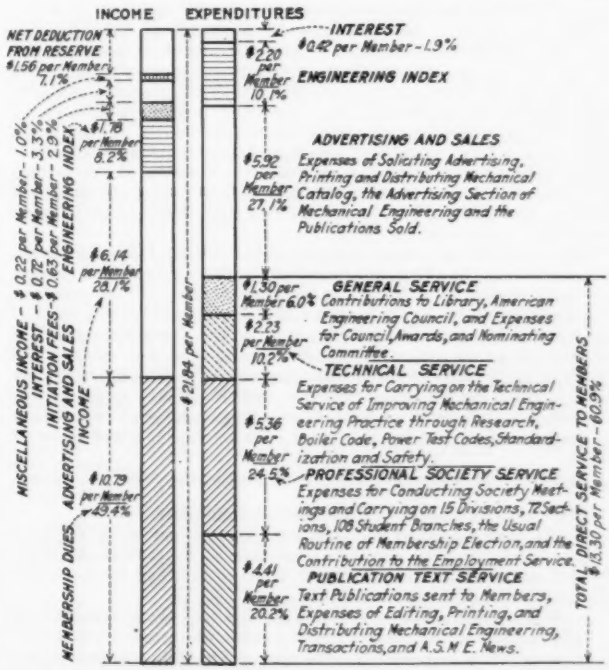
Surplus Changes

The surplus statement in Exhibit B was prepared by the auditors to show the changes made by action of the Council in writing down the current assets resulting from prior years' operations in addition to those changes resulting from operations in 1932–33.

The credits to surplus consist of $12,795.44, initiation fees; $15,000.00, reversal of reserve for depreciation of securities; and $2,456.62, the unused portion of the provision in 1931–32 for the expense of the Mechanical Catalog that year.

The deductions from surplus consist mainly of reduction in receivables and inventories, made for the purpose of clearing the current assets of questionable items that were considered to have doubtful cash value. The write-off of obsolete and inactive items in the inventories of supplies and publications for sale amounts to $78,660.48; the write-off of uncollected dues for prior years amounts to $49,317.89, as previously stated; the write-off of all unpaid accounts contracted for advertising space in *Mechanical Engineering* and Mechanical Catalog, for the purchase of publications, and for subscriptions to the Engineering Index card service, previous to October 1, 1932, from which there seemed to be but slight likelihood of settlements, amounts to $20,384.69; the write-off of obsolete and discarded office furniture and equipment, amounts to $6,984.74. Other items shown resulted in a total charge of $165,576.83 against surplus for prior years' operations, which, together with the deduction for the current year's deficit, brought the surplus down to $205,303.71.

A.S.M.E. INCOME AND EXPENDITURES PER MEMBER, 1932-1933



The following statement regarding the initiation fee reserve is quoted from the report of the auditors: "The Council passed resolutions requiring 50 per cent of initiation fees to be placed in the 'Reserve Fund,' commencing on October 1, 1922, and later adopted an amendment to the By-Laws requiring the entire amount of initiation fees to be deposited in the 'Reserve Account,' presumably commencing on October 1, 1924. If the foregoing actions were to be interpreted as requiring a specific reserve for the initiation fees received during the years involved, rather than a transfer to the Society's Reserve (Surplus) then the amount of such Initiation Fees Reserve at September 30, 1933, would be $258,794.92, based on information contained in the annual reports of the former auditors. The inclusion of such a reserve in the balance sheet would change the Surplus shown therein from $205,303.71 to a deficit of $53,491.21."

Balance Sheet

The final statement in this report is the balance sheet at September 30, 1933, as prepared by the auditors. The details of four accounts in the balance sheet are shown in the following schedules: Schedule 8, Securities Owned; Schedule 9, Trust Fund Securities; Schedule 10, Custodian Funds; and Schedule 11, Trust Fund Reserves.

In order to pay its obligations promptly and to improve the cash situation during the latter part of the year, the Society, like other going concerns, found it necessary to borrow money. Loans on short term notes were advanced by the bank upon the Society's securities as collateral. The bank loan amounted to $164,000.00 at the end of the year. The assessments for the Library, the Employment Service, and rent were paid by notes to the United Engineering Trustees, Inc., during the last nine months of the year. These notes, amounting to $16,353.27, were paid in full in October, 1933.

Because of the large reductions made in the current assets, it was considered inadvisable to increase the current operating deficit by a reduction in the deferred charge for the Fiftieth Anniversary. Likewise it was decided to postpone increasing the reserve for employees' retirement allowances.

Considering the present conditions, the financial position of the Society is sound. Inactive inventories and uncollectable accounts owing the Society have been written off and such large allowances have been made for unpaid dues that the current assets shown on the balance sheet represent a conservative figure. The debt has been reduced. The current assets exceed the current liabilities by $240,635.27. The deferred credit resulting from dues paid in advance exceeds the deferred charges by $11,899.49, and there is a previously accumulated reserve set aside from surplus of $23,432.07 for employees' retirement allowances. The investment in furniture and equipment has been written down to the present market value. The valuation of the Society's quarter interest in the Engineering Societies Building amounts to $496,948.48. The investments represented by securities are carried at cost upon recommendation of the auditors, since such a large portion are in real estate mortgage bonds for which there is no market value at present. However, the soundness of the securities is evidenced by the fact that the bank has accepted them as collateral and further by the fact that they yielded interest amounting to $13,081.19.

With expenditures cut to a minimum by a well-established program of economy, only one thing is needed for the Society to show a surplus each year instead of a deficit, to be clear of debt, and to increase, once more, its reserves. It is that which is vital to the well-being of any membership organization—the loyalty and prompt payment of dues by all its members. While our Society has not escaped the impact of the present economic disturbances, it is being guided safely through these trying times by the resourcefulness and devoted services of the officers, members of committees, and staff upon whom has rested directly the responsibility of administering the Society's affairs.

The Certificate submitted with the Report of the Auditors follows:

> The American Society of Mechanical Engineers:
>
> We have examined your accounts for the year ended September 30, 1933. The accounts receivable at September 30, 1933, and collections during the year, were not verified by confirmations obtained from the debtors.
>
> Subject to the above, it is our opinion that the accompanying balance sheet, and statements of income and expenses and surplus set forth, respectively, your financial condition at September 30, 1933, and the results of your operations for the year ended that date.
>
> [Signed] Haskins and Sells
>
> New York,
> November 24, 1933

Respectfully submitted,

Kingsley L. Martin, *Chairman* — Walter Rautenstrauch
John H. Lawrence, *Vice-Chairman* — D. Robert Yarnall
E. A. Muller

W. L. Batt, H. V. Coes } *Council Representatives.*

EXHIBIT A

COMPARATIVE STATEMENT OF INCOME AND EXPENSES

For Two Fiscal Years Ended September 30, 1933

Income	1932-33	1931-32
Initiation Fees (carried to Surplus)—See Exhibit B	$ 12,795.44	$ 16,288.33
Student Dues	5,667.10	1,158.00
Membership Dues (Less Allowance for Dues Uncollected at September 30, 1933, $115,000.00, September 30, 1932, $55,000.00)	212,289.41	261,137.28
Interest and Discount	14,546.85	16,168.57
Mechanical Engineering Advertising	43,464.10	103,757.32
Mechanical Catalog Advertising	41,213.35	60,807.47
A.S.M.E. News	178.00	802.00
Publication Sales	40,441.16	53,134.03
Engineering Index	35,965.62	63,552.17
Miscellaneous Sales	1,553.53	2,212.07
Contributions Unrestricted	2,845.00	
	$398,164.12	$562,728.91
Less: Bad Debts	1,229.83	2,427.10
Total Income	$396,934.29	$560,301.81
Expenses and Charges		
Committee Expense—See Schedule 1	$ 75,770.24	$ 96,523.86
Publication Expense (Printing and Distribution Only)—See Schedule 2	90,690.78	156,606.36
Office Expense—See Schedules 3 and 4	221,863.29	306,905.89
Engineering Index Expense—See Schedule 5	44,524.68	84,552.41
Total Expenses—See Schedules 6 and 7	$432,848.99	$644,588.52
Interest and Discount Charges	8,432.87	
Total Expenses and Charges	$441,281.86	
Net Loss for the Year (Carried to Surplus)—See Exhibit B	$ 44,347.57	

SCHEDULE 1. COMMITTEE EXPENSE

(Direct Appropriations Only)

	1932-33	1931-32
Council	$ 3,851.44	$ 6,221.91
Engineering Societies Library	8,925.24	10,297.98
American Engineering Council	12,694.96	17,506.65
Sections Committee Mileage	629.91	1,611.21
Sections Delegates Mileage	7,012.79	6,884.14
Sections Operation	15,799.76	23,103.99
Awards	203.57	569.53
Nominating Committee	649.54	897.26
Relations With Colleges—Mileage	471.56	
Research	205.00	300.00
American Standards Association	1,124.98	1,500.00
Divisions Meetings	2,431.15	5,827.73
Divisions Operation	320.72	2,064.47
Divisions Mileage		218.01
Society's Meetings	9,324.25	10,745.72
Engineering Societies Employment Service	8,169.93	6,708.69
Student Branches Operation	3,955.44	2,066.57
Total	$75,770.24	$96,523.86

SCHEDULE 2. PUBLICATIONS EXPENSE

(Printing and Distribution Only)

	1932-33	1931-32
Mechanical Engineering Text Pages	$24,110.06	$ 31,676.16
A.S.M.E. News	2,666.31	9,133.48
Transactions	22,863.83	45,591.20
Membership List		197.63
Mechanical Engineering Advertising Pages	9,416.55	22,836.73
Mechanical Catalog	15,989.11	21,783.92
Publications for Sale	15,644.92	25,387.24
Total	$90,690.78	$156,606.36

SCHEDULE 3. OFFICE EXPENSE*

(Classified by Departments)

	Salary Expense, 1932–33	Other Expense,** 1932–33	Total Expense, 1932–33	Expense Budget, 1933–34
General Department—General supervision, staff service to Council, Executive Committee, Special Committees of Council, Committees on Finance and Publications, central accounting, purchasing, stores, mail, addressograph, duplicating, files, clerical and stenographic	$ 56,844.46	$13,631.80	$ 70,476.26	$ 58,000.00
Field Department—Staff service for Sections, Branches, meetings, Divisions, membership, awards, Employment Service, Chicago and Tulsa Offices	31,053.69	6,224.07	37,277.76	32,000.00
Technical Department—Staff service for Committees on Boiler Code, Power Test Codes, Research, Safety, and Standardization	24,523.35	2,637.03	27,160.38	17,600.00
Editorial Department	23,649.49	791.10	24,440.59	21,000.00
Advertising Department	31,920.50	7,510.08	39,430.58	30,000.00
Publication Sales Department	5,528.46	3,818.62	9,347.08	9,300.00
Rent, Insurance, Depreciation		13,730.64	13,730.64	10,500.00
Relief and Contingencies				1,600.00
TOTAL	$173,519.95	$48,343.34	$221,863.29	$180,000.00

* See pages 49 and 50 for more complete description of duties and functions of the Office of the Society.
** Other Expense includes postage, printed matter, travel, telephone and telegraph, office supplies, stationery, maintenance and repair, and unclassified.

SCHEDULE 4. OFFICE EXPENSE

(Classified by Items)

	1932–33	1931–32
Salaries	$173,519.95	$232,088.79
Commissions		293.95
Postage	8,827.00	11,401.97
Printed Matter	7,862.99	9,457.06
Traveling	4,051.72	8,146.38
Collections	15.15	291.21
Telephone and Telegraph	4,952.52	6,085.55
Insurance	1,360.92	1,965.14
Office Supplies	3,521.57	6,991.14
Secretary's Emergency Fund	32.90	200.66
Rent	11,369.72	14,708.96
Stationery	1,547.50	2,297.34
Maintenance and Repair	573.42	1,101.13
Cost of Miscellaneous Sales	862.69	1,143.52
Depreciation of Furniture and Equipment	1,000.00	1,696.28
Unclassified	3,128.66	11,826.13
	$222,626.71	$309,695.21
Less: Receipts	763.42	2,789.32
TOTAL	$221,863.29	$306,905.89

SCHEDULE 5. ENGINEERING INDEX EXPENSE

	1932–33	1931–32
Salaries	$34,423.16	$ 58,049.11
Postage	1,372.89	3,403.23
Printed Matter	1,774.04	3,069.46
Traveling	1,572.02	3,817.34
Telephone and Telegraph	191.60	503.50
Insurance	53.37	260.78
Office Supplies	1,817.76	2,739.39
Rent	2,933.64	5,541.72
Stationery	90.64	263.62
Maintenance and Repair	167.50	38.52
Paper for Card Service	1,134.61	2,800.37
Distribution of Card Service	839.72	1,425.79
Cost of Annual Volumes Sold	14,498.41	15,719.54
Interest on Loan		6,403.18
Depreciation of Furniture and Equipment	300.00	588.36
Unclassified	355.32	1,962.50
	$61,524.68	$106,586.41
Less: Receipts	0.00	34.00
	$61,524.68	$106,552.41
Less: Accrued Editorial Charge to Annual Volume	17,000.00	22,000.00
TOTAL	$44,524.68	$ 84,552.41

SCHEDULE 6. DIRECT AND INDIRECT COST OF ACTIVITIES

(For the Fiscal Year Ended September 30, 1933)

	Direct Committee Expense	Direct Office Expense	Indirect Office Expense	Total Cost
General Service—Contributions to Library, American Engineering Council, and expenses of Council, Awards and Nominating Committees	$ 26,324.75			$ 26,324.75
Society Activities—Expenses for Society Meetings, Professional Divisions, Local Sections, Student Branches, Membership Election, and contributions to Employment Service	48,115.51	$ 37,277.76	$22,700.00	108,093.27
Technical Service—Expenses of Committees on Boiler Code, Power Test Codes, Research, Safety, and Standardization	1,329.98	27,160.38	16,600.00	45,090.36
Publication Text Service—Expenses of Editing, Printing, and Distributing Text Section of *Mechanical Engineering*, Transactions, and *A.S.M.E. News*	49,640.20	24,440.59	15,000.00	89,080.79
Advertising and Sales—Expenses of Soliciting Advertising, and Printing and Distributing Mechanical Catalog, Advertising Section of *Mechanical Engineering*, and Publications Sold	41,050.58	48,777.66	29,906.90	119,735.14
Engineering Index—Expenses of Editing, Printing, Soliciting and Distributing Card Service and Annual Volume		44,524.68		44,524.68
TOTAL	$166,461.02	$182,181.07	$84,206.90	$432,848.99

SCHEDULE 7. TOTAL COST OF ACTIVITIES

(Comparison of Costs for 1931–32 and 1932–33 With Budget for 1933–34)

	Budget, 1933–34	Actual, 1932–33	Actual, 1931–32
General Service	$ 14,000.00	$ 26,324.75	$ 35,493.33
Society Activities	94,000.00	108,093.27	150,769.97
Technical Service	29,600.00	45,090.36	55,356.48
Publications Text Service (Editorial, Printing, and Distribution)	74,400.00	89,080.79	142,573.54
Advertising and Sales (Soliciting, Printing, and Distribution)	102,047.00	119,735.14	175,842.79
Engineering Index (Editorial, Soliciting, Printing, and Distribution)		44,524.68	84,552.41
TOTAL	$314,047.00	$432,848.99	$644,588.52

EXHIBIT B

STATEMENT OF SURPLUS

For Fiscal Year Ended September 30, 1933

BALANCE, OCTOBER 1, 1932, PER SOCIETY'S ANNUAL REPORT.			$376,706.01
ADD ADJUSTMENTS IN PROPERTY FUND RESERVE TO SEPTEMBER 30, 1932			8,270.04
BALANCE, OCTOBER 1, 1932, AS ADJUSTED			$384,976.05
CREDITS			
Initiation fees		$12,795.44	
Reversal of reserve for depreciation of investments		15,000.00	
Unused portion of provision in prior year for expense of Mechanical Catalog		2,456.62	30,252.06
TOTAL			$415,228.11
CHARGES			
Net loss for the year, per exhibit A		$44,347.57	
Net write-down on account of obsolete and inactive items in inventories, applicable to prior years		78,660.48	
Write-off of, and provision for, prior years' receivables still uncollected, and related adjustments:			
Dues	$49,317.89		
All other	20,384.69	69,702.58	
Write-off of obsolete and discarded office furniture and fixtures, at depreciated values		6,984.74	
Cancelation on account of suspensions, resignations, deaths, etc., of dues billed in prior years		4,497.03	
Relief payments to former employees		4,292.00	
Excess transfer from Life Membership Fund to income in prior year		1,440.00	209,924.40
BALANCE, SEPTEMBER 30, 1933—SEE EXHIBIT C			$205,303.71

EXHIBIT C

BALANCE SHEET, SEPTEMBER 30, 1933

ASSETS

CURRENT ASSETS			
Cash			$ 16,222.09
Advance Receivable—Committee on Thermal Properties of Steam			15,000.00
Accounts Receivable (not verified by confirmations from debtors):			
Dues (less reserves for uncollected items)	$ 12,101.56		
Publications and Advertising	56,061.46		
Engineering Index	6,944.58		
Miscellaneous	250.80	75,358.40	
Inventories (not verified as to quantities):			
Publications for Sale	65,708.87		
Publications in Process	10,351.32		
Supplies	4,359.06	80,419.25	
Securities—At Cost; market value not obtainable ($264,250.00 face value deposited as collateral to notes payable at bank)—See Schedule 8		316,833.89	
TOTAL CURRENT ASSETS			$503,833.63
TRUST FUND INVESTMENTS—			
Securities—At Cost; market value not obtainable. See Schedule 9		166,127.16	
Uninvested Cash		5,700.16	
TOTAL TRUST FUND INVESTMENTS			171,827.32
PROPERTY FUND INVESTMENTS			
One-Fourth Interest in Real Estate and Other Assets of United Engineering Trustees, Inc., Exclusive of Trust Funds		496,948.48	
Equipment			
Office Furniture and Fixtures (depreciated value)		25,577.16	
Library Books		1.00	
Engineering Index		1.00	
TOTAL PROPERTY FUND INVESTMENTS			522,527.64
DEFERRED CHARGES			
Fiftieth Anniversary ($15,000.00 written off prior to October 1, 1931)		24,399.66	
Society Activities Applicable to Subsequent Years		1,135.46	
TOTAL DEFERRED CHARGES			25,535.12
TOTAL			$1,223,723.71

LIABILITIES

CURRENT LIABILITIES			
Notes Payable:			
Bank (See securities pledged, contra)	$164,000.00		
United Engineering Trustees, Inc	16,353.27	$180,353.27	
Accounts Payable		34,763.78	
Unfilled Commitments		18,687.31	
Custodian Funds—See Schedule 10		28,933.18	
Gift for Thurston Biography—Unexpended Portion		460.82	
TOTAL CURRENT LIABILITIES			$263,198.36
TRUST FUND RESERVES (balances as shown by books)—See Schedule 11			171,827.32
PROPERTY FUND RESERVE			522,527.64
RESERVE FOR EMPLOYEES' RETIREMENT ALLOWANCES			23,432.07
DEFERRED CREDIT—DUES PAID IN ADVANCE			37,434.61
SURPLUS, PER EXHIBIT B			205,303.71
TOTAL			$1,223,723.71

SCHEDULE 8: SECURITIES OWNED, SEPTEMBER 30, 1933

	Description	Maturity	Face Value	Cost
Railroad Bonds				
	Baltimore & Ohio Railroad Co. refunding and mortgage 5%	1996	$ 2,500.00	$ 2,473.75
	Baltimore & Ohio Railroad Co. refunding 5%, series D	2000	5,000.00	5,115.00
	New York Central & Hudson River Railroad Co. debenture 4%	1942	25,000.00	23,062.50
	St. Louis, Peoria & Northwestern Railway Co. first mortgage 5%	1948	10,000.00	10,613.89
	Total Railroad Bonds		$ 42,500.00*	$ 41,265.14
Public Utility Bonds				
	Alabama Power Company 5%	1951	$ 5,000.00	$ 5,000.00
	Alabama Power Company 5%	1968	5,000.00	4,975.00
	Binghamton Light, Heat & Power Co. 5%	1946	5,000.00	5,000.00
	Central Maine Power Company 5%, series D	1955	10,000.00	10,100.00
	Commonwealth Edison Company 5%	1953	5,000.00	5,118.75
	Cumberland County Power & Light Company 4½%	1956	5,000.00	4,725.00
	Dallas Power & Light Company 5%, series C	1952	5,000.00	5,000.00
	Metropolitan Edison Company 5%, series C	1953	5,000.00	5,012.50
	Public Service Company of Northern Illinois 5%, series C	1966	5,000.00	4,962.50
	San Diego Consolidated Gas & Electric Co. 5%, series B	1947	5,000.00	5,075.00
	Texas Power & Light Company 5%	1956	5,000.00	4,850.00
	Total Public Utility Bonds		$ 60,000.00*	$ 59,818.75
Lawyers Mortgage Company Bonds				
1- 18199T	S. E. Corner of West 71st Street and West End Avenue, New York City, 5½%	Oct. 6, 1932	$ 50,000.00	$ 50,000.00
2-100172T	E. S. Davidson Avenue 175′ N. from West Clifford Place, Bronx, New York, 5½%	Dec. 15, 1932	10,000.00	10,000.00
26694	2018—75th Street, Brooklyn, New York, 5½%	May 17, 1933	7,000.00	7,000.00
3-100736	S. E. Corner of West 174th Street and Nelson Avenue, Bronx, New York, 5½%	Aug. 15, 1933	10,000.00	10,000.00
29340	1853—76th Street, Brooklyn, New York, 5½%	Aug. 18, 1933	9,000.00	9,000.00
11-100869	N. E. Corner of Arnow Avenue and Wallace Avenue, Bronx, New York, 5½%	Oct. 15, 1933	30,000.00	30,000.00
28- 19530T	S. S. of Walton Avenue 150′ W. to East 181st Street, Bronx, New York, 5½%	Dec. 20, 1936	27,000.00	27,000.00
31-100837T	E. S. Thayer Street 100′ N. Nagle Avenue, New York City, 5½%	Nov. 1, 1933	5,000.00	5,000.00
50188	S. W. Corner of James Street and Footpath, Pelham Manor, New York, 5½%	Apr. 1, 1934	7,750.00	7,750.00
38-200312T	19 Grace Court, Brooklyn, New York, 5½%	Aug. 10, 1935	6,000.00	6,000.00
	Total Lawyers Mortgage Company Bonds		$161,750.00*	$161,750.00
Strauss Company Bonds				
	45 East 30th Street, New York City, first mortgage, serial, 5¼%	1937	$ 45,000.00	$ 45,000.00
	610 Park Avenue, New York City, first mortgage, 6% sinking fund	1940	9,000.00	9,000.00
	Total Strauss Company Bonds		$ 54,000.00	$ 54,000.00
	Total Securities Owned		$ 318,250.00	$ 316,833.89

* Total, $264,250.00—Deposited as collateral to notes payable at bank.

SCHEDULE 9. TRUST FUND SECURITIES, SEPTEMBER 30, 1933

	Description	Maturity	Face Value	Cost
Lawyers Mortgage Company Bonds				
1- 18221T	E. S. of Pinehurst Avenue 433′ 6½″ South from West 187th Street, N. Y. C., 5½%	Oct. 25, 1934	$ 25,000.00	$ 25,000.00
3-101403T	N. W. Corner of Moran Place and Conger Avenue, Bronx, N. Y., 5½%	July 10, 1935	16,500.00	16,500.00
5-100551T	E. S. of Sheridan Avenue 100′ Northerly from East 172nd St., Bronx, N. Y., 5½%	June 15, 1933	7,750.00	7,750.00
9-101577T	W. S. of Montgomery Avenue, 150′ Northerly from West 174th Street, N. Y. C., 5½%	Oct. 20, 1936	3,000.00	3,000.00
10-101264T	N. S. Haviland Avenue, 229.67′ Westerly from Havemeyer Avenue, Bronx, N. Y., 5½%	Sept. 1, 1935	2,000.00	2,000.00
18-200623T	E. S. of Ocean Avenue, 60′ Southerly from Beverly Road and 95′ North of Cortelyou Road, Brooklyn, N. Y., 5½%	Nov. 25, 1936	17,500.00	17,500.00
19-101125T	S. S. of East 233rd Street, 230.02′ Westerly from White Plains Road, Bronx, N. Y., 5½%	Jan. 5, 1935	500.00	500.00
27-100264T	N. S. of Echo Place, 295′ Westerly from Anthony Avenue, Bronx, N. Y., 5½%	Nov. 15, 1932	500.00	500.00
33- 19806T	W. S. of Shakespeare Avenue, 295.07′ Northerly from Boscobel Avenue, Bronx, N. Y., 5½%	Feb. 20, 1935	2,000.00	2,000.00
51832	E. S. of Verdun Avenue, New Rochelle, N. Y., 5½%	July 10, 1935	17,000.00	17,000.00
68- 39729T	E. S. of 81st (24) Street, 135′ Southerly from 35th (Fillmore) Avenue, Queens, N. Y., 5½%	Dec. 20, 1935	25,000.00	25,000.00
73-200614T	W. S. of East 19th Street, 150′ Northerly from Albemarle Road, Brooklyn, N. Y., 5½%	Nov. 20, 1936	2,000.00	2,000.00
181-200651T	N. S. of Winthrop Street, 405′ 7″ Easterly from Flatbush Avenue, Brooklyn, N. Y., 5½%	Feb. 8, 1937	4,000.00	4,000.00
232- 28500T	S. W. of Albemarle Road and Ocean Avenue, Brooklyn, N. Y., 5½%	Jan. 1, 1935	15,000.00	15,000.00
	Total Lawyers Mortgage Company Bonds			$137,750.00
			Number of Shares	
Freeman Fund—Capital Stocks				
	Alabama Power Company 7% cumulative preferred		10	$ 1,080.00
	Aluminum Company of America 6% cumulative preferred		35	2,097.38
	American Power & Light Company $5.00 preferred		25	2,375.00
	Buffalo, Niagara & Eastern Power Corporation $5.00 first preferred		11	1,045.00
	Carolina Power & Light Company $7.00 preferred		10	1,080.00
	Central Illinois Public Service Company $6.00 preferred		21	2,093.20
	Commonwealth & Southern Corporation $6.00 preferred		11⅔	1,030.00
	Consumers Power Company 6.6% preferred		10	1,040.00
	Electric Power & Light Corporation $6.00 preferred		11	1,103.00
	Electric Power & Light Corporation $7.00 preferred		20	1,920.00
	Georgia Power Company $6.00 preferred		13	1,148.33
	Great Western Power Company of California 7% preferred		10	1,020.00
	Minnesota Power & Light Company 7% preferred		10	1,052.00
	Mississippi River Power Company 6% preferred		10	955.00
	New England Power Company 6% preferred		10	1,025.00
	Northwestern Electric Company first preferred		10	1,045.00
	Ohio Edison Company $6.00 preferred		12	1,182.00
	Southern California Edison Company common		18	439.25
	Southern California Edison Company original preferred		40	1,400.00
	Southern California Edison Company 6% preferred		40	1,070.00
	Southern California Edison Company 5½% preferred		44	1,100.00
	Tennessee Electric Power Company 7% preferred		10	1,035.00
	Utah Power & Light Company 7% preferred		10	1,042.00
	Total Freeman Fund—Capital Stocks			$ 28,377.16
	Total Trust Fund Securities			$166,127.16

SCHEDULE 10. CUSTODIAN FUNDS, SEPTEMBER 30, 1933

Absorption of Radiant Heat in Boiler Furnaces	$ 41.50
Alkalinity and Sulphate Relations in Boiler Water Salines	3,251.75
Bibliography and Monograph on Riveted Joints	294.49
Boiler Feedwater Studies	592.64
Boiler Furnace Refractories	50.43
Condenser Tubes	50.20
Critical Pressure Steam Boilers	518.42
Cutting Fluids	2.14
Cutting of Metals	641.86
Diesel Fuel Oil Specifications	186.49
Effect of Temperature on the Properties of Metals	2,562.96
Elevator Safeties	49.36
Fluid Meters	147.04
Heavy Duty Anti-Friction Bearings	137.52
Lubrication	975.45
Measures of Management	76.41
Mechanical Springs	252.01
Methods and Apparatus for Noise Measurement	99.03
Removal of Ash as Molten Slag from Powdered-Coal Furnaces	417.55
Sampling Pulverized Fuel in Moving Gas Streams	100.00
Screw Thread Survey	303.26
Standardisation of Bolts, Nuts and Rivets	1,156.09
Strength of Gear Teeth	548.21
Strength of Vessels under External Pressure	93.94
Stress Analysis of Heads of Pressure Vessels	29.24
Thermal Properties of Steam (including $15,000.00 advance)	15,265.45
Velocity Measurement of Fluid Flow	249.28
Wear of Metals	50.00
Wire Rope	668.01
Worm Gears	122.45
TOTAL	$28,933.18

SCHEDULE 11. TRUST FUND RESERVES, SEPTEMBER 30, 1933

(Balances as shown by books)

	PRINCIPAL	INCOME	TOTAL
Life Membership Fund	.....	....	$ 57,797.87
Freeman Fund	$ 26,544.00	$2,126.14	$ 28,670.14
Worcester Reed Warner Medal Fund	24,661.67	....	24,661.67
Major Max Tolts Fund for Assistance to Students	14,923.04	....	14,923.04
Hoover Medal Fund	13,000.00	....	13,000.00
Holley Medal Fund	6,000.00	1,398.63	7,398.63
Library Development Fund	4,886.06	2,371.09	7,257.15
Spirit of St. Louis Medal Fund	3,500.00	163.50	3,663.50
C. B. F. Waller Trust Fund	3,078.77	251.23	3,330.00
Chas. T. Main Award Fund	2,500.00	692.59	3,192.59
Weeks Legacy Fund	1,957.00	928.57	2,885.57
Melville Prize Medal Fund	1,055.85	913.58	1,969.43
Hess Prize for Juniors Fund	1,000.00	....	1,000.00
Hess Prize for Students Fund	1,000.00	....	1,000.00
A.S.M.E. Research Fund	636.63	....	636.63
Hunt Memorial Fund	212.35	14.03	226.38
Westinghouse Bust and Trust Fund	.....	214.72	214.72
TOTAL (Except Life Membership Fund)	$104,955.37	$9,074.08	$114,029.45
TOTAL TRUST FUND RESERVES			$171,827.32

MEETINGS AND PROGRAM

During the fiscal year, October 1, 1932, to September 30, 1933, only two Society meetings were held, namely, the Annual Meeting in New York, December 5 to 9, 1932, and the Semi-Annual Meeting in Chicago, June 25 to 30, 1933. As the general programs of both these meetings have been amply described in the publications of the Society, they will not be gone into here.

This year's work of the Committee on Meetings and Program has taken full cognizance of the difficult times which have affected our Society no less than they have others.

Need for the curtailment of expenses for meetings has been balanced against the need for continued improvement in the conduct and quality of technical sessions. By means of a questionnaire, valuable comments and criticisms were obtained upon the 1932 Annual Meeting. These have been recorded and have assisted greatly in future planning.

The Committee has cooperated fully with the spirit and purpose of the Special Committee on Policies and Budget. The final allotment of funds to the Meetings and Program Committee by the Council, amounting to $5500* for the fiscal year 1933–1934, will not be exceeded. Two meetings of the Society are contemplated during the coming year. These are the Annual Meeting in New York and the Semi-Annual Meeting, probably in Colorado.

A new subdivision of the work of the Committee has been made and specific responsibilities have been assigned to individual members. Under this plan a close coordination has been secured, and its effect at the Chicago Semi-Annual Meeting was generally beneficial.

One full meeting of the Committee was held in Chicago while the convention was in progress, in order to secure the full advantage of the observations of the committee members.

The policy of the Society with respect to the subject matter and treatment of technical papers was examined during the year, and as a result special important regulations were drafted and sent to all Professional Divisions.

The plan to eliminate the printing of papers in full for the Chicago meeting and to substitute abstracts was, in general, well received. Some of the Divisions which were especially desirous of having their papers printed in full, financed the expense through some of their own members. One case in point was that of the paper by Messrs. Wert, Cope, and Smith, on the design of pipe bends for steam lines, which could hardly have been understood without the complete text. The cost of publishing this paper was borne by the authors' company. The following authors arranged to preprint their own papers: B. D. Stevens, J. Jamieson, Temple Joyce, A. C. Jewett, Dr. H. D. Hubbard, H. R. Isenburger, J. E. Ridder, Cuneo Press, J. H. Taylor and E. O. Walters, R. M. Hardgrove, S. Timoshenko, Eugene Szepesi, Townsend Tinker, Ray Sorenson, E. K. Benedek, and R. F. Onsrud. Also several trade magazines cooperated splendidly by publishing other papers and in several cases supplied extra copies for discussers. The Symposium on Water Hammer was printed and the expense was underwritten by a group of engineers in the Hydraulic Division.

Instructions to presiding officers, authors, and discussers were revised during the year, and a special effort was made to point the technical sessions to higher standards.

As one economy measure, a new plan was introduced at Chicago, which required that participants in technical sessions, attending the authors' breakfasts and luncheons, should pay for their own meals. This change of Society policy was satisfactorily received.

In conclusion, the Committee wishes to report that its analysis of Society affairs in relation to present-day conditions makes it clear that its next year's activities shall, among other things, be especially directed to the energetic promotion of a deeper interest in the whole Society among those of its members not fortunate enough to be able to attend the regular, full Society meetings. There can be no question that the members attending such meetings have the opportunity to see the Society at work and are able to absorb quickly something of its spirit. They are proud to belong to the Society and to give their loyalty to it and to their fellow engineers, and to support the work of improving the status of the profession. This same sense of pride and usefulness in the upbuilding of the profession must be quickened among the men who cannot attend other than the regional or local meetings of their geographical groups.

Respectfully submitted,

J. W. Parker, *Chairman*
C. P. Bliss — E. C. Hutchinson
R. I. Rees — H. N. Davis.

* As compared with $9500 in 1932–1933.

PUBLICATIONS

Upon the Committee on Publications rests the responsibility for adding to the literature of engineering the most important contributions submitted to it through the Society's numerous committees and Professional Divisions. It is charged with the supervision of all of the Society's publications, which makes it the Society's largest single spending agency. And, because of advertising revenue accruing from the publications, the Committee in normal times is also the Society's largest income producer.

As the agency spending the greatest proportion of the Society's income, the Committee on Publications has the very serious obligation of making certain that its business is conducted wisely and with the greatest economy. As chief income producer, the Committee is in the position, in good times, of making it possible for the Society to finance many projects for the benefit of the members in particular and of the engineering profession in general. But, in periods of business stagnation, the severe reduction of the volume of revenue which the Committee can contribute to the Society's income forces the drastic curtailment of activities undertaken in better times. In the present emergency, this curtailment has extended to the very publications that produce the income as well as to the activities supported out of it, with the result that some of them, the Membership List and the *A.S.M.E. News*, have been suspended indefinitely and the Engineering Index faces suspension on January 1, 1934.

This dual role of spender and earner must be understood in order to appreciate the problems and responsibilities that the Committee on Publications has had to face during the last few years. While in 1929–1930 the income from the Committee's activities was 40 per cent greater than that from dues, in 1932–1933 it had shrunk to a point little more than 60 per cent of that from this other source.

In 1929–1930 income from publications more than paid the cost of publications—not one cent of a member's dues had to be used on the publications. Of the $11.80 average dues income per member in 1933, $3.79 had to be used on publications that in former years had been supported entirely by advertising. (Average dues equals receipts for dues divided by total number of members.)

Transactions

As a result of Council action at Chicago, each member will, in the future, receive the Transactions of two instead of three Professional Divisions. Copies of papers published in the Transactions of other Divisions continue to be available upon request. This is an economy measure.

A source of increasing economy, so far as Transactions is concerned, has been found in methods of printing, adopted during the preceding year, and more extensively used since. Preprints and reports, not to be set in type, are reproduced by the photo-offset process in accordance with a standard style. Synopses, with a request blank, are published in *Mechanical Engineering*. Copies are preserved, cataloged, and indexed for the Record and Index.

The Record and Index is a section of the Transactions. Only 5500 copies of the 1932 Record and Index were printed as compared with 21,000 for 1931. The net saving to the Society of $2062 was made possible by distributing the copies only upon request, instead of to every member. In this procedure there was no reduction in service, because a notice was sent to each member with the request that he specify whether or not he wanted a copy.

In the interest of establishing a Society policy for papers to be presented by way of Transactions, and otherwise, the Publications Committee approved the following statement that originated with the Committee on Meetings and Program:

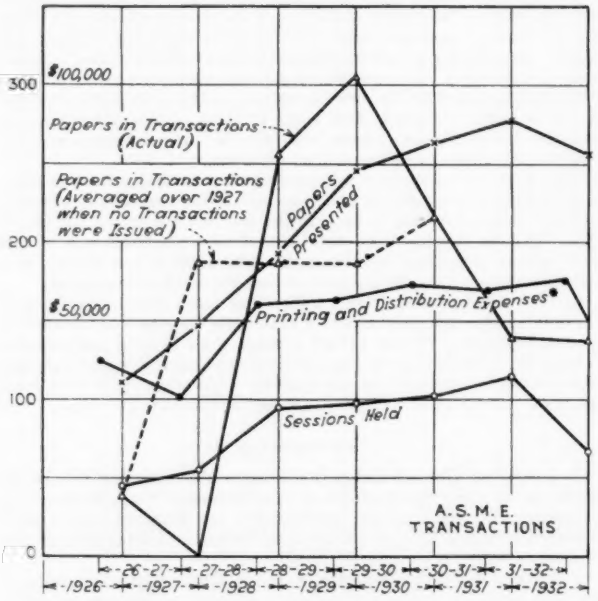
"The Constitution of The American Society of Mechanical Engineers mentions specifically as one of the objects of the Society the promotion of 'the art and science of mechanical engineering and the allied arts and sciences.' This gives wide scope to appropriate subjects for discussion.

"It should be a policy of the Society to accept papers for public presentation and publication on the basis of the merits of the individual paper as determined by a careful review of it by competent and responsible critics and in view of whatever related factors may be of importance at the time; and to open its forums and meetings for the discussion, under competent parliamentary supervision, of such appropriate subjects as can be intelligently and profitably treated. At the same time, the discussion of controversial subjects likely to arouse only fruitless argument and inconclusive debate should be controlled in the interest of economy of time and effort on the part of all concerned. Authority should be vested in the program-making agencies of the Society, in the chairmen who conduct the parliamentary procedure of every public meeting, and in the editor of publications, to enforce with firmness, tempered by a generous and understanding fairness, the spirit of this statement of policy, subject to the final ruling

of Council in cases where this authority is challenged as being improperly exercised."

The present scheme of Transactions was put into effect with the issue of the annual volume for 1928. Statistics of development are shown graphically in the accompanying chart. The change was designed to provide a means for distributing the much greater number of papers that were being presented as a result of the organization of the Professional Divisions. It was hoped that, *without increasing publication costs*, about 50 per cent more material could be published. Actually it was possible to print about 70 *per cent more material* than under the old scheme.

Differences in the fundamental points of view and states of technological and scientific development of the fields of interest represented by the Professional Divisions have naturally resulted in a variety of standards in judging technical papers. Those of the Divisions that are developing fields to which engineering has been only



*Estimated Expense, 1932-33, $29,000

DEVELOPMENT OF TRANSACTIONS, 1926–1932

recently applied are obliged to secure material of a descriptive character while others are presenting material of a highly technical nature or which is the result of valuable researches.

In 1928, 1929, and 1930, when funds were available, the Committee adopted the very liberal attitude of publishing in Transactions practically every paper recommended by the executive committees of the Divisions. This year, however, with almost three times as many papers recommended as there are funds out of which to pay for publication, the Committee has been forced to change this policy and publish only those papers that seem to be of permanent reference value. This means that Divisions and committees developing papers of this character are getting a greater volume of publications than are other Divisions, and it places an added responsibility on the Divisions to insist on high quality in the papers they recommend for publication.

When it established the Applied Mechanics Division, the Society provided a forum for advanced researchers and workers in this field and an outlet for the technical papers they produced. There are no other outlets for technical papers in applied mechanics—no papers published by societies or otherwise that render to engineers working in this field a service such as exists in the fields of power, machine-shop practice, railroads, iron and steel, aeronautics, and most of the others. This is true not only with respect to mechanical engineers interested in this subject, but for engineers of all types. No other engineering society has attempted to provide a forum and publication outlet for these men. The need is a very real one. No commercial publisher can undertake the task because of the relatively small number of men who would be potential subscribers and because their interests and buying power provide no profitable field for the advertiser.

Without the sponsorship and financial assistance of some group like the A.S.M.E., these engineers, whose original researches and mathematical studies in the field of applied mechanics provide the basis for fundamental engineering designs, and whose contributions are greatly enriching the literature for the immediate and permanent benefit of engineering designers, would be denied a necessary means by which they can exchange experience, make contacts with men of similar professional activities, and record the results of their work.

As a step toward providing this group with a periodical commensurate in usefulness and professional standing with the character of the work they are doing, the Committee on Publications has secured a separate post-office entry for the section of Transactions sponsored by the Division and now known as "Applied Mechanics." With the return of better times, the Division looks forward to developing this periodical into a broader and more valuable publication. In addition to papers presented at Division meetings, it is hoped to include other contributions, news of the men in the field and the work they are doing, reports on projects under way and contemplated in various laboratories, reviews of work and publications in other countries, book reviews, and similar features.

The program is an ambitious one, but it merits very serious consideration; the work done by this Division and its members is typical of what can be fostered only by professional societies because its publication is likely never to be commercially profitable. If engineering societies, in times of normal prosperity, cannot support activities of this sort, their right to be considered professional and educational bodies is justly questioned. It seems to us that they will fail to replenish the force that sustains them.

While the plans of the Applied Mechanics Division must be held temporarily in abeyance, awaiting the return of better times, they must receive attention at the earliest possible date. The Committee on Publications bespeaks the enthusiastic backing of members of the Society for this project, as well as for its policy of raising the standards of quality and value in all sections of the Transactions.

"MECHANICAL ENGINEERING"

It is a fact frequently commented upon that *Mechanical Engineering* now definitely aligns itself with the broader aspects of engineering. By far a majority of the comment has been favorable.

The Publications Committee is proud of the issues for this year. During the last few years the whole world has come to realize its regrettable ignorance concerning things social and economic. It is looking for help. Shall the mechanical engineer look to his Society? The Publications Committee thinks that he should, and has gladly endorsed the policy of a discerning editor which you now see put into practice in *Mechanical Engineering*.

Smaller than formerly, this magazine still contains 64 editorial pages. We have felt that fewer would not suffice; advertising revenue has enabled us to refrain from further reduction.

"A.S.M.E. NEWS"

To secure data upon which to base its decision in regard to suspending the *A.S.M.E. News*, the Committee canvassed the Local Sections, by means of a letter sent early in November, 1932, to Local Sections Delegates to the 1932 Annual Meeting Conference; and discussed the matter with the Employment Department, committees whose work might be affected, officers of the Society, and individual members.

A digest of the data convinced us that several of the services being rendered by the *News* could be discontinued, at least until the Society had improved its financial position; that the carrying on of other services could be transferred to *Mechanical Engineering;* and that still others could be conducted by special communications.

Accordingly, the recommendation was made to Council (December 5, 1932) that the *A.S.M.E. News* be indefinitely suspended.

Our formal communication to the Executive Committee of Council made the recommendation contingent upon the inclusion at no extra expense of the material transferred to *Mechanical Engineering;* no further publication of the necrology (except in Record and Index), personals, reports of Section meetings, material relating to Society organization and operation, and other material appearing regularly in the *News;* the transfer to *Mechanical Engineering* of only such material as would permit the maintenance of the existing high standards of quality, prestige, usefulness, and reader interest possessed by that publication.

The recommendation of the Publications Committee was adopted by the Council, and the *A.S.M.E. News* was discontinued with the publication of the December 22, 1932, issue. The saving for the balance of the fiscal year ending September 30, 1933, amounted to $5400. It is estimated that the saving for a full year would be approximately $8000.

Mechanical Catalog

Mechanical Catalog continues to reflect the general lack of business that prevailed up to the middle of the year. It must be remembered that the acceptance of contracts for the Catalog ceases about the middle of the year in order to allow time for editing, setting, proofreading, printing, and binding, to meet the established distribution date of September 30, a condition that prevented increased sales on account of the better business of the latter half of 1933.

Acquaintance on the part of the members of the Publications Committee with the personnel of the advertising sales force of the Society has convinced them that they know their work and that they have done as good a job as conditions would permit.

Membership List

No Membership List was printed during the year 1932–1933. It is the recommendation of the Committee that none be issued for 1933–1934. In addition to the need for economy it seems inadvisable to issue such a list at a time when there are so many changes in the addresses and business connections of members.

Engineering Index

At its meeting in Chicago, the Council voted to suspend the Engineering Index on January 1, 1934, unless sufficient subvention was received to relieve the Society from financial risk. In the meantime, all efforts possible are being made to secure the aid necessary to carry on this valuable agency.

The action looking toward the suspension of the Index was taken with deepest regret. When the Society, in 1919, purchased The Engineering Index from its former publisher, The Engineering Magazine, it exhibited a quality of aggressiveness and a sense of professional obligation that did justice to the high ideals which its founders had for its services to the profession. The maroon-bound volumes that are a familiar sight on the shelves of the best reference libraries will stand as a reminder to future generations of engineers that a serious business depression interrupted a service, the value of which cannot be estimated merely by the number of persons or organizations subscribing to it.

When the Index was established on a comprehensive basis in 1928, thorough studies had been made of the possibilities of developing it as a self-supporting venture. In 1930, with a total income in excess of $100,000, this condition was nearly approached. For the past two years, reductions in income have been effected more rapidly than corresponding reductions in expenses could be made and the Index is still not self-supporting. This loss has been a source of dissatisfaction to members of the Society and this, together with the necessity to sell the service at a high price, were the reasons for the action of the Council. In view of opposition to the Index, vigorously expressed by many members, speaking individually and through their Local Sections' representatives, the Committee on Publications, after having made reductions in operating expenses that would soon be reflected in the quality of the Index, was unwilling to continue its operation at a deficit. Inasmuch as it cannot be operated profitably during the business depression, the Index will be suspended on January 1, 1934, unless outside aid can be secured. Reductions in price of the service are absolutely essential if the Index is to have the widespread usage its value warrants. This cannot be done without outside aid, such as is received by Biological Abstracts and Chemical Abstracts, two comparable services. Every effort is being made to secure the aid necessary to make the Index of wider use and to carry it on without risk to the Society. Final decision will be reached before January 1, 1934.

Advisory Committee

During the year 1932–1933 the Committee presented to the Council a request that it approve the appointment of three advisory members. This request was granted, and the President appointed one from St. Louis, one from Blacksburg, Va., and one from San Francisco. In general, the functions of the advisory members are to comment on all actions of the Committee, to provide advice as requested on specific questions, to canvass the opinions of members of the Society about the publication policies and report to the Committee, and to fulfil any other functions that will aid the Committee in meeting its responsibility in the conduct of the diverse publication needs of the Society.

This plan is considered experimental, the term of appointment of the three members continuing until December, 1934, when the Committee will submit its conclusions and recommendations to the Council.

Budget for 1933–1934

The expenditures for printing and distribution of the publications of the Society for the past year and for the previous year, and the budget for next year are shown in the following table:

PUBLICATIONS INCOME BUDGET

	1933–1934
Mechanical Engineering Advertising	$ 42,000.00
Mechanical Catalog Advertising	40,000.00
Publication Sales	39,000.00
	$121,000.00
Less: Bad Debts	3,000.00
	$118,000.00

PUBLICATIONS EXPENSE BUDGET

(Printing and Distribution)

Mechanical Engineering Text Pages	$ 21,000.00
Transactions	19,000.00
Mechanical Engineering Advertising Pages	8,500.00
Mechanical Catalog	15,500.00
Publications for Sale	13,647.00
	$ 77,647.00

As discussed previously in this report, the volume of Transactions must be reduced if this budget is to be reached. The Transactions of two Divisions only will be issued to each member, instead of three as at the present time. The income from the publications is reduced seriously and the Committee is cooperating with the Council, the Executive Committee, and the Finance Committee in working to this scheme.

Conclusions

In conclusion we present the income and expense statement of Publications as prepared by the Finance Committee. In doing so we wish to emphasize that the amounts have to do with printing and distribution only.

Publications Income

	1932–1933	1931–1932
Mechanical Engineering Advertising	$ 43,464.10	$103,757.32
Mechanical Catalog Advertising	41,213.35	60,807.47
A.S.M.E. News	178.00	802.00
Publication Sales	40,441.16	53,134.03
	$125,296.61	$218,500.82
Less: Bad Debts	1,229.83	2,427.10
	$124,066.78	$216,073.72

Publications Expense

(Printing and Distribution)

Mechanical Engineering Text Pages	$ 24,110.06	$ 31,676.16
A.S.M.E. News	2,666.31	9,133.48
Transactions	22,863.83	45,591.20
Membership List	.....	197.63
Mechanical Engineering Advertising Pages	9,416.55	22,836.73
Mechanical Catalog	15,989.11	21,783.92
Publications for Sale	15,644.92	25,387.24
	$ 90,690.78	$156,606.36

Engineering Index

Income from Card Service and Annual Volumes	$35,965.62	$63,552.17
Expense (Editorial, Printing, Selling, and Distributing)	$44,524.68	$84,552.41

Respectfully submitted,

L. C. Morrow, *Chairman*
S. F. Voorhees, *Vice-Chairman*
S. W. Dudley
W. F. Ryan
M. H. Roberts.
A. J. Dickie
E. B. Norris
E. L. Ohle
Advisory Members.

LOCAL SECTIONS

Scope of Service

It has been urgently necessary for Local Sections to economize, even in the face of increased demands for service to the membership. Some activities, such as Registration of Engineers and the Economic Status of the Engineer, have been temporarily discontinued, in order that Sections may better apply their strength to problems such as the collection of funds for relief and the organization of unemployment committees, and committees to cooperate with Governmental agencies such as the Reconstruction Finance Corporation and the Public Works Administration. The Sections, too, have been urged to cooperate more fully with the Committees on Relations With Colleges, Professional Divisions, and Meetings and Program.

In order to effect economy of administration, the Committee has recommended and urged the combination of the St. Paul and Minneapolis Sections under one Section organization and title of "Twin Cities Section," with territory to cover a radius of sixty miles from the State Capitol.

Operation of Local Sections

The Local Sections of the Society deserve full credit for their loyal support and cooperation in carrying forward to fruition, full programs, in spite of the necessary curtailment of funds. In a number of cases, in order that Society prestige not suffer, Sections have carried on committed programs which have exceeded in cost the Sections' allotments, without calling upon the Society for additional funds. This has been accomplished in the interest of continuing the high standard of Section performance established in the past.

Meetings and Membership Chart

During the past fiscal year the Sections of the Society have held 394 meetings. The number of reported meetings held by each individual Section is shown in Table 1. This table also details the membership of the respective Sections on October 1, 1932, and October 1, 1933; the net increase or loss in membership; etc.

New Members

The potential possibilities of Society membership are big. But with business conditions so unsettled, no drive for new members has been undertaken. On the contrary, every effort has been put forth by Sections to hold and interest our present membership. With the improvement of business, however, the Committee feels some effort should be expended in an attempt to explore this potentially enormous field.

Prospective Sections

No new Sections were added to the rolls of the Society. In keeping with the policy laid down by the Committee on Policies and Budget, no effort has been made to interest members in the organization of new Sections.

There are still a few industrial areas in the United States where new Section possibilities seem to exist.

Measure of Sections' Operation

The Committee has undertaken the development of a Scale of Measure for Sections' Operation, in the effort to set a minimum standard of Section accomplishment. This has been made necessary by the temporary suspension of 14 Local Sections for lack of reported Section activity; and the question of reinstatement of these Sections. In the future, all Sections of the Society will be required to meet a definite minimum requirement of successful operation to continue on the active list of Sections.

Local Sections Visited by Committee

With improved health, William Lyle Dudley, of Seattle, Wash., has been able to resume his duties on the Committee, making desirable contacts with the Sections of Group VII. Due to curtailment of funds, however, the Committee visits have been substantially reduced, as compared to those of past years. It is the unified opinion of the Committee that means should be provided by the Society, not only for Local Section Committee member visits to the Sections, but visits by Council members as well. No other activity of the Society will do more to assist the Sections to operate successfully and to hold the interest of the membership. (See Table 2.)

Mid-West and Mid-Continent Offices

Under the budget set-up for 1933–1934, the Committee felt that it would be unwise to renew the present leases for these offices. Activity and contact, however, in these territories will be maintained.

The Mid-West Office continues to assist the Local Sections and Student Branches within its territory and to serve individual members in every possible way. The contacts afforded by this office accomplish a great deal in the development of professional consciousness and acquaintance with the problems of the profession, and help greatly in the increase of Society prestige.

The Mid-Continent Office has continued its efforts in organized investigation and reports on problems and developments in the oil industry. Public and membership interest in Society undertakings has been greatly increased by the operation of this office. Student activity has been fostered with good results; and assistance in unemployment relief has placed this office in equally as important a position in its territory, as the Mid-West office occupies in the Chicago area.

The Committee feels that these offices and operations should be continued for the welfare and up-building of the Society in these areas.

TABLE 1 MEMBERSHIP IN LOCAL SECTIONS, OCTOBER 1, 1933

Section	Total Membership 10/1/33	10/1/32	Increase or loss	1933 Dues paid	Dues part paid	Dues suspended	Dues unpaid	Basic membership for L.S. approp.	Meetings, 10/31-10/32
Akron-Canton	160	170	− 10	77	10	4	69	91	4
Anthracite Lehigh Valley	269	295	− 26	140	8	5	116	153	7
Atlanta	101	103	− 2	47	7	1	46	55	11
Baltimore	235	232	+ 3	143	11	2	79	156	9
Birmingham	84	89	− 5	34	5	3	42	42	8
Boston	758	775	− 17	476	24	34	224	534	8
Bridgeport	155	195	− 40	93	2	6	54	101	7
Buffalo	236	254	− 18	124	2	6	104	132	8
Central Pennsylvania	108	110	− 2	62	4	1	41	67	6
Charlotte	39	44	− 5	18	1	1	19	20	2
Chattanooga	20	25	− 5	7	1	0	12	8	2
Chicago	1104	1095	+ 9	629	29	38	408	696	7
Cincinnati	292	297	− 5	162	13	9	108	184	6
Cleveland	404	402	+ 2	214	17	11	162	242	6
Colorado	96	100	− 4	62	5	1	28	68	9
Columbus	87	87	0	50	5	1	31	56	5
Dayton	114	124	− 10	71	3	5	35	79	9
Detroit	543	539	+ 4	296	14	19	214	329	4
Erie	93	96	− 3	48	1	2	42	51	5
Florida	86	79	+ 7	46	4	6	30	56	2
Green Mountain	49	49	0	30	2	1	16	33	1
Greenville	37	36	+ 1	21	1	0	15	22	2
Hartford	134	141	− 7	91	4	2	37	97	3
Houston	145	153	− 8	83	3	4	55	90	3
Indianapolis	171	163	+ 8	110	6	6	49	122	5
Inland Empire	21	25	− 4	8	2	0	11	10	6
Kansas City	119	153	− 34	64	8	4	43	76	10
Knoxville	50	50	0	24	2	1	23	27	3
Los Angeles	460	467	− 7	266	21	13	160	300	12
Louisville	37	46	− 9	21	3	1	12	25	7
Memphis	33	37	− 4	17	2	1	13	20	3
Meriden	32	32	0	17	2	5	8	24	2
Metropolitan	5178	5070	+108	2929	189	113	1947	3231	32
Mid-Continent	225	198	+ 27	121	11	7	86	139	3
Milwaukee	275	280	− 5	170	6	7	92	183	3
Minneapolis	82	78	+ 4	53	1	4	24	58	2
Nebraska	54	44	+ 10	34	1	1	18	36	2
New Britain	55	50	+ 5	34	1	3	17	38	2
New Haven	128	127	+ 1	72	5	2	49	79	9
New Orleans	99	102	− 3	59	4	0	36	63	2
North Texas	80	87	− 7	42	1	0	37	43	7
Norwich	36	45	− 9	27	0	0	9	27	7
Ontario	148	183	− 35	82	2	5	59	89	3
Oregon	84	81	+ 3	41	4	0	39	45	6
Peninsula	83	88	− 5	36	2	3	42	41	2
Philadelphia	1173	1264	− 91	710	37	38	388	785	9
Pittsburgh	525	571	− 46	316	12	15	182	343	1
Plainfield	314	298	+ 16	184	10	7	113	201	3
Providence	209	204	+ 5	147	11	5	46	163	6
Raleigh	42	33	+ 9	21	5	2	14	28	4
Rock River Valley	73	82	− 9	30	3	4	36	37	4
Rochester	125	139	− 14	82	6	1	36	89	4
St. Joseph Valley	48	57	− 9	17	2	0	29	19	3
St. Louis	280	295	− 15	163	16	9	92	188	7
St. Paul	42	48	− 6	26	2	2	12	30	3
San Francisco	437	454	− 17	283	10	22	122	315	8
Savannah	22	21	+ 1	17	0	0	5	17	...
Schenectady	217	236	− 19	146	4	9	58	159	3
Susquehanna	68	78	− 10	46	5	2	15	53	4
Syracuse	134	137	− 3	79	5	6	44	90	7
Toledo	85	90	− 5	42	3	5	35	50	14
Tri-Cities	72	79	− 7	47	4	1	20	52	7
Utah	36	36	0	19	3	1	13	23	6
Utica	30	34	− 4	19	0	1	10	20	1
Virginia	188	185	+ 3	116	5	7	60	128	2
Washington, D. C.	217	222	− 5	153	11	5	48	169	7
Waterbury	87	92	− 5	64	5	0	18	69	4
Western Massachusetts	150	145	+ 5	88	10	5	47	103	4
Western Washington	127	114	+ 13	59	10	3	55	72	11
West Virginia	75	73	+ 2	50	2	2	21	54	2
Worcester	172	187	− 15	110	8	6	48	124	8
Youngstown	79	85	− 6	47	3	1	28	51	0
	17,826	18,155	−329	10,332	641	497	6356	11,470	394

Membership in Society, October 1, 1933, 20,200.
Membership in Sections, October 1, 1933, 17,826.
Membership in Society, October 1, 1932, 20,057.
Membership in Sections, October 1, 1932, 18,155.

Cooperation With Special Committee on Policies and Budget

The Committee on Local Sections has done everything possible to maintain the Sections at best efficiency, despite curtailment of funds, and has cooperated in every way with the Special Committee on Policies and Budget. Realizing that the Local Sections form the links of the Society's chain, the Committee on Local Sections urges that at the very earliest possible moment, full and adequate financial assistance again be accorded the Sections.

In order to keep expenditures within the allowed budget, in connection with the next Delegates' Conference, there will be a series of Regional Conferences by Groups and only one Delegate will be sent from each Group to the Annual Meeting at New York. These Group Delegates will confer at New York with members of the Council and Professional Divisions. Substantial economy will thus be effected in the cost of the Section Delegates' Conference.

Cooperation With Student Branches

The Local Sections of the Society encourage in every way Student Branch activity. Realizing their important obligation to Student Branches, they have entered actively into the work. Many Sections have established Student Prizes, to win and hold undergraduate interest. The District Student Branch Conferences in the Groups where the new student policy is in force, were held under the auspices of Sections in these Groups. These Conferences do much to promote closer relationship between Section membership and Student Branches.

Cooperation With Professional Divisions

The cooperation of Local Sections with Professional Divisions has consistently been developed, co-relation of activities being increased by placing both activities under one administrative head.

The Sections have availed themselves of every opportunity for such inter-relationship of opportunities.

It will be seen that due to budget curtailment, these activities have likewise had to be substantially reduced. With increase of business and increase of Society activities, the Committee feels this cooperative effort should be fostered.

Tours and Transportation

The Committee on Local Sections arranged for reduced fares to the Annual, Semi-Annual, and other national gatherings of the Society during the year, thereby saving thousands of dollars to the membership.

In connection with the Chicago Meeting, the Century of Progress Exposition and other points of engineering interest in and about Chicago were made more easily accessible to the members through the efforts of the Mid-West Office and the Chicago Section of the Society.

Foreign Relations

For a number of years foreign relations have been handled under the supervision of the Local Sections Department because the actual help rendered foreign visitors is developed through the Local Sections themselves. Last year, as previously, many of the Local Sections cooperated by giving considerable time and attention to engineers from abroad. In spite of the general conditions there was a greater number of visitors than in the previous year. Forty-four visitors from twelve countries were received and assisted. Eight of these were students who had just been graduated from the engineering course from Liverpool University, England.

In addition, letters of introduction to foreign engineering and technical societies, as well as to specific plants, were given to a large number of members of the Society who made trips abroad.

Cooperation With Publications Committee

The Committee is so strongly of the opinion that a new membership list is needed, that it has urged the Publications Committee to endeavor to publish such a list for the year 1933–1934.

The Committee further urges that in view of the economic pressure on some members, a column of *Mechanical Engineering* be devoted to "Men Available" as previously published in the *A.S.M.E. News*.

Cooperation With Meetings and Program Committee

It has been suggested by the Committee, and recommended to the Meetings and Program Committee, that all papers for presentation by members at National Meetings, be first presented before Section or Professional Divisional Meetings. It is felt that both Sections programs and National programs would be helped by such a procedure.

Respectfully submitted,

J. M. Todd, *Chairman*
J. W. Haney — W. W. Macon
W. L. Dudley — R. E. W. Harrison.

TABLE 2 LOCAL SECTIONS VISITED BY MEMBERS OF THE COMMITTEE ON LOCAL SECTIONS, 1932–1933

Section	J. M. Todd, *Chairman*	J. W. Haney	W. L. Dudley	W. W. Macon	R. E. W. Harrison	Ernest Hartford, *Secretary*
Atlanta	.....	.....	.....	.....	.....	Mar. 1933
Birmingham	.....	.....	.....	.....	.....	Mar. 1933
Chicago	June 1933	June 1933	June 1933	June 1933	June 1933	June 1933
Cincinnati	Sept. 1933	Sept. 1933	Sept. 1933	Sept. 1933	Sept. 1933	Sept. 1933
Indianapolis	Oct. 1932	.....	.....	.....	.....	.....
Metropolitan*	Dec. 1932	Dec. 1932	.....	(Resident)	Dec. 1932	(Resident)
Milwaukee	.....	.....	.....	.....	.....	Oct. 1932
Nebraska	.....	(Resident)	.....	.....	.....	Nov. 1932
New Haven	.....	.....	.....	.....	.....	May 1933
New Orleans	(Resident)	.....	.....	.....	.....	.....
Rock River Valley	Oct. 1932	Oct. 1932	.....	Oct. 1932	(Resident)	Oct. 1932
St. Louis	Oct. 1932	.....	.....	.....	.....	.....
Tri-Cities	.....	.....	.....	.....	.....	Nov. 1932
Virginia	.....	.....	.....	.....	.....	Apr. 1933
Western Washington	.....	.....	(Resident)	.....	.....	.....
Worcester	.....	.....	.....	.....	.....	Oct. 1932

* Annual Meeting.

PROFESSIONAL DIVISIONS

The Divisions have completed a year of most successful activity despite reduced appropriations. In order to support the great Society meeting at the Chicago Century of Progress in June, 1933, most of the Divisions provided comprehensive programs for that gathering and relinquished the privilege of holding separate Division meetings through the year. The result was a technical program for Chicago of fine quality and excellent diversity.

The study being conducted under the leadership of the Committee on Policies and Budget looking to a critical reexamination of the activities of the Society has met with the approval of the Committee on Professional Divisions, which is cooperating wholeheartedly. Some of the immediate problems relating to the Divisions are raised in this report.

The reports of individual Divisions appended are well worth careful study, as they demonstrate forcefully the fact that many individuals are giving of their time and effort, through the work of the Professional Divisions, in advancing the purposes of the Society.

Chicago Meeting

The holding of many technical sessions by the Divisions at the Chicago Meeting had the effect of consolidating the National Meetings of most Divisions. The resulting technical program was most excellent, equaling in extent and quality that of an Annual Meeting. However, curtailment in appropriations for publications brought about the need for issuing 1000-word abstracts in place of the usual reprints and made somewhat more difficult the problem of securing discussion in advance and of providing advance information to those attending the meeting. While there was much approval of the new scheme, because of the compactness of the book of abstracts there was some objection, which was relieved by various expedients. The trade press cooperated in some instances by printing papers in advance. Some authors provided preprints, and a few Divisions arranged the papers in pamphlet form for sale. This scheme of abstracts has been used by other societies with success and must, by reason of the lack of funds, be tried at the Annual Meeting in 1933. The Committee regards it as a necessary temporary expedient and believes it advisable to return to the scheme of preprints as soon as possible.

Other Meetings

Four meetings were held in addition to the Semi-Annual Meeting at Chicago and were successful in programs and attendance and were economically handled. These special meetings consisted of the following: Pacific Coast Applied Mechanics Meeting, January 20–21, in cooperation with California Institute of Technology and the Los Angeles Section; Air-Conditioning Conference sponsored by the Air Conditioning Committee of the Process Industries Group and the San Francisco Section, held February 9–10 at the University of California; National Lubrication Engineering Meeting, held at Pennsylvania State College, May 25–26, in cooperation with the College, the Central Pennsylvania Section, and the Lubrication Engineering Committee of the Petroleum Division; National Oil and Gas Power Meeting, held under the auspices of the Oil and Gas Power Division of the Society at Atlantic City, August 23–26.

Fundamental Problems

Present Organization and Relations With Other Societies. Are the Divisions properly organized to cover the technical developments in the field of mechanical engineering? This question is frequently raised, primarily because of the fact that some Divisions, as Textile and Petroleum, deal with a specific industry, while others, as Management and Machine Shop Practice, take up matters of interest to several industries. Various solutions have been advanced, such as consolidation of Divisions or turning over activities to related societies or associations that may be carrying on technical activities in fields of the individual Divisions.

In this connection there are some fundamentals we must keep clearly in mind. Our Divisions originate from demands for activity in given fields. New developments and new problems must arise in a changing technical civilization. Our members must turn to their Society for help in these developments and problems, and the groups having common interest in them must be provided with adequate forums for mutual discussion. Our organization should be flexible enough to provide for growth and rapid change. Furthermore, we must provide for interchange of experience and knowledge between the separate groups. Formal organizations must go on, and it is our policy to be continuously on the watch to bring about mutual understanding between the various groups both in the Society and outside. Committees that clear problems between Divisions have been organized, and close cooperation with other technical and industrial groups is actively fostered.

The Standing Committee and the Divisions are constantly alert as to these problems and the changes that may be required. At the present time, however, the Committee considers that our form of Division organization meets current needs as well as any other that may be suggested.

Reducing Registration in Divisions. As an economy measure the Council at the Chicago Meeting authorized the reduction from three to two as the number of Divisions in which a member may register. The present form of publication of Transactions by Division Sections was initiated in 1928 to serve our membership better by publishing more material. It was considered that many members of the Society would want and need only certain papers that the Society published, and that it would be wasteful to send them all the papers. Each member was permitted to select three Divisions from which he would automatically receive the papers published. If he wanted papers published in other Divisions he could get them without charge by requesting them when *Mechanical Engineering* announced that they were available. The reduction, now, from three to two merely means that a member will more often have to send in requests for papers, especially for those in the third Division that he has been receiving automatically. The Society still stands ready to furnish without charge to every member a copy of every paper published in the Society's Transactions.

Summary of Division Activities

The Aeronautic Division, during the present year, held its National Meeting as part of the Semi-Annual Meeting of the Society in Chicago, June 26 and 27, 1933, presenting twelve papers; assisted Local Sections in holding Aeronautic meetings in New York, Boston, and San Antonio; and held a session at the Annual Meeting.

In the latter part of 1932 there was organized a new aeronautical engineering society under the name of the Institute of Aeronautic Sciences. The first technical gathering of this new organization was held in New York, N. Y., on January 26, 1933, at Columbia University. The Aeronautic Division and the Society extended their greetings and offered their cooperation. The members of the Institute were invited to participate in the aeronautic program of the Chicago Meeting.

At the Chicago Meeting, the A.S.M.E. Special Committee on Aircraft Safety and Inspection, which was initiated by the Division, presented a further progress report. During the year the Society, at the request of the Aeronautic Division, recommended to the American Standards Association that the Aircraft Safety Code be revised.

For the year of 1934 the Aeronautic Division will hold its National Meeting on the Pacific Coast at the University of California.

The Applied Mechanics Division held its National Meeting in Chicago as part of the Semi-Annual Meeting of the Society, presenting twelve papers in joint sessions with the Structural Division of the American Society of Civil Engineers on Thursday and Friday of that week. Besides this, the Division held one session independently and two sessions on plasticity jointly with the Iron and Steel Division, making a total of seven sessions at the meeting in which the Division cooperated and for which it secured papers. At the 1932 Annual Meeting, the Division presented a symposium on working stresses which was unusually successful.

The Division held a Pacific Coast Applied Mechanics Meeting in cooperation with the Los Angeles Section and the California Institute of Technology at the Institute on January 20 and 21, 1933, at which nine papers were presented. The meeting was very successful, with about two hundred in attendance, and gave sufficient evidence of the need and advisability of occasional meetings on the Pacific Coast by this Division.

Several sessions are planned for the 1933 Annual Meeting, while for the coming year of 1934, the Division will actively cooperate in the International Congress of Applied Mechanics to be held in Cambridge, England, June, 1934. Cooperation in the summer meeting of the Society for the Promotion of Engineering Education at Cornell University is being considered.

The Division has six special subcommittees: Elasticity, Strength of Materials, Plasticity, Dynamics, Mechanics of Liquids and Gases, and Thermodynamics.

The Fuels Division held its National Meeting as part of the Semi-Annual Meeting of the Society in Chicago, presenting eight papers at four sessions. At the 1932 Annual Meeting the Division had two sessions, one of which was held jointly with the Iron and Steel Division, and it is sponsoring a session at the 1933 Annual Meeting. The Pure Air Committee has continued its activities. The work of its Subcommittee on Standards for Setting Heights has now been organized into a committee under the ASA. The special subcommittee of the Division on Boiler Furnace Explosions was disbanded, as there did not seem to be sufficient demand for specifications on explosion doors or information as to how to write such specifications.

The Hydraulic Division sponsored a splendid Water Hammer Symposium at the Chicago Meeting of the Society and also cooperated in two other sessions with the Power Division of the A.S.C.E. The Division sponsored two sessions at the 1932 Annual Meeting and is planning a water measurement symposium for the 1933 Annual Meeting.

Due to the decreased income of the Society, the Hydraulic Division's Water Hammer Committee took care of the publication in a separate pamphlet of the Water Hammer Symposium. This special committee not only did the splendid work of developing the symposium and coordinating the mathematical symbols but also underwrote the cost of publication.

For 1934 a Pacific Coast hydraulic meeting is being planned, to be held at Berkeley, California.

The Iron and Steel Division held three sessions at the 1932 Annual Meeting, one of which was in cooperation with the Fuels Division. At the Chicago Semi-Annual Meeting two sessions were held in cooperation with the Plasticity Committee of the Applied Mechanics Division. A joint meeting with the Pittsburgh Section and the Engineering Society of Western Pennsylvania was held in April, 1933. The Division some years ago initiated the Special Research Committee on Heavy-Duty Anti-Friction Bearings and this committee has now completed some very valuable research work and submitted its final report at the Chicago Meeting. The Division is planning further cooperation with the Plasticity Committee on deformation of metals. For the 1933 Annual Meeting, one session is planned and the Division will cooperate in the Heat Transfer Symposium.

The Machine Shop Practice Division held three sessions at the Chicago Semi-Annual Meeting and sponsored a similar number at the 1932 Annual Meeting. A representative program has been arranged for the 1933 Annual Meeting.

During the past year the Division has organized a Subcommittee on Welding, this committee to act jointly with the American Welding Society. In addition to sponsoring sessions on welding and initiating new activities in this field, this Committee is charged with the responsibility of keeping the members posted on the latest developments in welding.

Like the Welding Committee, the Subcommittee on Foundry Practice acts jointly with the American Foundrymen's Association.

This Subcommittee has continued its excellent activity and has sponsored sessions at all meetings.

The Division has a third Subcommittee on Machine Design which is developing its program of activity.

Divisional cooperation with the Special Research Committee on Cutting of Metals has successfully kept members informed on the latest developments in this field.

The Management Division sponsored two sessions at the Semi-Annual Meeting in Chicago and also cooperated in two sessions of the Econometric Society. At the 1932 Annual Meeting the Management Division sponsored three sessions and it will also sponsor several sessions at the 1933 Annual Meeting. During the last year, the Division has had a number of active subcommittees, among which are, (1) Waste Elimination Committee, which assisted in a Chicago Session; (2) Engineering Economics Committee, which sponsored sessions at the Annual Meeting; (3) the Society's Special Research Committee on Measures of Management, in cooperation with the Materials Handling Division, which is active in sponsoring papers each year; (4) the Job Shop Management Committee, which is continuing a survey it started last year; (5) the new Marketing Committee, which is working on a session for the coming Annual Meeting and has undertaken a survey to determine the present status of Marketing Research in Industry; and (6) a Special Metropolitan Section Committee of the Division, which sponsors informal evening management conferences in New York several times a month, during the fall, winter, and spring. There is also a Subcommittee on Maintenance which was not active during the year. The Engineering Economic Committee cooperated in the Third Annual Economic Conference for engineers held at the Stevens Institute of Technology Engineering Camp. The Division is represented on the International Management Institute and also on the new National Management Council, which was formed this year to coordinate management activities.

The Materials Handling Division sponsored at the 1932 Annual Meeting a joint session with the Railroad Division on materials handling on railroads and will also sponsor a session at the 1933 Annual Meeting. The Division organized a special Subcommittee on Foundry Handling with the American Foundrymen's Association and held a joint session at the American Foundrymen's Convention in Chicago, the Friday preceding the Semi-Annual Meeting. In January, the Division cooperated in two sessions held by the Materials Handling Institute at the National Road Building Congress in Detroit.

The Oil and Gas Power Division held its National Meeting at Atlantic City, N. J., August 23 to 26, presenting eleven technical papers and having an equipment exhibit in conjunction with the meeting. Preprints of the papers were made available at the meeting through the courtesy of a trade magazine. The Division had one session at the 1932 Annual Meeting and also will hold a session at the 1933 Annual Meeting. There is one special Subcommittee on Oil Engine Power Cost which presented its fifth yearly report at the Atlantic City Meeting. This report was for the year 1932 and covered 140 plants in comparison with 110 in the 1931 Cost Report. The Division formed a Subcommittee on Diesel Fuel Oil Problems which had several meetings during the year.

The Printing Industries Division held its National Meeting in Chicago as part of the Semi-Annual Meeting; presenting seven papers and including two informal discussions and one group conference. This Division each year conducts a forum for the discussion of the problems of the industry, to which all of the organizations and associations in the industry send delegates. Their Chicago sessions formed the Fourth Conference of the Technical Experts in the Printing Industry and were among the best attended of the Semi-Annual Meeting sessions. The proceedings of the meeting are being issued in a special pamphlet form by the Division. At the Chicago session a program of Cooperative Printing Research in America was presented by A. C. Jewett and received the endorsement of the meeting. For several years the Division has been encouraging the possibility of a printing research institute, which has resulted in the presentation at Chicago of this formal program. During the year the Division lost by death its Chairman, Floyd E. Wilder, who had been very active in organizing the work of the Division for the last few years. John Clyde Oswald was appointed chairman for the remainder of the year.

The Petroleum Division has a number of special subcommittees. Several of these are located at Tulsa, Okla. They are as follows.

The Fluid Flow Calculations Committee has started a series of tests on three separate operating crude-oil trunk lines of six, eight, and ten inches diameter.

The Subcommittee on Oil Metering Problems of the Special Research Committee on Fluid Meters has continued its work on the collection of data to determine the correctness of existing coefficients and correction factors for the flow of fluids of varying viscosities and temperatures through orifice plates, flow nozzles, and venturi throats at various velocities.

The Production Committee has completed the set-up and obtained some data on preliminary runs on the capacities of slush pumps handling mud fluids of various weights and at different pressures and velocities.

The Special Research Committee on Prime Movers for Rotary Drilling has completed its third study on the various types of prime movers for rotary drilling. The studies completed are the combination Gas-Electric Prime Mover, Steam Equipment for Rotary Drilling, and Electric Equipment for Rotary Drilling. The two studies that remain to be accomplished are Gas Engine and the Diesel Engine Prime Movers.

The Lubrication Engineering Committee held a National Meeting at Pennsylvania State College, May 25 and 26, in cooperation with the college, presenting nine papers. It was voted at the meeting to ask the A.S.M.E. to petition the American Standards Association to form a committee on the classification of industrial lubricants. Pending the formation of the committee by the ASA the Lubrication Engineering Committee will continue to gather information on classification of industrial lubricants. Another special committee was appointed to act as a clearing house on problems of application of lubrication. The Lubrication Engineering Committee sponsored one session at the 1932 Annual Meeting and will also sponsor one at the 1933 Annual Meeting.

The Petroleum Refining Committee this year issued a short test sheet to go with its Recommendations on Testing of Heat Exchange Equipment for Oil Refining Service. The committee is also cooperating with the Process Industries Committee in arranging the Heat Transfer Symposium for the coming Annual meeting. The Petroleum Utilization Committee sponsored a Metropolitan Section meeting.

The Process Industries Committee sponsored two sessions at the Chicago Meeting and has planned a Heat Transfer Symposium for the 1933 Annual Meeting. It has appointed a special Subcommittee on Heat Transfer Symposiums which it is anticipated will develop sessions for a number of meetings. Other Divisions and societies are cooperating in these Heat Transfer Symposiums. The Process Industries Committee was responsible for initiating the Society's Special Research Committee on Cotton-Seed-Oil Processing and a report on the work of the Committee was made at one of their sessions in Chicago.

The Subcommittee on Drying sponsored a Metropolitan Section meeting and the Pulverizing and Grinding Committee sponsored a session at Chicago.

The Committee on Air Conditioning sponsored a Pacific Coast Air Conditioning Meeting at San Francisco under the auspices of the San Francisco Section. The Section arranged for the printing of the papers in a special pamphlet form for sale. The Conference was very successful and attracted considerable interest.

The Power Division held two sessions at the last Annual Meeting, one on Industrial Power and one on Central Station Power, and three sessions at the Chicago Meeting. Sessions are also planned for the 1933 Annual Meeting. Besides this, several of the Society's special research committees in the power field hold sessions at the Semi-Annual and Annual Meetings, among these being Boiler Feedwater, Power Test Codes, and Steam Tables Research. The Division has a special Subcommittee on Industrial Power which holds a luncheon meeting during the Annual Meeting and at the 1932 meeting there were two reports presented, one on Power Plant Costs and Records, by W. W. Shoudy, and the other one on Power Plant Betterment, by D. M. Myers.

The Railroad Division held a research symposium of two sessions at the Semi-Annual Meeting at Chicago and is arranging to print the papers presented there in special pamphlet form. The research symposium was of unusual merit and attracted considerable attention. At the 1932 Annual Meeting the Division sponsored two sessions and cooperated in a third session with the Materials Handling Division. The Railroad Division organized a special Subcommittee for the Chicago District known as the Chicago Group. This group is composed of five members and a secretary. Its formation marks an important advance in the organization work of the Division, owing to the importance of Chicago as a railroad center. For the past year, the Executive Committee has been engaged in revising the organization and activities of the Division. A number of new and unique features have been incorporated into its organization, some of which are worthy of study and consideration by other Divisions of the Society.

The Textile Division held a session at the 1932 Annual Meeting and also sponsored an evening meeting in the Metropolitan Section. It also assists New England and Southern Sections in arranging textile

meetings. At the 1933 Annual Meeting a session and a luncheon meeting will be held.

The Wood Industries Division held its National Meeting as part of the Semi-Annual Meeting in Chicago and presented six papers. A Symposium on Wood Preservation which attracted considerable attention was held in the spring in New York. The papers of this Symposium are being prepared by the Division in pamphlet form for distribution.

The Wood Industries Division is planning the reorganization of two former Special Research Committees, one on Saws and Knives and the other on the Existing Supplies of Hardwood. It is the intention of the Division to have these carry on as subcommittees of the Division until such time as they may warrant being again organized as research committees. The Division also has a special survey research committee which is planning surveys of research problems.

Respectfully submitted,

P. T. Sowden, *Chairman*
C. B. Peck, *Vice-Chairman*
K. H. Condit
W. A. Shoudy
G. B. Pegram.

MEMBERSHIP

The Committee on Membership held twelve regular meetings and eleven by proxy during the fiscal year 1932–1933. In addition to special and miscellaneous matters placed before the Committee for consideration and recommendation to Council, the following applications were given attention in the transaction of the Committee's work.

Applications pending, October 1, 1932	255
Applications received during fiscal year 1932–1933	1757
Total applications handled during year 1932–1933	2012
Recommended for membership	1844
Transfers denied	1
Deferred	4
Cancelled	3
Incomplete	12
Applications pending, October 1, 1933	148
Total applications handled during year 1932–1933	2012

The 1844 recommended for membership were divided into the following grades:

Members	86
Transfers to Member	77
Associates	5
Transfers to Associate	2
Associate-Members	66
Transfers to Associate-Member	95
Transfer from Student Member to Associate-Member	1
Juniors	136
Transfers from Student Associate to Junior (R5-Rule 1)	379
Transfers from Student Member to Junior (R5-Rule 1)	997
Total recommended	1844
Total new members recommended	1670
Transfers (above Junior grade)	174
Total recommended	1844

During the fiscal year 1932–1933 the Membership Committee made the following recommendations:

Elections declared void	134
Resignations accepted	418

Respectfully submitted,

Hosea Webster, *Chairman*
O. E. Goldschmidt
H. A. Lardner
R. H. McLain
C. L. Davidson
H. W. Butler, *Advisory Member.*

CONSTITUTION AND BY-LAWS

Four meetings of the Committee held in December, 1932, were devoted to a study of the possibility of simplification of the By-Laws and Rules and to the consideration of suggestions the Committee had invited from the chairmen of the various committees and the staff, which indicated the advisability of a considerable number of changes. While the Committee found little opportunity for any material reduction in volume of the By-Laws and Rules, short of omissions that would leave some of the activities of the Society without definition and control, a complete revision is advisable in order to simplify and clarify the material, to improve the grouping of items, and to eliminate minor inconsistencies. The Committee has prepared a draft of the major part of the By-Laws and Rules in recommended form, but this draft cannot be completed until decisions are reached by those interested in a number of items.

In the meantime, the Executive Committee of Council, at its meeting on June 8, 1933, directed the Committee to discontinue its work on the proposed revision because of present conditions, and the Committee is therefore awaiting the pleasure of the Council before proceeding further.

During the year certain changes in the By-Laws and Rules have been presented to the Committee on Constitution and By-Laws for consideration. These amendments, approved by the Committee, are now before the Council for action, and include, first, a revision of the Code of Ethics; second, a change in the manner of electing delegates to the American Engineering Council by placing the choice of these delegates in the hands of the A.S.M.E. Council; third, a provision in the By-Laws and Rules for the Local Sections Delegates' Conference; and, fourth, a change in the wording regarding Life Membership payment.

Respectfully submitted,

T. C. McBride, *Chairman*
R. S. Neal
P. R. Faymonville
H. H. Snelling
J. A. Hall.

AWARDS

Faced once more with the impossible task of comparing practically incomparable papers for Society awards, the Committee on Awards took certain steps to have some of its duties handled more efficiently.

Papers submitted for the Student Awards were referred to the Committee on Relations With Colleges. This Committee accepted the responsibility because of its potentiality for improving its college contacts.

The Charles T. Main Award papers were referred to the Pure Air Committee because this year's subject was "Progress in the Prevention of Smoke and Atmospheric Pollution."

In order better to administer the Max Toltz Loan Fund, the Committee recommended to the Council that this responsibility be transferred to the Committee on Relations With Colleges. This was approved by the Executive Committee, September 22.

Dexter S. Kimball, past-president, was appointed to fill the unexpired term of D. B. Rushmore.

Announcement of all awards for the year 1933 will be made at the Annual Meeting in December.

Respectfully submitted,

K. H. Condit, *Chairman*
F. L. Eidmann
D. S. Kimball
W. L. Batt
R. C. H. Heck.

RELATIONS WITH COLLEGES

During the past college year the Committee on Relations With Colleges has extended the new plan of student membership into the New England, North Atlantic, and North Central States by personal visits to 48 engineering colleges. Without exception, the new plan has been adopted by the Student Branches. The result has been a student membership of 1900 in four out of seven divisions of the United States and four regional conferences—at Bethlehem, Birmingham, Chicago, and Worcester—with an average student attendance of more than 200 at each.

The support of the Student Branches and the membership total have been very gratifying to the Committee, particularly as college students have felt the depression severely. The dues from Student Members have been sufficient to balance the operating expenses of

the new plan within $43, which is less than one per cent of the budget. The Committee is confident that the student membership will carry its own division of the Society financially when all college Branches are under the new plan and economic conditions have improved. Even in the desperate financial situation of the past year an average of 30 students were enrolled in each of the 65 Student Branches functioning under the new plan. Of the approximately 1900 Student Members of the Society, composed of both Juniors and Seniors, and in some cases Sophomores, 1000 were transferred upon graduation to the grade of Junior. In view of the economic conditions the Council has extended the period during which these students may qualify in order to give them an opportunity to secure employment.

Under the new plan an engineering student attains regular membership in the Society, with the grade of Student Member, upon the approval of his application by the Council. Student Members receive *Mechanical Engineering* and other Society literature and hold their own conferences, at which they present their own papers. They are not only informed of engineering society affairs but taught active participation in them. The Committee on Relations With Colleges believes that this plan will annually introduce several thousand Junior Members to the Society as soon as it has been completely installed, and that these Junior Members will be prepared and anxious to take part in Society work.

This Committee appreciates the help of the Committee on Local Sections, which is not only fostering interest in the Student Members but making careful plans for retaining and developing their support as Juniors. The recommendation of the Committee on Policies and Budget, approved by the Council, that the student plan be extended as rapidly as financially possible, is an expression of confidence greatly appreciated by your Committee. The plan will be extended to the engineering colleges of Ohio, West Virginia, and Western Pennsylvania before the 1933 Annual Meeting.

The Committee has agreed to take over the responsibility for certain activities heretofore administered entirely by the Committee on Awards. These include the administration of the Max Toltz Loan Fund, which is available to engineering students, and the recommendation regarding the winners of papers submitted for the Student Awards.

Respectfully submitted,

D. B. Prentice, *Chairman*
E. F. Church, Jr. — E. W. O'Brien
W. L. Abbott — R. V. Wright.

EDUCATION AND TRAINING FOR THE INDUSTRIES

The reduction in the number of workers in industry during the past five years, particularly in the machine industries, brought about reductions in training programs for these industries. The sessions arranged for 1933 by the Committee were planned to give the present status of education for industry in various sections of this country, and to show the trend of such education. The subjects for these sessions were proposed by the Committee in 1932 under the chairmanship of Dean R. L. Sackett. One paper on the training of workers for industry in times of depression was also included. Following are the programs for the Semi-Annual Meeting and for the Annual Meeting.

Semi-Annual Meeting. "Education Within Industry in the Central States," by Robert H. Spahr, director, Instruction and Curriculum Development, General Motors Institute of Technology, Flint, Mich.; "Training Programs for Apprentices in Times of Depression," by John T. Faig, president of Departments, Ohio Mechanics Institute, Cincinnati, Ohio.

Annual Meeting. "Training Within Industry in the East," by Ovid W. Eshbach, American Telephone & Telegraph Co., New York, N. Y.; "Training Within Industry in the South," by Theodore S. Johnson, professor of industry, North Carolina State College of Agriculture and Engineering, Raleigh, N. C.

The Committee on Education and Training for the Industries confines its activities to sessions on training and education of non-college type for the industries, in order that its activities may not cross into the field of the Committee on Relations With Colleges. There is peculiar need for a forum for the presentation of education and training for the industries in our Society, and this our committee attempts to offer. In times of depression, training programs in many of the small and medium sized companies are altogether abandoned, or are reduced to skeleton organizations. When economic conditions improve, the personnel charged with the revival of the training program is frequently the reflection of the interest and enthusiasm of one of the executives of the plant, whose influence may be lost when he leaves the organization or when his increasing responsibilities leave little time for the problems of education and training.

Some years ago, when the volume of business was much larger, papers on this important subject were presented at many regional sessions, as well as at the Semi-Annual and Annual Meetings. More recently they have been offered only at these two meetings. However, many business organizations are daily losing some workers through death or retirement, and relatively few apprentices are being taken on, so that when business is again better, it is likely that a considerable dearth of workers will be experienced in some lines. We are therefore approaching a time when our Committee should become more active. It is wise for the Society to carry on with its program in education and training for the industries, so as to maintain the interest of specialists in these lines in the Society and to provide the opportunity for an exchange of ideas.

Respectfully submitted,

J. T. Faig, *Chairman* — S. S. Edmands
H. S. Falk — R. H. Spahr.
C. F. Bailey

LIBRARY

The Library Committee, as representatives of the Society upon the Library Board of the Engineering Societies Library, presents the following report.

The depression of the past year has brought no lessening of demands upon the library. On the contrary, unoccupied time and the need for information upon new lines of activity have brought members to it in larger numbers than before. Thirty-three thousand readers, seventeen per cent more than in the previous year, have visited it, and more than twelve thousand others have used it by mail. The readers have taxed the seating capacity on many days, although additional chairs were provided during the year.

Literary resources have also increased. Books and pamphlets to the number of 7800 were added, bringing the total up to 145,000.

The library has suffered severe financial restrictions during the year. Its budget has decreased from $54,500 in 1931 to $42,450 in the current year. This heavy reduction in the appropriation for its maintenance has necessitated drastic economies. To bring the cost of operation within the amount appropriated it has been necessary to reduce salaries, employ fewer workers, buy fewer books, and shorten the hours open. The Library Board has endeavored to meet this perplexing situation of increasing use and decreased funds in a way that will cause the least inconvenience to members.

If the library is to maintain its rank, as a place where all important periodicals and new books are to be found, additional funds are urgently needed. Increased use brings greater demands, which cannot be met unless funds for books and sufficient personnel to attend to inquirers are provided. It is earnestly hoped that ways may be found to enlarge the service given, and that any necessity for further curtailment of the work will be voided.

Respectfully submitted,

W. M. Keenan, *Chairman*
G. F. Bateman — G. F. Felker
E. P. Worden — Calvin W. Rice.

RESEARCH

The wisdom of the founders in providing that the encouragement of original research should be one of the principal objects of the Society has been fully demonstrated over the years, even these latter years of economic depression. Interest in the 25 projects sponsored by the A.S.M.E. has been sustained on the part of the engineering and industrial firms mostly concerned and especially on the part of the 450 members of the committees in charge of these problems.

During the year two new projects were authorized by the Council. The first of them was proposed by the A.S.M.E. Process Industries Committee and covers a study of the mechanical problems in conditioning, cooking, and pressing of cotton-seed meats. The second deals with methods of sampling pulverized fuel in moving gas streams. This latter project was endorsed by the A.S.M.E. Fuels Division, and while the difficulties of the problems involved are fully recognized, the Committee is receiving the full support of those members of the Society most interested. The A.S.M.E. Research Committee's part in the initiation and development of these research projects is of necessity limited to that of a helpful influence. The Committee members and the staff confer with the Special Committee members and give them the benefit of their years of experience in the organizing and financing of such activities.

The progress toward the completion of the study of the thermal properties of steam begun in 1921, has been steady during the year. The experimental work at the Massachusetts Institute of Technology and the Bureau of Standards has advanced satisfactorily so that January 1, 1934, has been set for the completion of this project. The Fluid Meters Committee has also been active. It has supplied the inspiration and some of the money for experimental work at the Ohio State University and the University of Oklahoma on flow coefficients for oil and gas measurement. The American Gas Association has cooperated in the first of these series of tests and the American Petroleum Institute is showing interest in them also. The important Part 3 of the Fluid Meters Report on Selection and Installation of Meters has been completed and will be published by the Society in December.

Business conditions notwithstanding, the Joint Research Committee on Boiler Feedwater Studies has been successful in financing two important researches closely related to the conditioning of boiler feedwater. One subcommittee has collected $5000 from industry this year and has an equal amount pledged next year for a two-year study of the alkalinity and sulphate relations in boiler water salines. The experimental work has been under way at the Bureau of Mines Non-Metallic Minerals Experiment Station, New Brunswick, N. J., since March. Another subcommittee has developed test methods for the determination of carbonates, hydroxides, and phosphates in boiler waters from experimental work at the University of Michigan. This study was financed by the Engineering Foundation* and the new methods are being published by the Society for trial by industry preliminary to their formulation as standard test procedures by the A.S.T.M. for referee and plant control purposes.

In the metallurgical field, the Special Research Committees on the Cutting of Metals, on Mechanical Springs, on Vessels Under External Pressure, and on the Effect of Temperature on the Properties of Metals have advanced their programs consistently. Progress reports on the experimental work and technical papers, prepared by members of the several committees and the research workers employed by the committees, have been presented at the meetings of the A.S.M.E. and other societies. Among the reports presented are (1) a code for the design and construction of unfired pressure vessels subjected to external pressure and (2) tentative test codes for short-time and long-time high-temperature tensile tests for metals.

Organization and Financing

The accumulated experience of the A.S.M.E. Research Committee, which has existed continuously over a period of 25 years, has shown that the Society can most effectively meet the needs of its membership in the field of research by sponsoring specific researches that are of broad interest to the engineering profession and industry, and that are especially suited to cooperative study. By virtue of its prestige, impartial character, and freedom from commercial influence, the Society is continuously demonstrating its ability to promote industrial cooperation in the solution of common problems. It is able to exert a coordinating influence and to prevent unnecessary duplication of effort.

This sponsorship of cooperative research by the Society carries with it certain obligations; there are implied obligations of service to the special committees, contributed funds must be held and carefully administered so that contributors receive full value for their money, and the provisions of the contracts entered into by the special committees with laboratories and research workers must be fulfilled.

As in the past, this year the Standing Committee on Research continued to assist the various Special Committees with their plans for financing their projects. The small grant of $300 which the Council made this year was used to aid three committees to carry on. It was possible, however, for the Committee to secure six grants from the Engineering Foundation, totaling $8000, for a corresponding number of Special Committees sponsored by the Society. This financial assistance has prompted industry to contribute, as in the past, from three to four times the total expenditure of the Society for this activity. This year the amount raised from outside sources was $40,500.

Employment

A fair number of jobs and highly technical training are provided the research workers employed through the practice of the committees in placing experimental programs in university and institutional laboratories. During the past year 25 men have been at work, full or part time, at the following laboratories: Bureau of Standards at Washington and Columbus, Bureau of Mines at Pittsburgh and New Brunswick, Battelle Memorial Institute, the Universities of Illinois, Michigan, Ohio, Oklahoma, and Tennessee, Carnegie Institute of Technology, Purdue University, and the Massachusetts Institute of Technology.

Publications

One tangible return to the members of the Society from its activity in the field of cooperative engineering research is represented by the reports and papers contributed by the committees to the technical sessions and the publications of the Society. Approximately twenty-five separate articles amounting to some 120 printed pages of original technical material were contributed during the past year to *Mechanical Engineering*, the Transactions, and the technical press. These cover problems in many of the specialized fields of mechanical engineering. In addition, reports and papers amounting to 400 typewritten manuscript pages were produced by the committees, some of which will be published if the necessary funds are available.

The texts of the following special publications were completed and will be published in pamphlet form during the first few months of the new fiscal year: Fluid Meters Report, Part 3, on "Selection and Installation," a group of reports on the "Determination of Carbonate, Hydroxide, and Phosphate in Boiler Waters," and collected "Research Reports and Papers for 1933." The "Report of Tests on Electrical Equipment for Drilling Rotary-Drilled Oil Wells" was issued in April.

International Cooperation

The amount of correspondence and the exchange of reports, and personal contacts, with members of committees of sister societies abroad, have been as in other years. The most important societies participating this year in this exchange are the Institution of Mechanical Engineers and the Department of Scientific and Industrial Research of Great Britain, the Société des Ingénieurs Civils de France, and the Verein deutscher Ingenieure. To Laurence V. Benét, managing director of Hotchkiss and Co., Paris, is due the credit for the development of our research relations with the French engineers, and he has promoted these relations during the past year. A prominent visitor was Dr. Ing. Friedrich Schwerd of the Technische Hochschule, Hannover, Germany, who presented a paper on his metal cutting research work at the 1933 Semi-Annual Meeting at the invitation of the Special Research Committee on Cutting of Metals.

The following two committees were discharged within the period of this report since they had served their purposes: the Special Research Committee on Existing Supplies of Hardwoods and the Special Research Committee on Methods and Apparatus for Noise Measurement.

Special and Joint Committee Reports

Lubrication, A. E. Flowers, Chairman. The Committee has selected a group of 14 original papers on lubrication, many of which are out of print or difficult to obtain, and has prepared abstracts of them. It is the Special Committee's recommendation that these be combined and published with a bibliography of two or three hundred important references on lubrication which it has collected.

Fluid Meters, R. J. S. Pigott, Chairman. Fluid Meters Report, Part 3 on Selection and Installation was completed and will be published in December, 1933, for sale by the Society. A revision of Part 1 on Theory and Application which will greatly broaden its scope, is under way.

The joint A.G.A.-A.S.M.E. subcommittee is analyzing the results of its test program, conducted by Prof. S. R. Beitler at Ohio State University, from which it expects to clear up existing discrepancies in the values of orifice coefficients for gas measurement. The American Petroleum Institute may cooperate in the sponsorship of the resulting handbook.

Tests at the University of Oklahoma under Prof. W. H. Carson to determine the accuracy of existing coefficients and correction factors for the flow of oil through orifice plates, flow nozzles, and venturi tubes have been hampered somewhat during the year by the proposed movement of the Engineering College from Norman to the State Agricultural and Mechanical College, at Stillwater.

Thermal Properties of Steam, Alex Dow, Chairman. At the Massachusetts Institute of Technology, Dr. F. G. Keyes completed the correlation of specific volume data obtained from his experimental work and has nearly completed that on compressed liquids. He is engaged in the measurement of the constant temperature and Joule-Thompson coefficients and the comparison of the Kelvin with the platinum resistance temperature scale. The establishment of this relation is necessary in order to provide a basis for the correlation of the experimental data.

The saturated liquid and vapor measurements at the Bureau of Standards under Dr. N. S. Osborne are to be completed and correlated by January 1, 1934. This will complete the experimental pro-

* The A.S.M.E. is represented on The Engineering Foundation and also upon the Division of Engineering and Industrial Research of the National Research Council. For the reports of these representatives see pages 45 and 47.

gram which has been financed by the Committee since 1921 at a total cost of $175,000 at three laboratories.

Dr. J. Havlicek presented the results of three years' valuable work at the Masaryk Academy of Work, Czechoslovakia, to the Committee's 1932 Annual Meeting session. It included nearly 800 measurements of the enthalpy of compressed water and superheated steam over a wide range of temperature and pressure. Work at M.I.T. and the Bureau of Standards was also reported on at this session.

It is hoped that economic conditions will permit the holding of the Third International Steam Tables Conference next year in the United States as was planned for this year. Much new data have been collected by the research workers here and abroad which should make it possible to narrow considerably the present tolerances of the international skeleton tables.

Strength of Gear Teeth, R. E. Flanders, Chairman. The investigation of the surface fatigue limits under wear and dynamic loading for materials used in the manufacture of helical gears has occupied the attention of this Committee during the year. Prof. Earle Buckingham and graduate students at the Massachusetts Institute of Technology employed the Lewis Gear Testing Machine owned by the Committee for this purpose. Lack of funds has hampered progress.

Cutting of Metals, F. C. Spencer, Chairman. The Subcommittee on Metal Cutting Materials under the chairmanship of Coleman Sellers, 3d, reports that the further introduction and development of shop practice in the use of cemented carbide cutting tools has been greatly retarded by the present business conditions. Nevertheless, the Subcommittee has kept the members of the Society informed of progress in this field through papers it sponsored at the 1932 Annual and 1933 Semi-Annual Meetings.

Another subcommittee, under the chairmanship of King Hathaway, has been successful in collecting and correlating a large volume of published and unpublished data on (1) speeds and feeds in machine operations, (2) choice of tool steels, and (3) heat treatment and grinding of tools. It is the purpose of this Subcommittee to develop material for a series of practical handbooks for the shop based on the principles advanced by F. W. Taylor, Carl G. Barth, and others but applied to modern tool and work materials.

In December, 1932, C. J. Oxford and Prof. O. W. Boston presented a final report of the Subcommittee on Cutting Fluids on experiments conducted for the Committee at the University of Michigan.

Mechanical Springs, J. R. Townsend, Chairman. Dr. D. J. McAdam, at the Bureau of Standards, is completing an extensive report summarizing the present state of the knowledge of the fatigue characteristics of spring materials. A section of this report was presented as a paper for general discussion at the 1933 Semi-Annual Meeting of the Society and it is planned to publish the complete report shortly.

Business conditions have slowed up the tests on fatigue life of helical and wire springs conducted by several cooperating industrial laboratories for the subcommittees under the chairmanships of C. T. Edgerton and F. P. Zimmerli.

Prof. M. F. Sayre at Union College has completed his report on the elastic behavior of spring materials and will present it at the 1933 Annual Meeting in December.

Effect of Temperature on Properties of Metals (Joint Committee with the American Society for Testing Materials), H. J. French, Chairman. Tentative test codes for short-time and long-time high-temperature tensile tests were completed and submitted to the sponsor societies for publication in June, 1933.

Results obtained from the experimental programs financed by the Committee at the University of Illinois under Prof. H. F. Moore and at Battelle Memorial Institute under Dr. H. W. Gillett will be reported at the 1933 Annual Meeting. Comparisons of creep and fatigue characteristics of cast and wrought 18:8 chromium-nickel steels at temperatures up to 1200 F will be available then. Business conditions have practically halted similar studies that were being carried on by cooperating industrial laboratories.

A subcommittee on which are represented the American Petroleum Institute, the American Association of Steel Manufacturers, and the Joint Committee was organized and is studying problems relating to high-temperature applications of metals in oil refineries.

The Committee assists in the review and abstracting of current high-temperature literature which is published monthly in *Metals and Alloys*.

Condenser Tubes, A. E. White, Chairman. During the past year this Committee has continued to encourage studies in certain central-station plants and to collect and report industrial experience in fighting condenser-tube deterioration. A report and two papers were presented at the 1932 Annual Meeting last December.

Boiler Feedwater Studies (Joint Committee with the American Boiler Manufacturers Association, American Railway Engineering Association, American Society for Testing Materials, American Water Works Association, and Edison Electric Institute), S. T. Powell, Chairman. A subcommittee under the chairmanship of C. H. Fellows has completed within the year the development of test methods for the determination of carbonates, hydroxides, phosphates, and sulphates in boiler waters. These test methods are the result of experimental work conducted at the University of Michigan and financed by a grant from the Engineering Foundation. They are to be given preliminary publication by the A.S.M.E. and when they have proved their reliability they are to be recommended to A.S.T.M. Committee D-19 as the bases for preparing standard referee and plant control test procedures.

With the aid of funds supplied by the Joint Committee, Prof. C. W. Foulk has continued his experiments at Ohio State University on priming and foaming of boiler waters. He presented a report at the 1933 Semi-Annual Meeting of the Society on the effect of suspended solids in promoting foam.

During the year, a subcommittee under the chairmanship of J. H. Walker has raised more than $5000 and has pledges of an additional amount from industry for a two-year study of alkalinity and sulphate relations in boiler water salines. The experimental work is progressing satisfactorily at the Bureau of Mines Non-Metallic Minerals Experiment Station, New Brunswick, N. J., and a preliminary report will be made at the 1933 Annual Meeting.

Boiler Furnace Refractories, W. A. Carter, Chairman. Again this year the study of the principal components of the slag refractory system has been continued at the Bureau of Standards Ceramic Station, Columbus, Ohio, under T. A. Klinefelter. Due to a lack of funds the Committee was obliged to withdraw financial support from this experimental program about the middle of the year, but the Bureau is continuing the work at its own expense. A progress report is in preparation.

Elevators, M. H. Christopherson, Chairman. Lack of funds compelled this Committee also to stop its buffer test program at the National Bureau of Standards on March first. However, the extensive equipment will be kept intact and studies of undercar safeties will be undertaken when business conditions improve. The Committee has continued to meet throughout the year.

Strength of Vessels Under External Pressure, W. D. Halsey, Chairman. A draft of a proposed code for the design and construction of unfired vessels subjected to external pressure was completed by the Committee. It will be recalled that this project was originally proposed by the A.S.M.E. Boiler Code Committee. So, within a few weeks, this tentative proposed code will be submitted to the A.S.M.E. Boiler Code Committee for review. This proposal applies only to the three main types of vessels in wide use and operated under external pressures not in excess of 500 lb per sq in. and temperatures not in excess of 700 F.

Absorption of Radiant Heat in Boiler Furnaces, W. J. Wohlenberg, Chairman. Considerable data on the measurement of furnace temperatures has been made available to the Committee during the year by twelve central stations. A report is being compiled by an engineer under the direction of the chairman at Yale University.

Velocity Measurement of Fluid Flow, W. F. Durand, Chairman. As teaching duties have permitted, Professors H. E. Hartig and H. B. Wilcox at the University of Minnesota have continued the development and testing of their methods of utilizing sound waves for the measurement of velocity of fluid flow.

Measures of Management, W. E. Freeland, Chairman. The Committee has continued its activity of collecting and reporting information on management research going on throughout the country. It is in touch with ten such projects under way in industry and universities. Prof. E. D. Smith and L. C. Lichty of Yale will report the results of their investigations of the human aspects of technological development at the Committee's session during the 1933 Annual Meeting. A round table conference will also be held at which Dr. Shewhart of the Bell Telephone Laboratories will discuss developments in the control of quality and its practical significance to the inspection function in manufacture.

Diesel Fuel Oil Specifications, L. H. Morrison, Chairman. Owing to the Government economy program, the tests to discover the effect of variations in the viscosity and Conradson carbon of fuel oils on Diesel engine operation have proceeded slowly at the U. S. Submarine Base, New London, Conn. However, the laboratories of several cooperating oil companies and engine manufacturers have investigated the ignitability of fuel oil in a new type of experimental engine.

Wire Rope, W. H. Fulweiler, Chairman. The Bureau of Standards completed tests on 150 samples of worn wire rope contributed by building operators, the mining industry, the U. S. Navy Bureau of Construction and Repair, the Panama Canal, etc. Funds to cover these tests were made available by the Engineering Foundation. The preliminary report indicates that a definite relation exists between wear and the number of broken wires in a rope and its remain-

ing strength. It is expected that this report will be published about the first of next year. The tests are being continued.

Heavy Duty Anti-Friction Bearings, W. Trinks, Chairman. The program which the Committee laid out for itself in 1929 was brought to a successful completion during the year within the limit of funds contributed by industry—which in the three years of the investigation amounted to more than $7500.

A paper on "Strength of Roll Necks," summarizing the work of the preceding year, was presented by the Committee at the 1932 Annual Meeting. The Ninth Progress Report, "Comparative Study of Various Types of Roll Neck Bearings," and the Tenth Progress Report (final), "Roll Neck Bearings," were completed during the present year and distributed to committee members and financial subscribers. An abstract of the final report was presented at the 1933 Semi-Annual Meeting in Chicago, and the entire report is now being revised for publication and sale by the Society.

While the Committee plans no further experimental work at present, it will continue to function as a clearing house for information on rolling and roll neck bearings. An investigation of plastic deformation is also being considered in cooperation with the Plasticity Committee of the Applied Mechanics Division.

Removal of Ash as Molten Slag From Powdered-Coal Furnaces, K. M. Irwin, Chairman. Samples of coal, slag, and fly ash received from twelve stations have been analyzed at the Bureau of Mines Experiment Station in Pittsburgh, Pa., cone fusions made, and flow temperature determined in air in the electric furnace. Cone fusion values are important for operators to know, as they indicate the probable qualities of slag which coals will give that have not been previously tried in slag tap furnaces.

A study has been nearly completed on the effect of the state of oxidation of the iron in slags on the flow temperature. The ratio of silica to alumina is another variable affecting flow temperature that has been studied.

The fluxing effect of some materials—principally limestone—as dependent on the state of the iron in the slag as fixed by the reducing effect of the furnace atmosphere is to be studied. The principles of absorption of fly ash on a molten slag surface are also to be briefly investigated.

The Committee presented a progress report at the 1932 Annual Meeting and will present another at the 1933 Annual Meeting of the Society.

Automatic Oil Pipe Line Pumping Stations, W. G. Heltzel, Chairman. Economic conditions have temporarily halted the research program of this Committee, but the same personnel is functioning in the way of holding meetings on subjects of vital interest to the pipe line industry just now. A pipe line symposium on Air Compressor Explosions in Diesel Engine Driven Pumping Stations was held in Tulsa on May 12, and a pipe line conference for the purpose of arriving at a suitable formula for the calculation of fluid flow in pipes on May 13, 1933. The Committee is at work on a report on the Economic Analysis and Costs of Pipe Line Pumping Stations. When finances permit, the Committee expects to improve its experimental station on the basis of the experience gained in field stations that were built with it as a model.

Rotary Drilling of Oil Wells, D. L. Trax, Chairman. At the University of Oklahoma, Prof. W. H. Carson made tests on an experimental set-up of boilers and slush pumps, such as are used in steam rotary drilling, and collected data for an addition to the Committee's Steam Rotary Drilling report published by the Society last year.

An investigation of the performance and efficiency of a gas-electric type of prime mover while employed in the drilling of a rotary-drilled oil well was made. The data are now being computed and compiled in the form of a report similar to the one on Steam Rotary Drilling.

The "Report of Tests on Electrical Equipment for Drilling Rotary-Drilled Oil Wells," supplementing the previous report for Steam Equipment, was published in April, 1933, by the Society.

Critical Pressure Steam Boilers, A. A. Potter, Chairman. A report was presented, at the 1932 Annual Meeting, covering enthalpy determinations made on steam at high pressures and temperatures in the experimental boiler. The results were in line with the Keenan and new German steam tables.

Tests with water and steam were conducted to secure data on friction losses during flow in pipes of various sizes and at a wide range of pressures and temperatures. Friction factors with cold water and saturated steam were consistent and reproducible; but great variations in friction factors were obtained with superheated steam. The cause for these variations is now being investigated and is apparently related to the condition of the feedwater. Head losses through valves will also be studied. In connection with this study use is made of an X-ray tube as a manometer and with unusually fine results.

Sampling Pulverized Fuel in Moving Gas Streams, J. C. Hardigg, Chairman. This project was organized at a meeting held during the 1932 Annual Meeting of the Society. The Committee is investigating the possibility of developing a satisfactory standard method for sampling pulverized fuels in a moving gas stream and for checking the efficiency of pulverizing equipment. Various methods now in use for sampling are being tested and compared with the hope that a generally satisfactory test method may be evolved.

Cotton Seed Processing. At its meeting on December 9, 1932, the Council approved the organization of this committee to study the mechanical problems in conditioning, cooking, and pressing cotton-seed meats. The project is centered at the University of Tennessee under Prof. W. R. Woolrich, and is moving forward slowly. A progress report was presented at the 1933 Semi-Annual Meeting in Chicago, the substance of which was published in the trade press. The Tri-States Cotton Oil Mill Superintendents Association has subscribed $500 and the University of Tennessee $675 toward the experimental program. A grant of $500 during 1933 was secured by the Society from The Engineering Foundation to support the study.

The following two Committees were inactive due to business conditions: the Special Research Committee on Worm Gears and the Joint S.A.E.-A.S.M.E. Committee on Diesel Fuel Combustion.

Respectfully submitted,

W. H. Fulweiler, *Chairman*
A. D. Bailey — D. B. Bullard
G. M. Eaton — C. R. Richards.

STANDARDIZATION

Notwithstanding the economic conditions which have prevailed during the year, satisfactory progress has been made by many of the committees, and the committee organization has grown somewhat. In all, 31 large sectional committees have been at work on as many separate projects under the procedure of the American Standards Association.* These include one new project, the sponsorship for which was accepted in May, 1933. These committees have subdivided themselves into 265 subcommittees and subgroups in order that the work assigned to them might be carried forward more promptly and effectively. The total number of members of these committees is now 1145, of which 412 are members of the A.S.M.E. During the year the Society was represented on 28 additional standards committees by 41 of its members. A total of 50 standards committee meetings were held within the twelve month period.

The broad nature of the Society's technical committee activity in the field of standardization is indicated by the nature of the several projects which have made the most significant progress toward completion during the year. The greatest interest has centered around the revision and extension of two American Tentative Standards originally completed and published in 1927, the standard for "Steel Flanged Fittings and Companion Flanges" and the standard for "Wrench Head Bolts and Nuts and Wrench Openings." On the other hand, the most general interest of the membership at large was shown in the two standards for "Engineering and Scientific Charts for Lantern Slides" and "Abbreviations for Scientific and Engineering Terms." The first of these projects grew out of recommendations of the A.S.M.E. Meetings Committee for the improvement of the form of lantern slides used in the Society's meetings to illustrate technical papers and committee reports. The chairman of the Committee was W. A. Shewhart. The second is part of a larger project set up for the standardization of letter symbols for use in technical publications of all kinds and was led by George A. Stetson, editor of A.S.M.E. publications. All four of these standards are now available in pamphlet form.

The list of standards completed includes also a number which are of special interest to certain branches of mechanical engineering. They are "Gear Materials and Blanks," "Shaft Couplings," "Spur Gear Tooth Form," and standards on several types of ball bearings.

Two projects which, though not completed, have been advanced well along toward completion by the energetic work of their committees are: (1) Code for Pressure Piping, E. B. Ricketts, chairman, and (2) Drawings and Drafting Room Practice, F. DeR. Furman, chairman of Sectional Committee, who was assisted in the editing of the standard by T. E. French.

The new project for which the Society was named sponsor, the Unification of Rules for the Dimensioning of Furnaces for Burning Solid Fuel, grew out of the necessity of burning fuels with higher efficiency and without the production of smoke. It was proposed by the Pure Air Committee.

Under the American Standards Association's procedure it is con-

* The A.S.M.E. has two representatives on the ASA. For their report see page 45.

venient to indicate the present status of the several projects for which the Society holds sponsorship or joint sponsorship by classifying them under the following seven heads which represent definite steps in that procedure: (1) Committee Organization, (2) Development of Proposal by Subcommittee, (3) Distribution of Tentative Proposal to Industry for Criticism and Comment, (4) Review of Comments by Subcommittee, (5) Revised Proposal Before Sectional Committee, (6) Proposed Standard Before Sponsor Bodies, and (7) Approval by ASA and Publication in Pamphlet Form.

This care and deliberation in the development of these standards has been found desirable and necessary since the designation "American Standard" implies general acceptance by all interested groups forming the branch of industry affected by the standard.

(1) Committee Organization

Classification and Designation of Surface Qualities, C. G. Mettler, Temporary Chairman. The organization meeting of the Sectional Committee on Classification and Designation of Surface Qualities was held in New York on December 9, 1932. Officers were elected for the Committee and five subcommittees organized to prepare draft standards covering: (1) Surfaces Produced by Tools and Abrasives, J. Cetrule, temporary chairman, (2) Surfaces Produced by Mold, Die, Rolls or Any Other Means of Deforming Materials, E. F. Cone, temporary chairman, (3) Coated Surfaces, V. W. Todd, temporary chairman, (4) Symbols for Indicating Surface Quality on Drawings, T. G. Crawford, chairman, and (5) Ways, Means, and Apparatus for Measuring Quality of Surface, R. T. Kent, temporary chairman.

Subcommittees Nos. 4 and 5 held their first meetings in Chicago during the Semi-Annual Meeting of the Society, June 26–30, 1933. A tentative proposal on Symbols for Indicating Surface Quality on Drawings is now in the hands of the members of Subcommittee No. 4 for review.

Unification of Rules for the Dimensioning of Furnaces for Burning Solid Fuel, C. W. Bronson, Temporary Chairman. Acting on a resolution passed at a conference called by the A.S.M.E. Pure Air Committee in December, 1932, the A.S.M.E. Standardization Committee recommended to the Council of the Society in March, 1933, that a project on standard setting heights of small boilers be initiated under the procedure of the American Standards Association. Favorable action was taken by the Council of the A.S.M.E. and the Mechanical Standards Advisory Council, and on March 29, 1933, the A.S.M.E. requested the ASA to authorize the organization of such a project and offered its services as sponsor or joint sponsor.

The ASA in turn approved the initiation of the project and assigned sponsorship to the A.S.M.E. in May. Letters inviting 21 national associations to appoint official representatives to the Committee were immediately issued, and the organization meeting of the Committee was held on June 26, 1933, in Chicago. At this time the temporary officers of the Committee were elected and five subcommittees appointed: No. 1 on Scope, No. 2 on Combustion and Design, No. 3 on Warm Air Furnaces, No. 4 on Steel Heating Boilers (internally and externally fired), and No. 5 on Cast Iron Boilers.

(2) Development of Proposal by Subcommittee

Ammonia Flanges and Fittings, W. R. Kremer, Chairman. The proposed American Standard for Ammonia Fittings was formally approved by Subcommittee No. 1 on Cast Iron Flanges and Flanged Fittings for release to industry for criticism and comment. It is now being set in type to provide the necessary copies for this distribution.

Center to Face Dimensions of Ferrous Flanged Valves, F. H. Morehead, Chairman. During this year Subcommittee No. 5 on Face to Face Dimensions of Ferrous Flanged Valves developed a proposed standard for Center to Face Dimensions of Ferrous Flanged Valves for various pressures. This proposal will be distributed to industry in the near future for criticism and comment.

Materials and Stresses, A. M. Houser, Chairman. A series of adjusted pressure ratings for steel fittings when used at temperatures other than the rated temperature of 750 F was developed by Subcommittee No. 4 on Materials and Stresses. Final approval and publication of this table is delayed pending the modification of certain of the ratings to permit the use of the table in oil refinery practice.

Port Openings. The work on the revision of Port Openings for the 125 lb cast iron flanged fittings, sizes 14 to 24 in., which was assigned to Subcommittee No. 9 with W. W. Hubbard as chairman, has been discontinued. This action was taken upon the recommendation of the chairman, who reported after meetings of his Committee and a canvass of the industry that there was no pressing need for the changes proposed.

The four preceding projects are being developed by subcommittees of the Sectional Committee on Pipe Flanges and Fittings, of which C. P. Bliss is chairman.

Taper Pipe Threads, S. B. Terry, Chairman. During this year Subcommittee No. 2 on Taper Pipe Threads completed the revision of pamphlet B2-1919, which outlines the Taper Pipe Thread Standard with the exception of one or two points concerning gages which are still in question. It is expected that these will be settled shortly, and that the revision will then be submitted to the Sectional Committee for approval. This project comes within the scope of the Sectional Committee on Standardization of Pipe Threads.

Screw Threads, R. E. Flanders, Chairman. The Sectional Committee met in January, 1933, and at that time all details incidental to the revision of the Screw Thread Standard (B1a-1924) were thoroughly reviewed. This work is now under way, and the principal departures from the original standard are: (1) addition of three new series, 8, 12, and 16 pitch, (2) fixing of pitches for sizes 3 to 4 in., inclusive, and (3) revision of minor diameters of nuts. The combined work of Subcommittee No. 2, C. W. Bettcher, chairman, and Subcommittee No. 3, R. E. Flanders, acting chairman, is represented in the revision.

Screw Thread Survey, R. E. Flanders, Chairman. The report of the survey written by Earle Buckingham was discussed at the January, 1933, Sectional Committee meeting, and two copies were subsequently circulated to the members of the Committee for review prior to final editing and publication.

Acme Threads, Earle Buckingham, Chairman. Subcommittee No. 4 on Acme Threads has prepared a proposal covering the several types of Acme threads in more common industrial use. It is anticipated that this proposal will soon be duplicated for transmittal to the members of the Subcommittee and to industry in general for criticism and comment.

Lag Screw Threads, George Case, Chairman. On recommendation of its chairman, Subcommittee No. 6 on Lag Screws was discharged at the January, 1933, Sectional Committee meeting.

The four preceding items are parts of the work for which the Sectional Committee on Standardization and Unification of Screw Threads, R. E. Flanders, chairman, is responsible.

Gage Sizes, H. B. Reynolds, Chairman. In the preparation of its report Subcommittee No. 3 on Gage Sizes and Mounting Dimensions, issued a request in April, 1933, to a number of manufacturers of gages of the bourdon type for information concerning their present practice. A considerable amount of data which has been received by the Committee will be used as a basis for drafting its tentative report.

Accuracy and Test Methods for Pressure Gages, O. J. Hodge, Chairman. This Subcommittee completed a draft of its proposed report in February, 1933. However, since certain information seemed to be necessary to complete this section of the proposed standard on pressure and vacuum gages, a questionnaire was sent to a number of manufacturers in June, 1933, together with a copy of the tentative report. Replies are still being received to this inquiry.

Plain and Lock Washers, C. W. Squier, Chairman. Subcommittee No. 1 on Plain Washers developed a tentative standard for plain punched steel and cast washers with round holes, copies of which were distributed to the members of the Subcommittee in October, 1932. The replies to this distribution indicated that while the proposed sizes and proportions were agreeable to practically all of the large users, they were not entirely acceptable to the majority of manufacturers. On the basis of the replies received, a further study is being made in an effort to establish a logical compromise between the two principal points of view represented on the Committee.

Cast Iron Soil Pipe and Fittings, J. J. Crotty, Chairman. Splendid progress has been made during the year in the development of the proposed American Standard for Cast Iron Soil Pipe and Fittings. The Subcommittee responsible for this work held three meetings during the year, one of which was a three-day meeting, and completed a draft of the proposed standard, numbering some 45 pages, which has been released recently by the Subcommittee for distribution to industry for criticism and comment. Chairman Crotty presented the proposed standard for information and discussion at the annual meeting of the American Society of Sanitary Engineering, co-sponsor for this project, which was held in Chicago during the week of August 5.

Traps, A. R. McGonegal, Chairman. A tentative report on plumbing traps was received in August from Subcommittee No. 5 on Traps. It will now be duplicated for general distribution to industry for criticism and comment.

These two projects, together with a number of others in the same field, constitute the work undertaken by the Sectional Committee on the Standardization of Plumbing Equipment, of which W. C. Groeniger is chairman.

Machine Tapers, F. S. Blackall, Jr., Chairman. The Technical Committee met on December 6, 1932, and at this time the Steep

Taper proposal was referred back to Mr. Bouvier's Subgroup for further investigation of the patent situation and additional tests.

The Slow Taper proposal also was referred back to A. H. Lyon's Subgroup for minor changes with the recommendation that subsequently it be set up in draft form for final review by Technical Committee No. 3 and transmittal to the Sectional Committee for letter ballot vote. The revised draft has now been submitted by the Subgroup and is ready for duplication and distribution to the Technical Committee for review.

Other Projects in Course of Development. While no specific progress has been reported for the following projects during the past year they are being actively developed by the several committees concerned:

Machine Pins
Graphic Presentation
Malleable Iron Steel and Brass Seat Unions
Allowances and Tolerances for Cylindrical Parts and Limit Gages (Revision of standard approved and published in 1925)
Small Tools: Circular Forming Tools and Holders; Splines and Splined Shafts; Electric Welding Dies and Electrode Holders; Milling Machine Tables; Multiple Spindle Drilling Heads; Nomenclature
Revision of 125 Lb and 250 Lb Cast Iron Screwed Fittings
Leather Belting
Foundry Equipment
Stock Sizes, Shapes, and Lengths for Hot and Cold Finished Iron and Steel Bars
Wood Screw Threads
Inspection of Gears

(3) Distribution of Tentative Proposals to Industry for Criticism and Comment

Speeds of Machinery, A. E. Hall, Chairman. The Sectional Committee on Standardization of Speeds of Machinery released the proposed American Standard for Machine Speeds for distribution to industry for criticism and comment, and this proposal was mailed to 376 firms and individuals in April, 1933. The replies received are now being sent to the Committee for its consideration and further action. This project gained considerable publicity through an article in the April, 1933, issue of *Mechanical Engineering* and an editorial in the June, 1933, issue of *Power Engineering*.

Welding Neck Flanges, C. H. Haupt, Chairman. A special Subgroup of Subcommittee No. 3 on Steel Flanges and Flanged Fittings has been responsible for the development of a proposed American Standard for Welding Neck Flanges. This proposal was sent to industry in May, 1933, and the replies received from this distribution are now being correlated for transmittal to the subgroup for its review. This project is one of the many activities undertaken by the Sectional Committee on Pipe Flanges and Fittings, of which C. P. Bliss is chairman.

(4) Review of Comments by Subcommittee

Drawings and Drafting Room Practice, F. DeR. Furman, Chairman. Upon revision of the several sections of the proposed American Standard for Drawings and Drafting Room Practice by their respective subcommittees after review by industry, they were turned over to the Editing Committee, of which T. E. French is chairman. Professor French completed the editorial work during the past winter, and after preliminary review by the members of a special subcommittee of the Sectional Committee, this proposed American Standard will be submitted to the entire committee membership for vote on approval by letter ballot.

Code for Pressure Piping, E. B. Ricketts, Chairman. The sections of the Code for Pressure Piping covering (*a*) Power Piping, (*b*) Gas and Air Piping Systems, (*c*) Oil Piping, (*d*) District Heating Piping, and (*e*) Fabrication Details, have been completed by their respective subcommittees, edited, and released by the Editing Committee. Under the direction of Chairman Ricketts these sections are now being combined and unified and an extensive index is being developed. This work, however, is practically completed, and in the near future copies of the entire code will be prepared in quantity for distribution to the membership of the Sectional Committee for consideration prior to a meeting to be held in December at which final action will be taken.

Brass Fittings for Flared Copper Tubes, F. L. Riggin, Chairman. This Subcommittee of the Sectional Committee on Plumbing Equipment, of which W. C. Groeniger is chairman, completed a revision of its proposed standard on Brass Fittings for Flared Copper Tubes based on the comments received from the distribution of copies of the standard to industry in August, 1932. The proposed standard is now ready for discussion and approval by the Sectional Committee.

Wrought Iron and Wrought Steel Pipe and Tubing, H. H. Morgan, Chairman. During the present year Subcommittees Nos. 2 and 3 on Pipe and Tubing for Low and High Temperature Service, respectively, have been cooperating in the preparation of the proposed standard for Wrought Iron and Wrought Steel Pipe. In February, 1933, proof copies of the proposed standard were sent by Chairman J. J. Shuman of Subcommittee No. 2 to industry for criticism and comment. On March 8, 1933, the two subcommittees held a joint meeting in New York at which the 120 replies received were discussed and recommendations were made for the necessary revisions. At this same meeting Chairman Tanner of Subcommittee No. 3 presented a tentative proposal covering the specific application of this proposal to the 150, 300, 400, 600, 900, and 1500 lb steel flange standards.

In June, 1933, the Sectional Committee met in Chicago and at this meeting revised page proofs of the proposed standard for Wrought Iron and Wrought Steel Pipe and also a revised draft of the table showing wall thicknesses of wrought steel pipe for working pressures of 150, 250, 300, 400, 600, 900, and 1500 lb at 750 F were discussed in detail and were referred back to their respective subcommittees for certain additional refinements.

Socket Head Cap and Set Screws, Herman Koester, Chairman. This Subcommittee submitted the proposed American Standard for Socket Type Set and Cap Screws to industry for criticism and comment in March, 1933. The suggestions received as a result of this distribution are now in the hands of the chairman of the Committee and it is anticipated that the final draft will be ready for submittal to the Sectional Committee on the Standardization of Bolt, Nut, and Rivet Proportions in October.

Large Rivets, J. E. Kiernan, Chairman. The proposed American Standard for Large Rivets is still in the hands of Subcommittee No. 1 of the Sectional Committee on Bolt, Nut, and Rivet Proportions. This proposal was completed last fall with the exception of some further consideration by the manufacturers of the high-button-head shape of rivet head and the addition of dimensions and tolerances for rivet sets. Since that time the data on the sets have been accumulated and the manufacturers have conducted considerable research on the modification of their manufacturing processes or the design necessary in order to make the production of the high-button-head type of rivet commercially feasible. With these two remaining points apparently on the way toward completion, the submission of this standard for approval by the Sectional Committee is expected in the near future.

Lock Washers, C. H. Loutrel, Chairman. The work of Subcommittee No. 2 on Lock Washers has not progressed appreciably during the current year, principally on account of the present economic conditions. At the Committee's last meeting, which was held in April, 1932, substantial agreement was reached in the Committee on all the differences then existing as a result of replies received from industry. It is accordingly expected that some progress toward completion of the work will be made at an early date.

Pressure and Vacuum Gages, M. D. Engle, Chairman. Subcommittee No. 2 on Definitions, C. F. Schwep, chairman, held a meeting on December 6, 1932, at which revisions to its preliminary proposal covering Definitions for Pressure and Vacuum Gages, Dial Type, were completed. Copies of this proposal were distributed broadly for criticism and comment to industry in March, 1933. The replies received are now being reviewed by the Subcommittee for the purpose of making final revisions, after which the report will be submitted to the Sectional Committee for approval.

Spindle Noses and Collets, J. E. Lovely, Chairman. In October, 1932, Subgroup No. 4 of Technical Committee No. 4 on Spindle Noses met and recommended that the March, 1931, draft of the Spindle Nose proposal be adopted by the single and multiple spindle automatic lathe groups and the turret lathe group of the N.M.T.B.A. This recommendation has since been adopted by these groups, but no dates for the application of the standard designs to stock equipment have been set.

Jig Bushings, C. E. Rundorff, Chairman. In November, 1932, a revised draft of the proposed standard for Jig Bushings was distributed to industry for criticism and comment. The 145 replies received were abstracted and sent in February to all members of this Technical Committee for their review. On the basis of the comments received as a result of this distribution, a further revised draft is now being prepared for final approval by the Technical Committee.

Chucks and Chuck Jaws, J. E. Lovely, Chairman. During this year little progress was made in this project due to the economic conditions. It is expected that the work will develop rapidly, however, just as soon as conditions improve.

Punch and Die Sets, Sidney Diamant, Chairman. The Technical Committee held a meeting in December, 1932, and recommended that the proposed standard for Punch and Die Sets be revised in accor-

dance with the recommendations of a special subgroup of which W. C. Mueller was Chairman. These recommendations were based on the replies received as a result of the distribution of the proposal to industry in June, 1932.

On June 5, 1933, a meeting of Subgroup No. 3 on Details of Design and Editing was held and further progress was made toward the preparation of the proposed standard to comply with the criticisms received from industry.

The Technical Committees handling the preceding four projects are responsible to the Sectional Committee on the Standardization of Small Tools and Machine Tool Elements, C. W. Spicer, chairman.

(5) Revised Proposal Before Sectional Committee

Screw Threads for Hose Couplings, H. W. Bearce, Chairman. In October, 1932, the Sectional Committee on Standardization of Screw Threads for Hose Couplings met in New York and agreed upon a rearrangement of Table 1, the principal table of the proposed standard for Screw Threads for Hose Couplings. The Committee further authorized the addition of two tables giving complete detail dimensions for the thread form. In December, 1932, a revised draft of the standard was submitted to the members of the Sectional Committee for vote by letter ballot, but the returns from this ballot indicated considerable disagreement over the form of Table 1. In January, 1933, A. O. Boniface sent out a questionnaire to certain members of the committee in an effort to obtain a consensus of opinion on certain definite points in question. This was followed in April, 1933, by the distribution of a further revised draft of Table 1 in an effort to crystallize the ideas which had been put forth as a result of the first distribution of the standard and Mr. Boniface's questionnaire.

A study of all of the replies received in reply to these three mailings and a survey of the views of several manufacturers and dealers in New York was made by the A.S.M.E. staff and another draft of Table 1 was prepared in June and discussed with several members of the Committee in New York, Chicago, and Washington. This draft with some slight alterations was approved by Chairman Bearce in August for submittal to the members of the Sectional Committee for their formal consideration and vote by letter ballot.

Shafting. C. M. Chapman, Chairman. The revision of the standards for stock keys was completed by the Sectional Committee and submitted in October, 1932, to its membership for vote by letter ballot. The result of this vote was 15 for and 1 against. The reason given for this negative vote was that the branch of industry casting this vote does not approve the addition of the $^{5}/_{16}$ in. key now proposed by the Committee. After considerable correspondence between Chairman Chapman and the representative of that industry, a compromise was agreed upon whereby both the $^{1}/_{4}$ and $^{5}/_{16}$ in. sizes would be shown for the $1^{1}/_{4}$ in. shaft. This compromise proposal was submitted to the members of the Sectional Committee in August of this year and was approved by a vote of 14 to 2. It is altogether likely, however, that a unanimous agreement will be reached within the very near future, at which time the proposed revision will be ready for submittal to the Society as sponsor and to the American Standards Association for approval and designation as an American Standard.

Twist Drills. At the December 8, 1932, meeting of the Sectional Committee on Small Tools and Machine Tool Elements the proposed standard for Drill Sizes was discussed at length. As a result of this discussion, at least four members of the Committee each conducted a separate and independent thorough study of the entire situation, in his own plant. As a result of this further study, suggestions were made as to certain minor changes which would further simplify the list of Drill Sizes under discussion. This list, as amended, was adopted by mail ballot by the Technical Committee, and later by the Sectional Committee on Small Tools and Machine Tool Elements, and is now ready for approval of the sponsors.

(6) Proposed Standard Before Sponsor Bodies

Ball and Roller Bearings, F. W. Gurney, Chairman. In January, 1933, the Sectional Committee completed an extension of the tables for the light, medium, and heavy series of angular contact ball bearings contained in the American Standard for Ball and Roller Bearings approved and published in 1930.

A new table covering adapter sleeve bearings was also developed and approved by the Sectional Committee. These additions were submitted to the sponsor bodies in January, 1933, by the Committee. The S.A.E. approved the new sections at its meeting held during that month and the A.S.M.E. gave its approval on May 22, 1933.

A proposed standard for Taper Roller Bearings was presented by the Sectional Committee in August, 1933, to the two sponsor bodies for approval and transmission to the American Standards Association for designation as an American Standard. Action is now being taken by the S.A.E. and the A.S.M.E.

(7) Approval by ASA and Publication in Pamphlet Form

The following eight standards were approved as American Standards by the ASA during the past twelve months. The first six of them were published by the Society within the year and it is interesting to note that already a total of 4300 copies of the six have been sold.

Shaft Couplings, developed by A.S.M.E. Committee on Shaft Couplings, D. J. McCormack, Chairman.

Spur Gear Tooth Form, developed by Subcommittee No. 4, H. J. Eberhardt, Chairman, of the Sectional Committee on the Standardization of Gears, B. F. Waterman, Chairman.

Steel Flanges and Flanged Fittings, developed by Subcommittee No. 3 of the Sectional Committee on Standardization of Pipe Flanges and Fittings, C. P. Bliss, Chairman.

Wrench Head Bolts and Nuts, developed by Subcommittee No. 2, H. A. Spanagel, Chairman of the Sectional Committee on Standardization of Bolt, Nut, and Rivet Proportions.

Engineering and Scientific Charts for Lantern Slides, developed by Subcommittee No. 4, W. A. Shewhart, Chairman, of the Sectional Committee on Standards for Graphic Presentation, E. F. DuBrul, Chairman.

Abbreviations for Scientific and Engineering Terms, developed by Subcommittee No. 9, G. A. Stetson, Chairman, of the Sectional Committee on Standardization of Scientific and Engineering Symbols and Abbreviations, J. F. Meyer, Chairman.

Gear Materials and Blanks, developed by Subcommittee No. 8, T. D. Lynch, Chairman, of the Sectional Committee on the Standardization of Gears, B. F. Waterman, Chairman.

Ball Bearings, developed by Sectional Committee on Standardization of Ball and Roller Bearings, F. W. Gurney, Chairman.

Respectfully submitted,

W. S. Monroe, *Chairman*
Earle Buckingham — Alfred Iddles
C. W. Spicer — L. A. Cornelius.

POWER TEST CODES

The Committee on Centrifugal and Turbo-Compressors and Blowers, A. T. Brown, Chairman, and the Committee on Instruments and Apparatus, C. F. Hirshfeld, Chairman, have held the center of the stage during the past year in the A.S.M.E. Power Test Codes activity.

The proposed standard Test Code for Centrifugal Compressors, Exhausters, and Fans was completed during the fall of 1932, so it was scheduled for a public hearing at the time of the Annual Meeting of the Society, December, 1932. The discussion at this hearing indicated that Part 1 on Compressors and Exhausters was generally satisfactory to industry in the form presented. The addition of a few minor paragraphs was proposed. This hearing, however, developed considerable disagreement over Part 2 on Fans, since the test procedure advocated therein differed somewhat from that required by the test code which was prepared in 1923 by a joint committee of the American Society of Heating and Ventilating Engineers and the National Association of Fan Manufacturers. At the close of the discussion a motion was passed recommending to the Power Test Codes Committee that the fan part of the proposed A.S.M.E. Code be not printed at this time and that a new committee be appointed consisting of members of the American Society of Heating and Ventilating Engineers, the National Association of Fan Manufacturers, and The American Society of Mechanical Engineers, to produce a test code for fans which will be generally acceptable.

However, the members of Committee No. 10 and the Main Committee were convinced that it would be a serious mistake to publish a fan code which would be entirely independent of the compressor code. Not only are the problems involved in the testing of the two classes of apparatus similar fundamentally, but often it will be difficult to distinguish between the two classes of apparatus. It was accordingly decided to make the fan code Part 2 of the combined code, rather than a completely independent publication. The Main Committee endorsed this plan of procedure and authorized Committee No. 10 to develop its test code in two parts, i.e., Part 1 on Compressors and Exhausters and Part 2 on Fans. It is understood that while these parts will be first published separately, they will later be made available as one pamphlet.

Prof. M. C. Stuart, of Lehigh University, who has been named chairman of the new Subcommittee on Fans, has presented a plan of procedure which is approved by the Main Committee and which calls for the setting up in the Packard Laboratory at Lehigh University of a neutral proving ground for the demonstration of apparatus and code procedures in the development of fan testing.

In November, 1932, the members of Committee No. 10 met with the members of the A.S.M.E. Special Research Committee on Fluid Meters, R. J. S. Pigott, chairman, and the P.T.C. Committee No. 19 on Instruments and Apparatus to consider certain objections which the Fluid Meters Committee had raised to the section of the Code dealing with the measurement of volume by the use of the flow nozzle and the formulas and constants to be employed in the corresponding computations. The methods endorsed by Committee No. 10 are taken from Instruments and Apparatus, Part 5 on Measurement of Quantity of Materials, proposed Section 4 on Flow Measurement With Orifices, Flow Nozzles, and Venturi Tubes. At this conference, it was agreed finally (1) that the material so far developed by Committee No. 10 was satisfactory for its purpose, (2) that the forms of the equations employed were not in conflict with the principles laid down by the Fluid Meters Committee, and (3) that, while the conference recognized the immediate need on the part of Committee No. 10 for the methods of measurement and formulas which it had developed, the A.S.M.E. Special Research Committee on Fluid Meters was encouraged to investigate the relative merits of the venturi, the diaphragm, the orifice, and the flow nozzle for the measurement of gas flow in square and round ducts. Under the direction of the Research Committee on Fluid Meters, a subcommittee reviewed the section on flow measurement for the purpose of suggesting changes and additions to make these rules more generally applicable. Its report is now in the hands of the Chairman of the Committee on Instruments and Apparatus, No. 19.

While these negotiations were in progress, the Main Committee received communications from the International Standards Association and the British Compressed Air Society urging the adoption of the proposal of the I.S.A. Committee, which in effect is the V.D.I. form of nozzle with slight modifications in the test procedure over that recommended by our sister society.

Committee No. 19 is now studying carefully the details of design of nozzle and rules for use prepared by (1) its subcommittee on Flow Measurement, (2) the special subcommittee of the A.S.M.E. Special Research Committee on Fluid Meters, and (3) the I.S.A. Committee on the Measurement of Fluid Flow.

Work completed by Committee No. 19 during the year just closed includes the following five sections which were approved for publication, namely, Part 3, Temperature Measurement—Chapter 8 on Optical Pyrometers; Part 4 on Head Measuring Apparatus; Part 6 on Electrical Measurements; and Part 21, Leakage Measurement—Chapter 2 on Boiler and Piping Leakage and Chapter 3 on Steam Engine Leakage.

A draft of the proposed revision of the Test Code for Reciprocating Steam Engines was reviewed by the members of Committee No. 5, A. G. Christie, chairman, and certain members of the Main Committee. Important omissions were pointed out which have since been supplied. The final draft of the proposed revision of the Test Code for Reciprocating Steam Engines is now in shape for submission to the Main Committee.

At a meeting which was held in New York in January, 1933, Committee No. 6 on Steam Turbines, C. H. Berry, chairman, discussed the preliminary draft of the first revision of the Test Code for Steam Turbines. Many changes in this draft were suggested, and several items were referred to the members present for detailed study and report. The proposed redraft of the Code differs from the existing code in several respects: (*a*) It covers bleeder and mixed-pressure turbines; (*b*) It permits the use of orifices, nozzles, etc., for the measurement of steam flow, with certain qualifications concerning expected accuracy; (*c*) It permits the determination of exhaust pressure by thermometer for condensing turbines, with certain precautions; (*d*) It permits the use of electric resistance thermometers and thermocouples for measuring temperatures; and (*e*) The methods of making corrections have been more clearly set forth and have been extended to cover bleeder and mixed-pressure turbines. The proposed draft also conforms to the standard form recently adopted for all of the A.S.M.E. Power Test Codes.

Committee No. 3 on Fuels, W. J. Wohlenberg, chairman, has been reviewing the preliminary draft of the Test Code for Gaseous Fuels. Certain constructive criticisms of this draft have been received from the American Chemical Society through the secretary of its Gas and Fuel Division and are under consideration. Standard methods for determining the heating value of gas, especially the gases with heating value of 1000 Btu per cu ft or more, will also be recognized in the proposed code.

Committee No. 12 on Condensers, Water-Heating and Cooling Equipment, Geo. A. Orrok, chairman, is revising the Test Code for Steam Condensing Apparatus. It has also made progress in the development of a suitable method for determining the cleanliness factor of condenser tubes which is now to be incorporated in this test code. The subject of the cleanliness factor was thoroughly and fully discussed at a meeting of the A.S.M.E. Metropolitan Section in November, 1932, when a paper was presented entitled "A Test Method for Determining the Quantitative Effect of Tube Fouling on Condenser Performance," by P. H. Hardie and W. S. Cooper, of the Brooklyn Edison Company. Several methods of measuring the condenser cleanliness factor were described and data on several tests were presented showing the practicability of obtaining a specific value for the degree of tube fouling existing at the time of test.

The following committees held regularly called meetings during the past twelve months: Main Committee, four meetings—October, December, May, and June; Committee No. 6 on Steam Turbines; Committee No. 10 on Centrifugal and Turbo-Compressors and Blowers; Committee No. 11 on Complete Steam Power Plants; and Committee No. 18 on Hydraulic Power Plants.

A résumé of the progress which has been made during the past year in the preparation of the various parts and chapters of Instruments and Apparatus is as follows: The completion of Part 3, Temperature Measurement—Chapter 8 on Optical Pyrometers was reported in the November, 1932, issue of *Mechanical Engineering;* Part 10 on Flue and Exhaust Gas Analyses in the November, 1932, issue; and Part 14 on Linear Measurements in the July, 1933, issue. The development of the following parts and chapters has progressed appreciably within the past year: Part 2, Pressure Measurement—Chapter 2 on Static and Total Pressure, Static Holes and Tubes, Impact Tubes, and Chapter 3 on Pipes for Pressure Measurement; Part 3, Temperature Measurement—Chapter 2 on Radiation Pyrometers; Part 10 on Flue and Exhaust Gas Analyses; Part 15 on Measurement of Surface Areas; and Part 20 on Smoke-Density Determinations.

The question of a change of date of the meetings of the Main Committee on Power Test Codes was discussed at some length during the year. It was decided, however, that there shall be a December meeting during Annual Meeting week, since this annual meeting, to which all members of the Individual Committees are invited, offers an excellent opportunity for an interchange of ideas and information relative to the advancement of the work. A questionnaire was prepared and sent out to the members of the Main Committee and the chairmen of Individual Committees to ascertain their views on this question of meeting dates. A tabulation of the replies received indicates that the majority of members favor the holding of three meetings a year on the first Monday of the months of October, December, and April.

International Cooperation. Through membership in the U. S. National Committee of the International Electrotechnical Commission,* the Society, and for it the Main Committee on Power Test Codes, has been actively engaged in assisting to bring about international agreements on acceptance tests for steam turbines and internal-combustion engines. The Secretariat for the Advisory Committees on Steam Turbines, Internal-Combustion Engines, and Hydraulic Turbines is held by the United States National Committee of the I.E.C. The Society was represented this year on this Committee by Harvey N. Davis, Francis Hodgkinson, and Ely C. Hutchinson, with C. Harold Berry and Paul Diserens serving as alternates.

I.E.C. Internal-Combustion Engines. During the past year preliminary negotiations were completed which resulted in the scheduling of a meeting in the United States in September of Advisory Committee No. 19 on the Testing of Internal-Combustion Engines. However, due to the unsettled economic conditions throughout the world the cancelation of this meeting was deemed advisable.

The first meeting of this international committee, which was held in Stockholm in July, 1930, was attended by official delegates from Czechoslovakia, France, Germany, Great Britain, Italy, Sweden, and the United States of America. After reviewing the documents which had been presented by the several national committees, the Committee devoted its efforts to the preparation of a standard specification and information form which, when completed, will be available for use when international tenders or estimates are desired. Good progress had been made toward the completion of this document, Part I, the Specification, of the I.E.C. Publication on Internal-Combustion Engines, by correspondence since that meeting, and it was hoped that definite agreements might have been reached concerning it had a meeting been held this year.

The Secretariat had prepared also a rewritten document for Part II, Rules for Acceptance Tests, based on the comments and opinions of the several National Committees, which was to have been discussed at the September meeting. At Stockholm the U. S. National Committee had presented a proposal for these rules. The shortness of time, however, did not permit its detailed study at that time.

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* The A.S.M.E. has three representatives on the U. S. National Committee of the International Electrotechnical Commission. For their report see page 47.

That proposal was based primarily on the A.S.M.E. Test Code for Internal-Combustion Engines.

Respectfully submitted,

F. R. Low, *Chairman*
C. Harold Berry
A. T. Brown
E. H. Brown
A. G. Christie
Harte Cooke
Paul Diserens
R. H. Fernald
E. R. Fish
C. F. Hirshfeld
Francis Hodgkinson
O. P. Hood
G. A. Horne
E. C. Hutchinson
D. S. Jacobus
L. S. Marks
L. F. Moody
I. E. Moultrop
H. B. Oatley
G. A. Orrok
R. J. S. Pigott
E. B. Ricketts
E. N. Trump
Wm. Monroe White
W. J. Wohlenberg.

SAFETY

Consistently throughout the year in its dealings with the working committees of the Society, the Safety Committee has maintained that the prevention of industrial accidents is essentially a problem of management and is, therefore, of direct interest to the members of the A.S.M.E. Our Society is frequently referred to as the "Society of the Industries" and, since it includes within its membership men in all positions of authority from the senior executive down to the junior engineer by whom accident prevention measures are and must be administered, the importance of this activity is obvious.

The Committee holds that industrial, or, more properly, occupational accidents, can be prevented. The evidence in support of that statement is so ample that its truth cannot be doubted by any one familiar with the facts. Accidents do not happen. They are caused and their causes are removable. It is estimated that not more than 2 per cent of industrial accidents can properly be considered as not preventable. Out of the remaining 98 per cent of preventable accidents, perhaps 10 per cent can be charged to physical hazards or causes, whereas fully 88 per cent are chargeable to supervisory causes. Both of these causes are within the sphere of management. It is estimated that 15,000 persons were killed by occupational accidents in the United States in 1932, and that in addition there were 1,200,000 non-fatal injuries, of which about 45,000 resulted in permanent disability and 1,155,000 in temporary disability.

A new feature was introduced into the 1932 Annual Meeting at the suggestion and through the enterprise of the Safety Committee. Under its auspices a lecture entitled "Conservation an Essential in Industrial Recovery" was given by H. W. Heinrich, assistant superintendent, Engineering and Inspection Division, Travelers Insurance Company, on the afternoon of Wednesday, December 7, of the 1932 Annual Meeting Week.

While the past year has not recorded any unusual progress on the part of the five sectional committees handling safety code projects for which the A.S.M.E. is sponsor or joint sponsor, definite advances have been made by some. These advances and the present status of the committees are covered in the paragraphs which follow.

Aircraft Safety. The special committee on aircraft safety and inspection of the A.S.M.E. Aeronautic Division has been at work during the year in preparing a revision of the tentative draft of the proposed "Inspection Manual for Maintenance of Conventional Aircraft" originally distributed for criticism and comment in May, 1932. At the 1933 Semi-Annual Meeting of the Society held in Chicago, Jerome Lederer, chairman of the Committee, presented a paper which outlined the present need for the Manual and the nature of its contents in general.

In January, 1933, acting on the recommendation of the A.S.M.E. Safety Committee and the A.S.M.E. Aeronautic Division, the Society requested the American Standards Association that steps be taken to bring about the revision of the present Aeronautic Safety Code originally approved and published in 1925. This recommendation was made in the belief that progress and safety in aeronautics are now being sacrificed because of a lack of unity in the present regulations for inspection, operation, and maintenance. The recommendation of the Society was placed before the Society of Automotive Engineers, sponsor for this project, by the ASA.

Safety Code for Elevators, Sullivan W. Jones, Chairman. The Sectional Committee on a Safety Code for Elevators held a meeting in March, 1933. At this meeting a tentative draft of the Elevator Inspectors' Handbook, prepared by Subcommittee No. 2, K. H. Colahan, chairman, was reviewed and discussed in detail. The resulting suggestions for changes and additions were of such a nature that the Committee considered it necessary to refer this draft back to the Subcommittee for revision.

Consideration was given also at the meeting to certain changes proposed by the Subcommittee on Elevator Research, Interpretations, and Recommendations in the American Standard "Safety Code for Elevators" published in July, 1931. This latter Subcommittee, under the Chairmanship of M. H. Christopherson, held four meetings during the year, in November, 1932, January, 1933, March, 1933, and September, 1933. Under the direction of the Committee tests on several types of elevator buffers have continued to be carried on at the National Bureau of Standards for several manufacturers of elevators.

Safety Code for Cranes, Derricks, and Hoists, J. C. Wheat, Chairman. The last section of the proposed Safety Code for Cranes, Derricks, and Hoists was completed during the year by Subcommittee No. 4 on Miscellaneous Equipment for Cranes and Hoists (including Slings, Chains and Hooks, Wire Rope and Attachments, Sheaves and Pulleys), L. W. Hopkins, chairman, and distributed to the Subcommittee members in February, 1933.

Safety Code for Machinery for Compressing Air, H. D. Edwards, Chairman. As no progress had been reported by the Committee, the continuance of the preparation of this code was discussed at the October 17, 1932, meeting of the Standing Committee on Safety. W. M. Graff expressed the opinion that the work should certainly not be dropped, but suggested that possibly with a revision of the Committee's scope progress could be made. The Standing Committee approved his recommendation and a revised scope will presently be submitted to the ASA.

Society Representation on Safety Code Committees. During the year the Committee has prepared recommendations for the presidential appointments of representatives of the Society on the three following safety committees: Safety Code for Specifications and Methods of Test for Safety Glass; Safety Code for Exhaust Systems; and Safety Code for Work Under Compressed Air. This brings the total of the safety code committees on which the Society has official representation to 32.

Respectfully submitted,

H. W. Mowery, *Chairman*
T. A. Walsh, Jr.
M. H. Christopherson
W. M. Graff
H. H. Judson.

PROFESSIONAL CONDUCT

The Committee is pleased to report that few cases of unprofessional conduct on the part of members of the Society have come up for consideration this year.

Last year the Committee redrafted the Code of Ethics of the Society to meet certain criticisms. The new Code is now before the Council for final approval.

During recent months, there has been tremendous activity in the formulation of trade and industrial codes to control our industrial reconstruction. The time seems particularly opportune for a reconsideration of a joint Code of Ethics for all of the national engineering societies. Such a joint code was under consideration about twelve years ago, but was not adopted by all the national societies. The Committee recommends to the Council that the president of the Society be authorized to suggest to the other national societies that the preparation of a joint code is timely and desirable in order to standardize the rules governing the conduct of all engineers.

Respectfully submitted,

A. G. Christie, *Chairman*
H. S. Philbrick
C. G. Spencer
E. R. Fish
J. H. Herron.

BOILER CODE

During 1932–1933 the Committee held nine regular meetings and several Executive Committee meetings, which were devoted to interpretations of the various sections of the Code and to the formulation of revisions and addenda thereto.

At the request of the U. S. Navy Department the Committee reviewed the U. S. Navy Specifications for Machinery in so far as the proposed new rules for welding were concerned. A conference was held at Annapolis and in Washington with representatives of the Bureau of Engineering of the U. S. Navy Department, the American Welding Society, and the Boiler Code Committee to discuss tentative changes and additions for inclusion in the specifications. The representatives of the Bureau of Engineering were highly appreciative of the cooperation. In response to an invitation extended by the Boiler

Code Committee the Officer-in-Charge of the Specifications Section of the Design Division of the Bureau of Engineering was appointed by the Bureau as a representative to attend the Committee meetings. Commander C. S. Gillette, who took an active part in the conference, is the present Officer-in-Charge. This appointment places the Boiler Code Committee in touch with the experience and practice of the Navy Department. It was arranged that the Commandant of the Navy Yard, New York, should designate an officer of the rank of Commander or of higher rank to attend the meetings of the Committee when it is impossible for the Bureau's representative to be present and report to the Bureau.

The Committee reports the passage in Congress of a Bill to amend sections of the Revised Statutes, as amended, relating to the construction and inspection of boilers, unfired pressure vessels, and appurtenances thereof, which provides for the revision of the present Steamboat Inspection Rules by empowering the Board of Supervising Inspectors to prescribe formulas, rules, and regulations. This Board has cooperated actively with the Boiler Code Committee for many years and has designated a representative to attend all of its meetings.

The Committee was requested by the National Board of Boiler and Pressure Vessel Inspectors to review a set of rules for repairs by fusion welding of boilers and other pressure vessels. These rules were considered at several regular meetings of the Boiler Code Committee and also by the Executive Committee, and a number of revisions were proposed which were approved by the National Board. The National Board was advised that the rules as amended are consistent with the rules in the Boiler Code for welded construction of new boilers and should give safe results.

The X-ray requirements for power boilers and unfired pressure vessels were revised as a result of experience and the development of new X-ray technique. One of the important changes was increasing the maximum wall thickness for X-raying power boilers and unfired pressure vessels to 4¼ in.

Under conditions where tubes were subjected to a high temperature the limiting creep stress became an important factor. As a result, it was necessary to provide for a weldable material with a higher creep stress value, and Specifications S-17 for Lap-Welded and Seamless Steel and Lap-Welded Iron Boiler Tubes were revised to conform with A.S.T.M. Specifications A83-32T.

The Committee is giving consideration to the use of alloy steels for both riveted and welded construction of unfired pressure vessels which are not as yet provided for in the Material Specifications Section of the Code. It is also giving consideration to carbon steels of higher tensile strength suitable for fusion welding.

Revisions were made in Specifications S-1, S-5, S-6, S-7, and S-18 to indicate that steel made by the electric furnace process will be acceptable under the Code on an equal basis with open-hearth steel.

As stated in the Annual Report for the fiscal year ending 1932, revised rules for the reinforcement of openings were incorporated in the Code. As a result of many questions in regard thereto, and to help in understanding and applying the rules, the Committee prepared and published in *Mechanical Engineering* a statement showing their derivation, which is now available in reprint form.

The Committee has made numerous revisions and additions to the various sections of its Code as a result of suggestions from manufacturers, inspectors, users, and others. These revisions were formally adopted by the Council on July 26, 1933, and distributed in pink-colored addenda sheets. 1933 Editions of the Power Boiler Code and the suggested Rules for Care of Power Boilers in Service were published in which were incorporated all revisions issued to date.

A new Subcommittee of the Boiler Code Committee on Special Design was appointed, consisting of D. B. Wesstrom, chairman, W. L. Bowler, R. E. Cecil, Paul Diserens, H. E. Rockefeller, W. H. Rowand, and F. S. G. Williams.

At the request of the Boiler Code Committee, the Council appointed F. W. Dean and Prof. E. F. Miller honorary members of the Committee. It is with deep regret that the Committee reports the death of Professor Miller on June 12, 1933. William Ferguson was appointed a member of the Subcommittee on Rules for Inspection.

Respectfully submitted,

F. R. Low, *Chairman*	A. M. Greene, Jr.
D. S. Jacobus, *Vice-Chairman*	F. B. Howell
C. W. Obert, *Honorary Secretary*	J. O. Leech
H. E. Aldrich	M. F. Moore
W. H. Boehm	I. E. Moultrop
R. E. Cecil	C. O. Myers
F. S. Clark	H. B. Oatley
W. F. Durand	James Partington
E. R. Fish	C. L. Warwick
V. M. Frost	A. C. Weigel
C. E. Gorton	H. LeRoy Whitney

ECONOMIC STATUS OF THE ENGINEER

In 1931 this Committee rendered two reports, one on the Earnings of Mechanical Engineers in 1930 and the other a recommendation for the establishment of a joint agency of engineering bodies through which the general status of the engineering profession could be improved.

The general economic situation has made it undesirable for the Committee to take active steps during the present emergency. The Committee, however, is holding itself in readiness for action, and its findings on previous occasions will be of tremendous value within the next year or so when general industry comes out of the depression and engineers are in greater demand.

As a result of the action of this Committee, the Engineers' Council for Professional Development,* which was organized in temporary form on October 3, 1932, has developed a charter and rules of procedure which have been adopted by the seven constituent bodies. It is expected that the formal organization of ECPD will be perfected early in October, 1933.

Respectfully submitted,

C. F. Hirshfeld, *Chairman*	H. B. Oatley
D. S. Kimball	H. L. Whittemore
C. N. Lauer	W. E. Wickenden
J. M. Todd, D. B. Prentice	*Ex-Officio.*

REGISTRATION OF ENGINEERS

The Committee on Registration of Engineers submitted to the Council at the last Annual Meeting the recommendation that the Society approve the adopting of the Model Law for the Registration of Professional Engineers and Land Surveyors by any state or municipality desiring to enact an engineering registration law. The Committee did not ask the Council to approve the principle of registration of engineers. The Council approved the recommendation of the Committee.

The Legislature of the State of New York passed a law during its last session rectifying the wrong which had been done to the engineering profession a few years ago. Engineers are now on the same level with architects as far as the filing of building plans is concerned.

The Committee again calls to the attention of the membership of the Society the necessity of engineers' knowing about the requirements of the registration laws. A member should consult with a competent authority in his state if he has any doubt about his standing to carry on professional work. Engineers must realize that the laws are on the statute books of many states and even though failure to register may result in no immediate difficulty, the lack of a license may be a bar to promotion to a position of greater responsibility. In some states it is becoming more difficult each year to obtain a license.

Respectfully submitted,

J. H. Lawrence, *Chairman*	C. F. Hirshfeld
D. S. Kimball	V. M. Palmer
Paul Doty	J. M. Todd.

Joint Activities

Reports for the year 1932–1933 were also presented to the Council by representatives of the Society on a number of joint activities. Reference to some of these will be found in the Report of the Council and others are presented or summarized in the following pages.

AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE

SECTION M, ENGINEERING

The American Society of Mechanical Engineers, in cooperation with other engineering societies, participates in the conduct of Section M, Engineering, of the American Association for the Advancement of Science.

* For report of A.S.M.E. representatives on Engineers' Council for Professional Development, page 46.

During the fiscal year starting October 1, 1932, Section M held two meetings—one at Atlantic City and one at Chicago. The session at Atlantic City was given over to a joint discussion with Section K, Economics and Social Sciences, of a number of important economic problems. The papers presented were published in a separate book and received widespread attention.

The Chicago meeting of the American Association for the Advancement of Science immediately preceded Engineering Week. The meetings of Section M were held during Engineering Week. In view of the comprehensive technical program at Chicago, under the auspices of some twenty-eight engineering societies, Section M conducted only one session, at which a paper was presented by A. P. M. Fleming, of Manchester, England, on "The Industrial Developments of the Century." Section M participated in the general planning of the meeting, however, provided the liaison between Engineering Week and Science Week, and aided by securing the participation of several distinguished engineers and scientists from other countries.

During July, 1933, N. H. Heck, who had served as secretary of Section M for seven years, retired, and his position is to be taken up by Vannevar Bush, dean of the Massachusetts Institute of Technology.

Plans for the meeting of Section M in Boston, in connection with the A.A.A.S. meeting during the week of December 27 to January 2, are already under way.

Respectfully submitted,

W. S. Monroe, *A.S.M.E. Representative.*

AMERICAN STANDARDS ASSOCIATION*

Without doubt the most important event of the year for the American Standards Association is the transfer to it of the standardization activities and code work of the following five divisions of the Federal Government: Division of Trade Standards, Division of Specifications, Division of Simplified Practice, the Building Code and Plumbing Code work of the Building and Housing Division, and Safety Codes. While the direct reason for these transfers was the need for economy of Government expenditures, it is believed by many to be partly the result of natural trends in industry. The functions of these several groups will be absorbed by the ASA staff as fast as it is physically possible and the finances of the Association will permit. The July issue of the ASA monthly publication, "Industrial Standardization," was a joint publication with the "Commercial Standards Monthly." Howard A. Coonley, president of the ASA, and the members of his Board of Directors are taking steps which they hope will lead to the adequate financing of this enlarged activity of the ASA.

The standard having the most general interest among those approved during the year was an American Standard Practice for the Inch-Millimeter Conversion Factor for Industrial Use. By securing the approval of technical societies and industry on the ratio of 25.4 mm to the inch, the ASA has helped to clarify a very confused situation where slightly different equivalents had been in use. It is especially significant that this agreement was reached in the general conference which was called for this purpose without a dissenting vote.

The Annual Joint Meeting of the ASA Board of Directors and the Standards Council was held in New York on November 30, 1932. At this meeting the election of President Coonley and the reelection of C. M. Chapman as chairman of the Standards Council, were announced. Both men are members of the A.S.M.E. and the latter is one of its two representatives on the Standards Council.

The mid-year meeting of the Standards Council was not held in June, but throughout the year it has continued to function with the assistance of the staff by correspondence. Eleven new projects have been proposed during the twelve months, and within the same period forty-one completed standards were approved, receiving the designation "American Standard." These additions bring the total number of standards approved by the ASA up to 245, of which 58 are in the mechanical field.

The Year Book of the Association issued in May reports the number of member-bodies cooperating with the ASA as 28 societies and associations and 9 Federal Government departments. To these are added 245 firms and individuals who maintain memberships. It should be kept clearly in mind, however, that the number of organizations officially cooperating with the Association is far greater than the number of member-bodies. They total approximately six hundred.

Respectfully submitted,

C. M. Chapman
K. H. Condit } *A.S.M.E. Representatives.*

* See report of A.S.M.E. Standing Committee on Standardization, page 38, for projects for which the Society is a sponsor.

AMERICAN ENGINEERING COUNCIL

Most important meetings of the Assembly and Administrative Board of the American Engineering Council were held in Washington January 13 and 14, 1933. A complete record of these meetings is contained in the minutes which have been filed at the Society's office for reference. There follows a list of the most important matters which claimed the attention of these great engineering conferences, attended by 49 representative engineers from all parts of the United States:

Report of Committee on Water Resources and Control
Report of Committee on Membership and Representation
Report of Committee on Engineering Features of the Public Domain
Report of Committee on Government Expenditures
Report of Committee on Public Affairs (including consideration of various bills before Congress)
Report of Engineers Committee on Economic Balance
Consideration of Resolution on Technocracy.

In addition to this Annual Meeting, attention should be called to the work of the American Engineering Council which goes on uninterruptedly day by day.

(1) The A.E.C. has never had a busier year. The last Congress proposed and enacted legislation which will give entirely new direction to social, economic, and industrial trends of the nation. The A.E.C. has been closely identified with this movement.

(2) The A.E.C. has concerned itself with the new governmental agencies—National Industrial Recovery Administration, Federal Emergency Administration of Public Works, Tennessee Valley Authority, and Agricultural Adjustment Administration. Our Washington office is constantly in touch with these agencies, aiding them in every possible direction, submitting to them the names of qualified engineers, advising with them as to policies.

(3) Through the activity of the A.E.C., there has recently been secured from the $3,300,000 public works fund, seven or eight million dollars for geodetic and topographic surveys.

(4) Washington is the busy center of national activity today. Probably never in the history of the American Engineering Council has it been able to render greater or more direct service to the profession than during these last months.

(5) Some of our members have wondered whether the change from a Hoover to a Roosevelt Administration would change the influence of the A.E.C.; our Washington office is enjoying direct and influential contacts with the Roosevelt Administration.

Respectfully submitted,

A.S.M.E. Representatives

A. A. Potter, *Chairman*
D. Robert Yarnall, *Vice-Chairman*
L. P. Alford
B. M. Brigman
W. S. Conant
Hugo Diemer
Paul Doty
C. E. Ferris
R. E. Flanders
W. A. Hanley
E. C. Hutchinson
Ernest Kreher
C. N. Lauer
J. H. Lawrence
R. C. Marshall, Jr.
G. E. Pellissier
J. W. Roe
W. H. Trask, Jr.
B. R. Van Leer
T. R. Weymouth.

THE ENGINEERING FOUNDATION

The Engineering Foundation Board is a body of representatives of the four Founder Societies associated for the purpose of fostering research and the application of the sciences through engineering, and of performing other functions, under its deeds of gift and charter, "for the good of mankind." It has an endowment of approximately $880,000 which is held by the United Engineering Trustees, Inc., as trustee for the Founder Societies. It functions chiefly in assisting, by the appropriation of funds or otherwise, specific researches organized and directed by the Founder Societies.

The Engineering Foundation Board has held three meetings, and

its Executive Committee three meetings, in New York during the period from October 1, 1932, to October 1, 1933.

There follows a review of noteworthy activities during the year:

(1) *Founder Society Projects.** Within the calendar year 1932, an amount of $26,500 was appropriated to the Founder Societies to aid certain specific projects. An additional $9000 was appropriated for special projects relating to engineering education, a preliminary study of the industrial system, and the Personnel Research Federation.

The budget for the calendar year 1933 totals $49,900, of which $6000 is set aside as reserve for contingencies, and $3262.50 is unappropriated. The remaining amount, $40,637.50, is broken up into $12,937.50 for administration, office, and publication expenses, and $27,700 for research projects. This latter amount is apportioned as follows:

A.S.C.E.	$7,200
A.I.M.E.	7,000
A.S.M.E.	8,500
A.I.E.E.	2,000
Engineering education	500
Summer school for engineering teachers	1,000
Personnel Research Federation	500
Transport research	1,000
Total	$27,700

(2) *The Engineering Societies Library.* The Engineering Societies Library reported an expected deficit of $4000 for the year 1933. Portions of this deficit may be made up from certain sources, but for the remainder no other source than the Foundation is in sight. Mr. Yarnall reported for the representatives of the A.S.M.E. that the Society would consent to diverting $1000 from its appropriation to the research committees to the Library. The Executive Committee of the Foundation was instructed to work out the problem of providing for the Library such portion of its expected deficit for the current year as cannot be provided for by other sources than the Foundation.

(3) *New Platform.* Consideration was given to the possibility of revising the Foundation's Program of Action adopted in 1925. A special committee submitted a new program intended to increase the usefulness of the Foundation to the advancement of the engineering profession and the good of mankind. A large conference held in May, 1933, and attended by the Foundation Board and representatives of the Founder Societies, discussed the matter at length and expressed varied opinions that ranged from favoring concentration on engineering research to calling for inclusion of other kinds of activities. The proposed new platform is now being studied by the Foundation's Executive Committee for report at the October meeting of the Board. It is understood that any basic change in policy or procedure would be submitted first to the Founder Societies for approval.

(4) *Courses for Disengaged Engineers.* On the initiation of the Engineering Foundation, advanced study courses were provided for disengaged engineers in the Engineering Societies Building from January through May. Certain colleges in the New York and New Jersey area laid out the courses and the teaching was done by professors from these colleges, unemployed teachers and engineers. The Professional Engineers Committee on Unemployment financed the project. More than 600 engineers took the courses, which were considered very helpful.

(5) *Basic Study of the Industrial System.* Conferences between the representatives of A.E.C. and Engineering Foundation on the financing of a joint study of economic balance continued during the first half of the year. It was finally decided that the financing of such an investigation at the present time would not be successful.

(6) *Publications.* A pamphlet entitled "Engineering: A Career—A Culture," prepared by a special committee headed by H. N. Davis, was published and 100,000 copies were distributed through libraries, schools, etc.

It was decided to discontinue the publication of "Research Narratives."

The final report of the arch dam investigation sponsored by the A.S.C.E. was published and the committee discharged.

Respectfully submitted,

D. Robert Yarnall
E. R. Fish
A. E. White
A.S.M.E. Representatives.

* See report of A.S.M.E. Standing Committee on Research, page 35, on projects toward the support of which The Engineering Foundation has contributed.

ENGINEERS' COUNCIL FOR PROFESSIONAL DEVELOPMENT

The Engineers' Council for Professional Development is a direct outgrowth of the work of the Committee on the Economic Status of the Engineer.* Following the recommendations of the latter committee, the A.S.M.E. issued invitations to certain other organizations to appoint representatives to meet and discuss the advisability and means of joint action leading toward the professional development of the engineer.

Out of this invitation there developed a meeting on April 14, 1932, at which a general plan was evolved for the inauguration of a joint body consisting of representatives of the following organizations:

American Society of Civil Engineers
American Institute of Mining and Metallurgical Engineers
The American Society of Mechanical Engineers
American Institute of Electrical Engineers
American Institute of Chemical Engineers
Society for the Promotion of Engineering Education
National Council of State Boards of Engineering Examiners.

The idea was to include not only engineering societies as representative of what may be called the professional aspects of the engineer's life, but also bodies representing the educational and legal aspects. The name suggested for this joint body was Engineers' Council for Professional Development.

All seven of the organizations named acted favorably on the recommendations of this joint committee and authorized the organization of such a Council with three members from each of the cooperating bodies. At a meeting held in October, 1932, the Council was organized with a temporary set of officers, a constitution and program were adopted for submission to the parent bodies, and working subcommittees were appointed so that there might be no delay in starting what is believed to be a most important move on the part of the engineering profession.

All the cooperating bodies have now accepted the recommendations of the Engineers' Council with respect to its composition, its constitution, and its program for the immediate future. It is therefore a going concern at the present time.

This Council has defined itself as "**** a conference of engineering bodies organized to enhance the professional status of the engineer through the cooperative support of those national organizations directly representing the professional, technical, educational, and legislative phases of an engineer's life."

It has stated its objective as: "**** the enhancement of the professional status of the engineer. To this end it aims to coordinate and promote efforts and aspirations directed toward higher professional standards of education and practice, greater solidarity of the profession, and greater effectiveness in dealing with technical, social, and economic problems."

The Engineers' Council has already appointed four working subcommittees and these are organized and working. Their names, chairmen, and tentative statements of their immediate objectives are:

(1) Committee on Student Selection and Guidance; chairman, R. L. Sackett, dean, College of Engineering, Pennsylvania State College. This committee "shall report to ECPD means for the educational and vocational orientation of young men with respect to the responsibilities and opportunities of engineers in order that only those may seek entrance to the profession who have the high quality, aptitude, and capacity which are required of its members."

(2) Committee on Engineering Schools; chairman, Karl T. Compton, president, Massachusetts Institute of Technology. This committee "shall report to ECPD means for bringing about cooperation between the engineering profession and the engineering schools. As an immediate step, the committee shall report criteria for colleges of engineering which will insure to their graduates a sound educational foundation for the practice of engineering."

(3) Committee on Professional Training; chairman, R. I. Rees, assistant vice-president, American Telephone and Telegraph Company. This committee shall report to ECPD plans for the further personal and professional development of young engineering graduates, and also of those without formal scholastic training.

(4) Committee on Professional Recognition; chairman, Conrad N. Lauer, president, Philadelphia Gas Works. This committee shall report to ECPD methods whereby those engineers who have met suitable standards may receive corresponding professional recognition.

The character of the men who have accepted the chairmanship of these committees and who are now actively engaged in producing

* See report of Committee, page 44.

results through those committees speaks for itself. There can be no doubt of the value of the plans of the Engineers' Council when such men will devote large fractions of their time to its purposes. These men all realize that the time has arrived at which the engineer begins to assume a different place in the social structure and additional and more important responsibilities than those he has carried heretofore. They also realize that it is necessary that he prepare himself properly for such responsibilities and that he provide means within the profession for maintaining such standards of professional ability and ethics as shall be required under these new conditions.

The Engineers' Council for Professional Development has a huge and important task before it. This task can be accomplished satisfactorily only if it has the whole-hearted cooperation of the bodies represented on it, the individual members of those bodies, and all other engineers throughout the country.

Respectfully submitted,

C. F. Hirshfeld
J. H. Lawrence
W. E. Wickenden
A.S.M.E. Representatives.

INTERNATIONAL ELECTROTECHNICAL COMMISSION

U.S. NATIONAL COMMITTEE

In 1924 the first World Power Conference, held in London, requested the International Electrotechnical Commission to take up actively the development of international agreements relative to test codes or specifications for prime movers. Prior to that time committees in Great Britain, Switzerland, Germany, and the United States had been at work developing national codes for water turbines, steam turbines, internal-combustion engines, etc. In February, 1925, the U.S. National Committee of the I.E.C. invited The American Society of Mechanical Engineers to accept membership, recognizing the A.S.M.E. Committee on Power Test Codes* as the authoritive group in the United States on the testing of prime movers and other auxiliary apparatus. This invitation was accepted and the Society through its representatives on the U.S. National Committee has since taken an active part in the international conferences on this subject.

In December, 1931, with the completion of the organization of the Electrical Standards Committee, the U.S. National Committee of the I.E.C. became identical in personnel with the E.S.C., plus three representatives of the A.S.M.E. and certain individuals appointed by the Committee. By this reorganization all the national bodies having interest in the work of the I.E.C. are directly represented on the international committee, and the work of the U.S. National Committee of the I.E.C. is closely associated with that of the American Standards Association.

The U.S. National Committee of the I.E.C. functions as Secretariat for Advisory Committees No. 1 on Nomenclature, No. 4 on Hydraulic Turbines, No. 5 on Steam Turbines, No. 14 on Rating of Rivers, and No. 19 on Internal-Combustion Engines. During the year preparations had been made for meetings in the United States of Advisory Committees No. 1 on Nomenclature, No. 2 on Rating of Electrical Machinery, No. 5 on Steam Turbines, and No. 19 on Internal-Combustion Engines, but due to the unsettled conditions generally, these plans have been abandoned indefinitely. It is expected, however, that wherever possible the programs of these committees will be carried on by correspondence until meetings can again be scheduled, although it is recognized that the work of Advisory Committees No. 5 on Steam Turbines and No. 19 on Internal-Combustion Engines, has reached a point where little more can be accomplished without international advisory committee meetings.

Respectfully submitted,

H. N. Davis
E. C. Hutchinson
Francis Hodgkinson
A.S.M.E. Representatives.

NATIONAL MANAGEMENT COUNCIL

Under the leadership of Harry A. Hopf there has been brought into being a joint body known as the National Management Council. The purpose is to bring into one fellowship all of the societies working in the management field, in order that their efforts may be coordinated and constructive work planned and carried through to realization.

The organization meeting was held on June 8, 1933, at which time officers were elected. The Constitution and By-Laws of the National Management Council had previously been ratified by seven management organizations, including The American Society of Mechanical Engineers for its Management Division.

Too short a time has elapsed to show definite results from this new management group. One responsibility, however, has been taken over, that of organizing and planning for International Management Congresses, a task formerly exercised by the International Management Congress Committee.

Respectfully submitted,

L. P. Alford
H. V. Coes
F. E. Raymond
A.S.M.E. Representatives.

NATIONAL RESEARCH COUNCIL

DIVISION OF ENGINEERING AND INDUSTRIAL RESEARCH

The purpose of the Division is to encourage, initiate, organize, and coordinate fundamental research in the field of industry and to serve as a clearing house for research information of service to industry.

It is one of eleven Divisions of the National Research Council, which is a cooperative organization for scientific and technical men of America, most of whom are representatives of the major scientific and technical societies. The administrative expenses of the National Research Council are largely paid from an endowment, but the actual research projects sponsored by the Division are financed by interested industries or from other sources in a manner similar to the A.S.M.E. system of research sponsorship.

The Division of Engineering is made up of five representatives of the four Founder Societies* and of representatives from the A.S.R.E., A.S.T.M., A.S.S.T., A.S.H.&V.E., I.E.S., W.S.E., S.A.E., and A.W.S. Charles F. Kettering, vice-president and general director of research, General Motors Corporation, is chairman of the Division this year.

Projects

Electrical Insulation. The Committee on Electrical Insulation held its annual meeting in connection with the Baltimore Regional Convention of the A.I.E.E. There were several sessions devoted to papers on progress in installation research, problems in connection with high voltage cables, etc. These papers and reports were published in several issues of *Electrical Engineering.*

Welding. A conference on Fundamental Welding Research was held during the Annual Meeting of the American Welding Society, at Buffalo, in October, 1932. A comprehensive list of fundamental research problems in the welding field that need solution has been compiled by the Fundamental Research Committee of the American Bureau of Welding and published in the December, 1932, issue of the *Journal* of the American Welding Society. Forty investigations are now under way in the various institutions of the country. Reports were submitted during the year by a number of these investigators.

At the meeting of the American Bureau of Welding in April, 1933, progress reports were presented by the Committee on Welding Wire Specifications, which has practically completed a new set of specifications for welding wire. A new investigation on structural steel welding was promulgated, and investigation on the impact resistance of welded joints is under way. The final report of the Welded Rail Joint Committee was published in September, 1932, and its 350 pages summarize the results of a ten-year program of investigation.

Highways. At the Twelfth Annual Meeting of the Highway Research Board in Washington, December, 1932, thirty papers and reports were presented dealing with factors of highway management and automobile design from the point of view of road performance as well as highway construction and maintenance. Two books were published covering these reports and the results of the special investigations. The special project on "Best Laying Methods," has been actively prosecuted during the year and two new projects were organized, one on "Roadside Development" and the other, "Use of High Elastic Limit Steel as Concrete Reinforcement."

Heat Transmission. This Committee is bringing to completion the preparation of a critical digest of information on heat trans-

* See report of Committee, page 41.

* See report of A.S.M.E. Standing Committee on Research, page 35.

mission. A comprehensive treatise on the several types of heat transmission, based on considerable unpublished information, as well as a critical examination of the literature, was prepared by the committee under the supervision of W. H. McAdams, of M.I.T., and published in book form in December, 1932, by the McGraw-Hill Book Company. A book on heat insulation is now in preparation.

University Relations

A plan for regional conferences was inaugurated in July, 1932, for the study of university research facilities and research capacities of the engineering research personnel of all universities in the country. The sum of $4600 has been made available for the conduct of a national survey of these resources. The purpose of this survey is to carefully catalog not only the physical facilities but also the various capacities of these universities to do research. Conferences will be held in suitably located centers in each of eight geographical districts. The first conference, which was held in Chicago in April, 1933, was a considerable success, and the next will be held in New York in October, 1933.

Surveys

A number of surveys initiated early in the year were continued. An analysis of the research expenditures in the chemical industry appeared in the August, 1932, issue of *Chemical and Industrial Engineering Chemistry*, which indicated the relative amount spent on research for 1931 as compared with 1929; the comparison of research programs with size of companies; and the change in trends in emphasis of research programs. Similar reports were published covering the food, metals, and machinery industries.

Addresses

A number of addresses by officers of the Divisions were presented before appropriate groups representing executives, trade associations, and technical societies. Aid is being given to several trade associations in organizing cooperative research programs.

Conclusions

A thorough examination of the accomplishments of the Division during the past fourteen years has led to the conclusion that there is a place for the Division which is not being filled by any other existing organization. This field may be briefly divided into three main divisions. First, the administration of research projects of national scope involving several branches of engineering or science; second, promotion and stimulation of research in universities, trade associations, and private companies; and third, maintenance of a central clearing house for research information in the industries which support research through the maintenance of about 1600 industrial research laboratories.

Respectfully submitted,

D. S. Jacobus, *Vice-Chairman*
W. D. Ennis
F. M. Farmer
G. W. Lewis
Albert Kingsbury
W. R. Webster
Calvin W. Rice, *Ex-Officio*

A.S.M.E. Representatives.

UNITED ENGINEERING TRUSTEES, INC.

On the initiative of senior national societies of engineers this corporation was granted a special charter by the State of New York for "the advancement of the engineering arts and sciences in all their branches, and to maintain a free public engineering library.... It shall have power to take real and personal property and to use, maintain, occupy, lease, mortgage, and convey the same."

Its Founder Societies are the American Society of Civil Engineers, American Institute of Mining and Metallurgical Engineers, The American Society of Mechanical Engineers, and American Institute of Electrical Engineers. For them it holds and administers the Engineering Societies Building and Library, and several funds. It has three departments, Administrative, Library, and The Engineering Foundation, the last devoting its energies mostly to research. It may create other departments at the direction or with the approval of the Founder Societies.

The financial resources of the corporation are payments made by the Founder and Associate societies for the use of offices and meeting halls in the building and for the support of the Engineering Societies Library, income from endowment and other funds, and special contributions. The funds include endowments for the Library ($224,500) and the Foundation ($883,000) and a depreciation and renewal fund ($317,800) for the building, which with the land cost $2,000,000. The depreciation and renewal fund is increased each year by the addition of its own income and a stipulated allotment from other sources, and against it are charged major repairs.

The book value of the securities owned for all funds is	$1,394,355.40
and the market value as of Dec. 31, 1932, is	1,102,741.88
Shrinkage, 21 per cent	$ 291,613.52

Total annual income from all sources would be, for normal conditions, $246,300, divided as follows:

From endowment funds	$ 53,000
From Founder Societies	60,000
All other sources	78,300
Estimated income 1933	191,300
Reduction of income due to depression	55,000
Total income, normal years	$246,300

The corporation's annual expenditures, and those of its departments, in *recent normal* business times, are compared below with the budget for 1933:

	Normal annual expense	Budget for 1933
(1) Salaries and wages for operating building	$46,200	$41,000
(2) Electric current, steam heating, and supplies for operating building	42,000	33,500
(3) Depreciation and renewal allotment for building	12,000	.....
(4) Telephone expense—private branch exchange for building tenants	15,400	13,500
(5) Bookkeeping, auditing, secretarial, financial, legal, and other services and insurance	18,400	14,100
(6) Operating cost Engineering Societies Library	68,800	53,000
(7) The Engineering Foundation		
(a) Administrative expense	4,200	4,100
(b) Promotion of research	6,300	5,100
(c) Allotment of Engineering Foundation income to all projects (part from previous unexpended balance)	33,000	31,500
Total expenditures	$246,300	$195,800

The difference between the 1933 estimated income of $191,300 and the budget of $195,800, or $4500, is more than offset by certain items of income from investments, classified as doubtful as a precaution of extreme conservatism at the time the income was estimated, but on which substantial payments have been received.

The corporation is administered by a Board of twelve Trustees, three appointed by each Founder Society for three-year terms. The Board elects from its members a president and two vice-presidents; also a treasurer, an assistant treasurer, and a secretary, who need not be members. The Board appoints the following standing committees: Finance, House, Memorials, and American Engineers' Memorial, the last an international body connected with the war memorial at Louvain.

Its executive officer is the secretary (part-time). It has a clerical, mechanical, and janitorial staff totaling 34.

The Library is managed by a board of 21 members, including the secretaries of the four Founder Societies, and is operated by a director with a staff of library and clerical assistants, totaling 24.

The Engineering Foundation has a board of 16 members elected by the Board of Trustees. It has a director, a stenographer-secretary, and a junior clerk, a total of 3, all on part time.

The Coordination Committee of the Engineering Societies and its subcommittees have been active. Under date of May 1 a report was submitted to the Founder Societies and United Engineering Trustees, Inc., suggesting certain actions and inquiring whether the Coordination Committee should proceed with its studies. Action by the Board of Trustees was deferred pending action by the Founder Societies.

The Library financial needs have had repeated consideration on account of the decrease in contributions by the Founder Societies enforced by severe reductions of their incomes. Even the reduced budget, it developed, could not be fully met because of a deficit caused by falling off of the business of the Library's service departments and the expectation of failure to receive income from certain investments of endowment funds. The Engineering Foundation has underwritten this deficit. In June the Trustees appointed a Library Fund Committee, having for members the chairman and the vice-chairman of the Library Board, and a Trustee, on the request of the chairman of the Library Board. This committee is seeking new contributions toward the deficit and the endowment of the Library.

Financial Matters. The Finance Committee has held meetings at least monthly and some additional meetings. The budget for the year was drawn on the basis of reduced income, and salaries have

been cut and other decreases of expenditures made to meet the conditions. The American Society of Mechanical Engineers was forced to request deferment of payment of part of its obligations to United Engineering Trustees, Inc., until October and is giving its notes monthly for the amounts deferred. To meet this situation, the Trustees are borrowing from the depreciation and renewal fund until payment shall have been received from the A.S.M.E.

The Trustees appointed the firm of Wood Struthers & Co. its financial advisers about the first of the year and has found the service very helpful in the administration of the various investments.

A number of sales and reinvestments of securities have been made from time to time under the guidance of the investment adviser and the treasurer, in order to improve the soundness of the portfolio and insurance income.

Bank accounts were moved in March from the National City Bank to the Bankers Trust Company. Current payments of income to the Library and the Foundation were reduced because of doubt as to receipt of interest and dividends from a small number of investments. To date the full amount has been received from the questioned railroad securities and approximately 50 per cent of the interest due on mortgages on real estate.

If any of the members of the A.S.M.E. are interested in further details, they can be secured by addressing United Engineering Trustees, Inc., 29 West 39th Street, New York, and asking for a copy of "History, Charter and By-Laws" or for any of the reports and leaflets pertaining to the Library and the Foundation.

Respectfully submitted,

W. L. Batt
E. R. Fish
Harold V. Coes,
President, U.E.T.
} *A.S.M.E. Representatives.*

JOINT AWARDS

The A.S.M.E. participates in a number of joint awards. The following paragraphs give the purpose of these awards, the organizations represented upon the boards of award, the names of A.S.M.E. representatives for 1932–1933, and the awards made between October 1, 1932, and September 30, 1933.

John Fritz Medal, for notable scientific or industrial achievement; A.S.C.E., A.I.M.E., A.S.M.E., A.I.E.E.; W. L. Abbott, R. V. Wright, C. N. Lauer, A. A. Potter; award for 1933 to Daniel C. Jacklin, April 20, 1933.

Herbert Hoover Medal, for distinguished public service; A.S.C.E., A.I.M.E., A.S.M.E., A.I.E.E.; D. S. Kimball, Ambrose Swasey, Calvin W. Rice; no award.

Washington Award, for advancing human progress through engineering; A.S.C.E., A.I.M.E., A.S.M.E., A.I.E.E., Western Society of Engineers; W. I. Westervelt, G. F. Gebhardt; no award.

Daniel Guggenheim Medal, for notable achievement in advancement of aeronautics; A.S.M.E., S.A.E., Canada, England, France, Germany, Holland, Italy, Japan; G. W. Lewis, H. I. Cone, P. H. Adams, E. E. Aldrin; award for 1932 to Juan de la Cierva, June 28, 1933; award for 1933 to Jerome C. Hunsaker, August 31, 1933.

Gantt Gold Medal, for distinguished achievement in industrial management as a service to the community; Institute of Management, Management Division of A.S.M.E.; R. A. Wentworth, J. W. Nickerson, J. W. Roe; award for 1932 to Henry S. Dennison, June 30, 1933.

Alfred Noble Prize, for most meritorious technical paper published by participating Societies; A.S.C.E., A.I.M.E., A.S.M.E., A.I.E.E., Western Society of Engineers; H. L. Seward; award for 1932 to Frank M. Starr, January 23, 1933.

OTHER REPORTS

Reports for the year 1932–1933 were also received from the following organizations on which the A.S.M.E. is represented as indicated. These reports are on file at the headquarters of the Society and will be embodied in the report of the Secretary to the A.S.M.E. Council on December 4, 1933.

American Bureau of Welding, James Partington
International Management Congress Committee, L. P. Alford, W. L. Batt, Calvin W. Rice
Joseph A. Holmes Safety Association, O. P. Hood
Professional Engineers' Committee on Unemployment, W. A. Shoudy, Calvin W. Rice, H. N. Davis, H. V. Coes, W. W. Macon, A. H. Meyer, Walter Rautenstrauch, R. I. Rees
State Councils of Professional Engineers, Paul Doty, J. A. Hall, B. M. Brigman.

Service Rendered by the Office of the Society

The number and diversity of the activities of the Society are summarized in the Report of the Council and detailed in the Reports of Society Committees and Joint Activities. These are the reports of those appointed to bring to accomplishment the purposes of the members of the Society. The Office of the Society in turn, under the direction of the Secretary, carries out the policies and procedures laid down by the Council and Committees. The duties of the Office, therefore, are as numerous and as varied as are the activities of the Society.

The Office, then, serves the Council, its Executive Committee, and all standing and special committees and representatives on joint activities. It keeps all records, handles correspondence, arranges for meetings, assists the Council and Committees in their reports, and prepares material for publication.

Moreover, as national headquarters of the Society, not only the members but also an ever-increasing number of other individuals and organizations, both of the United States and from abroad, look to it for assistance and advice in many ways. Requests for information and help in matters of import to the entire profession, come to the Society from the four corners of the world.

In other words, the Office of the Society not only performs the regular duties connected with many committee activities but also acts as a service bureau for the members, and on behalf of the members, as a service bureau to the public, in so far as facilities and resources permit.

The Reports of Standing Committees show how their activities have been curtailed. Fewer meetings, reduction of volume of published material, and some curtailment of technical committee work, have reduced the volume of the work of the Office. But certain curtailments, such as the holding of fewer committee meetings, have added to, rather than subtracted from, the work of the Office; because progress once made in committee meetings is now attained through correspondence handled by the staff for the committees. Nevertheless, changes in office methods for the purpose of saving time as well as money, have been made, thereby permitting the accomplishment of effective work in spite of a decreased staff.

A reduction of 26 per cent in the operating expenses of the Society was made during the past year, 1932–33, from a budgeted allowance of $545,500.00 at the beginning of the year to an actual expenditure of $432,848.99. This has been largely accomplished by reducing the number of persons on the staff and by reducing salaries. Since October, 1931, the number of persons on the staff has been reduced from 107 to 68. This is exclusive of the Engineering Index employees, the number of which has been reduced from 36 to 18 since 1931. The percentages of reductions in salary rates from the high point of 1929–30 ranged from 40 per cent for those in supervisory positions to 22 per cent for those in lowest positions. As the result of the reductions both in the number of employees and in the salary rates, the total salary expense has been reduced 42 per cent, from $282,283.45 in 1929–30 to $173,519.95 in 1932–33.

The Office is organized in six departments as follows:

(1) *Supervision and General.* Calvin W. Rice, Secretary, and C. E. Davies, Executive Secretary, supervise the office operations and the staff service that is rendered to the Council, Executive Committee, other special committees of the Council, and the Committees on Finance, Publications, Constitution and By-Laws, and Professional Conduct.

General (J. E. Hannum, Office Manager). Includes accounting, incoming and outgoing mail, addressograph, stenographic, filing, stores, shipping, and other routine services.

Number of employees*—28.

* The number of employees is as of September 30, 1933.

1932-1933* Expenditures, $70,476.26; 1933-34 Budget, $58,000.00.

(2) *Field* (Ernest Hartford, Assistant Secretary). Provides service required in the operation of the meetings of the Society, the 16 Professional Divisions, 72 Local Sections, 108 Student Branches, the Committee on Awards, the Employment Service, and related matters.

Number of employees—10 in New York, 1 in Chicago, 1 in Tulsa.

1932-33 Expenditures, $37,277.76; 1933-34 Budget, $32,000.00.

(3) *Technical Committees* (C. B. LePage, Assistant Secretary). Provides service for the 439 Committees and subcommittees engaged under the general leadership of the Committees on Standardization, Research, Safety, Power Test Codes, and Boiler Code.

Number of employees—7.

1932-33 Expenditures, $27,160.38; 1933-34 Budget, $17,600.00.

(4) *Editorial* (George A. Stetson, Editor). Edits and publishes *Mechanical Engineering* text pages, Transactions, and the papers provided in the off-set process in lieu of printed papers. Material equivalent to 2600 Transactions size pages was handled in 1932-33.

Number of employees—6 (2 additional part-time).

1932-33 Expenditures, $24,440.59; 1933-34 Budget, $21,000.00.

(5) *Advertising and Sales* (Frederick Lask, Advertising Manager).

(*a*) Secures and publishes advertising in *Mechanical Engineering* and the Mechanical Catalog.

Number of employees—11.

1932-33 Expenditures, $39,430.58 (1932-33 Advertising income, $84,855.45); 1933-34 Budget, $30,000.00.

(*b*) Handles sale and distribution of codes, standards, reports, and special publications sold by the Society.

Number of employees—4.

1932-33 Expenditures, $9,347.08 (1932-33 Sales Income, $40,441.16); 1933-34 Budget, $9,300.00.

(6) *Engineering Index* (J. E. Hannum, Editor, F. Y. Stewart, Sales Manager). Edits, publishes, and sells the daily and weekly card service and the annual volume of The Engineering Index, which involves the annual preparation and printing of more than 40,000 descriptive references to information published in some 2000 current technical publications in 20 languages and received from 40 countries.

Number of employees—18.

1932-33 Expenditures, $44,524.68; no budget for 1933-34.

In addition to these budgeted allowances for 1933–34, $12,100.00 has been appropriated for rent, insurance, depreciation of furniture and equipment, relief, and contingencies, making the total budgeted allowance $180,000.00 as compared with an actual expenditure for 1932–33 of $221,863.29, exclusive of The Engineering Index.

CALVIN W. RICE, *Secretary.*

* Tabulation of Salaries, and other expenses for all office departments appears as Schedule 3 in Report of Finance Committee, page 23.

Reports of Meetings

THE annual reports of the Meetings and Program and Professional Divisions Committees, elsewhere in this issue, summarize the meetings of the Society and of its Professional Divisions for the fiscal year.

The Semi-Annual Meeting, held at Chicago, June 25–30, 1933, during Engineering Week of the Century of Progress Exposition, was reported in the August, 1933, issue of *Mechanical Engineering*, pages 481–485. The program as presented was substantially the same as the advance program printed in the June, 1933, issue of *Mechanical Engineering*, pages 393–400.

The Annual Meeting, held at New York, December 4–8, 1933, will be reported in the January, 1934, issue of *Mechanical Engineering*. The advance program was printed in the November issue, pages 713–720, and the final program differed from it only slightly.

Of the four meetings sponsored by Professional Divisions, or Groups, three have been reported in *Mechanical Engineering*.

An Air-Conditioning Conference was held by the San Francisco Section at the University of California, Berkeley, February 9–10, 1933, with the cooperation of the Air Conditioning Committee of the Process Industries Group. This conference was reported in *Mechanical Engineering*, April, 1933, page 272. Titles and authors of the papers presented are given in the report, and the papers themselves have been published in book form.

The July, 1933, issue of *Mechanical Engineering*, page 463, contains an account of the National Lubrication Engineering Meeting held at Pennsylvania State College, May 25–26, by the Lubrication Engineering Committee of the Petroleum Division in cooperation with the Central Pennsylvania Section and Pennsylvania State College. The proceedings of this conference have since been published as Technical Bulletin No. 18 of the School of Engineering of the college. A review of this bulletin to be published in the February, 1934, issue of *Mechanical Engineering* contains the titles of the papers presented and the names of their authors.

In the October, 1933, issue of *Mechanical Engineering*, pages 655–656, will be found an account of the National Oil and Gas Power Meeting, held at Atlantic City, N. J., August 23–26, 1933, under the auspices of the Oil and Gas Power Division. The account contains the titles of papers presented and names of authors.

The Applied Mechanics Division held a Pacific Coast meeting on January 20–21, 1933, with the cooperation of the Los Angeles Section and the California Institute of Technology at the Institute in Pasadena. The following papers were presented:

Fluid Mechanics

A New Apparatus for the Determination of the Size Distribution of Particles of Fine Powders, R. T. Knapp

The Determination of Viscosity by the U-Tube Method, E. D. Howe

The Measurement of Viscosities of Liquids Saturated With Gases at High Pressure, B. H. Sage

Viscosity of Oils at High Temperatures, F. L. Maker

Problems Related to Earthquakes

On the Problem of Earthquake Resistant Construction, R. R. Martel

Calculation of the Effect of Earthquakes on Buildings, M. Biot

Vibration of Continuous Frames, W. Porush

Vibrations of Buildings, P. Byerly

Water Pressures on a Tank During a Simulated Earthquake, L. M. Hoskins and L. S. Jacobsen

Design of the Pine Canyon Dam With Reference to Earthquake Stresses, S. B. Morris and C. E. Pearce

General Lecture

The Modern Revival of Applied Mechanics, Dr. Theodor von Karman

Fluid Mechanics

An Application of Theoretical Hydrodynamics to Pumps and Turbines, G. Wislicenus

A Criterion for the Instability of Laminar Velocity Profiles, W. Tollmien

The Water Jet Pump, M. P. O'Brien and J. E. Gosline

The Turbulent Mixing of Two Jets of Different Velocities, A. M. Kuethe

Experiments on Fluid Resistance of Ultra-Rough Surfaces, R. G. Folsom

Theory of Viscosity and Turbulence Pumps, A. Hollander

Vibrations of Mechanical Systems

Some Factors Which Affect the Vibration of a Single "Spring-Mass" System, M. Asimow

Variation of Harmonic Components of Exciting Force (and Torque), due to the Gaseous Pressure, in an Internal Combustion Reciprocating Engine of the Constant Volume Type With Conditions of Operation, L. M. K. Boelter

Periodic Forces Due to the Reciprocating Masses in V-Type Engines, C. J. Vogt

Vibration Characteristics of Automobile Headlamps and Methods of Test Employed to Predict Mechanical Performance, D. O. Rusk

Elasticity

Graphical Integration of Differential Equations, L. H. Donnell

The Ultimate Strength of Thin Sheets in Compression, E. E. Sechler

Effect of Shear Deflection on Axially Loaded Beams, A. S. Niles

Effect of "Creep" on Working Stresses as Applied to Steel for High-Temperature Pressure Vessels, F. L. Maker.

Necrology

ASHEL CYRUS ABELL (1884–1933)

Ashel Cyrus Abell, head of the department of mechanical engineering at The State College of Washington, Pullman, Wash., died suddenly on January 1, 1933. Professor Abell had been at the college since he went there as a student in 1907. He received a B.S. degree in mechanical engineering in 1912 and an M.S. in 1929. He joined the teaching staff as instructor in shop practice and after successive advancements became head of the mechanical engineering department in 1930. His teaching specialties were steam power heating and ventilation.

Professor Abell was born at Arkona, Ontario, Canada, on February 4, 1884. He was a machinist apprentice from 1900 to 1902 at the Waters Engine Works Company, Brantford, Ont., and from then until the fall of 1905 at the Canadian Locomotive Company, Kingston, Ont. During the next two years he was in charge of the motor department of the Canada Cycle & Motor Co., in Toronto, directing the construction of automobile engines, and for the first six months of 1907 was engaged in testing automatic screw machines for the Cleveland Automatic Screw Machine Company, Cleveland, Ohio.

Professor Abell became an associate of the A.S.M.E. in 1918. He was chairman of the Inland Empire Section at the time of his death. He belonged to several fraternal organizations, including the Masonic and Theta Xi.

EDGAR E. ARISON (1873–1933)

Edgar E. Arison, president of The Harrington Emerson Counselors, Indianapolis, Ind., died on April 22, 1933.

Mr. Arison was born on March 4, 1873, in Monongahela, Pa., the son of William Henry and Elizabeth (Coulter) Arison. After attending the United States Naval Academy at Annapolis, Md., he spent several years in civil, mining, and railway engineering with McKinney and Smith, Pittsburgh, Pa., and the Baltimore & Ohio R.R.

For ten years beginning in 1894 Mr. Arison was connected with The Carborundum Company, Niagara Falls, N.Y., engaged in general operating, administrative, and sales work. In 1904 and 1905 he was with The Ozone Vanillin Company, of Niagara Falls.

Mr. Arison became associated with Harrington Emerson in 1905, while Emerson was applying his principles of efficiency to the problems of the Atchison, Topeka & Santa Fe R.R. He continued to work with him until 1918, becoming vice-president, in 1910, of The Emerson Engineers. From 1918 to 1920 he was chief of the Western Division of the U.S. Army Ordnance, advancing from the rank of major to that of colonel. He had been president of The Harrington Emerson Counselors since 1926.

Mr. Arison became a member of the A.S.M.E. in 1916. He was a Presbyterian and a Mason.

Surviving Mr. Arison are his widow, Marie Wallace (Rae) Arison, whom he married in 1899, and four children, Mrs. Margaret Feldon, Wilmette, Ill., Lt. Rae E. Arison, U.S. Naval Academy, and Mrs. Ethel Heiberger and Alan Arison, Evanston, Ill.

THEODORE PROSPÈRE ARTAUD (1876–1933)

Theodore Prospère Artaud, who had been in ill health for about two years, died on August 5, 1933. Prior to his illness he was associated with the National Park Planning Commission in the extension of the system of parks in Washington, D.C.

Mr. Artaud was born on April 29, 1876, at Jackson, Miss. His early education was secured at the Passy School, Paris, and Colliere's, in Washington. He then attended Rock Hill College, Ellicott City, Md., for three years and Harvard University, receiving A.B. and M.S. degrees from the latter in 1895 and 1897.

From 1898 to 1900 Mr. Artaud served in the Infantry of the U.S. Army, and during the next two years taught mathematics and drawing at the University School, Baltimore, Md. In 1903 he entered the employ of the Hudson Companies, New York, with which he was connected in various capacities, leading to that of purchasing agent, until 1908, when he was made assistant to the vice-president and purchasing agent of the Hudson & Manhattan R.R. In 1912 he took on additional responsibilities as superintendent of the Hudson & Manhattan Terminal in New York and the following year was given the office of assistant general manager of the company.

Mr. Artaud terminated his connection with the Hudson & Manhattan R.R. in 1915 and since then had been located in Washington. For the greater part of the time he was connected with the Interstate Commerce Commission, in charge of land appraisal work, for some years executive assistant of the Bureau of Valuation. In 1929 and 1930 he was vice-president, and director of research, of the National Association of Owners of Railroad & Public Utilities Securities.

Mr. Artaud became an associate of the A.S.M.E. in 1907. He is survived by a brother, Major Frank E. Artaud, Hapeville, Ga.

JOHANNES IMMANUEL ASTROM (1873–1933)

Johannes Immanuel Astrom, whose death occurred on August 15, 1933, was born at Helsingfors, Finland, on March 9, 1873, the son of F. A. and Marie (Klein) Astrom. His ancestors were sea captains of Swedish descent, but naturalized in Finland.

Mr. Astrom secured his M.E. degree from the technical university at Helsingfors in 1896 and soon afterward came to the United States. He found employment as a draftsman with the Nordberg Manufacturing Company, Milwaukee, Wis., and remained with that company until 1908, except for short periods as draftsman with the E. P. Allis Co., Milwaukee, in 1897, C. H. Bradley, Jr. & Co., Pittsburgh, in 1898, and the Westinghouse Machine Company, East Pittsburgh, 1899, and as mechanical engineer at a shipyard in Helsingfors from November, 1900, to July, 1902. He held the position of designer when he finally left the Nordberg Manufacturing Company in 1908 to become chief engineer and vice-president of the Fort Wayne Engineering & Manufacturing Co., Fort Wayne, Ind., where he remained for 18 years. Since 1926 he had engaged in consulting work and since 1929 had also been chief engineer and general manager of the Water Refining Company at Fort Wayne.

Mr. Astrom held a number of patents covering improvements in pumps and related apparatus, such as water softeners, filters, etc.

Mr. Astrom became a junior member of the A.S.M.E. in 1900 and a member three years later.

AUGUSTUS BAINBRIDGE (1863–1932)

Augustus Bainbridge, whose death occurred on December 23, 1932, was born at Swindon, England, on December 7, 1863. His early education was secured in Sicily and at the Netherfield Academy, England, and he later attended the Liverpool School of Science, receiving a certificate for passing the government examinations upon the completion of his studies there, which continued for about ten years, beginning in 1880. During this period he worked in Liverpool for Messrs. Fawcett Preston & Co., drafting parts for marine, hydraulic, and sugar crushing machinery, etc., and for Messrs. John Jones & Sons, designing machinery for passenger and cargo steamers, dredges, boilers, and other equipment. He continued with these companies until 1898, when he became connected with the Palmers Shipbuilding & Dry Dock Co., Ltd., Jarrow-on-Tyne, England. After four years there in charge of design of propelling machinery and equipment for different types of vessels, he went into business for himself, having purchased a small general engineering shop at Newcastle-on-Tyne. He handled repair and construction work not only in the marine field but also in connection with refrigerating, ore crushing, and other plants.

In 1906 Mr. Bainbridge came to the United States, where he was employed for the first six years by the Newport News Shipbuilding & Dry Dock Co., Newport News, Va. He specialized on cylinder design but also worked on other features of a number of naval and merchant vessels. From 1912 to 1917 he was acting chief draftsman in the Machinery Division at the Charleston Navy Yard, and since then had held a similar position at the Brooklyn Navy Yard.

Mr. Bainbridge became a member of the A.S.M.E. in 1923.

LEIGH KNOWLTON BAKER (1894–1933)

Leigh Knowlton Baker, engineer at the National Carbon Company, Inc., Cleveland, Ohio, died on May 7, 1933. He was born at Washington, Pa., on May 29, 1894. After being graduated from the Case School of Applied Science, Cleveland, with a B.S. degree in mechanical engineering, in 1915, he worked for three years for the Upson Nut Company, successively as assistant master mechanic, foreman, and assistant general foreman.

In 1918 and 1919 he was a lieutenant in the Coast Artillery Corps of the U.S. Army. He then spent about a year in each of the following connections: research engineer and assistant to the president of the National Safe Company, assistant mechanical engineer, National Screw and Tack Company, designer, E. W. Bliss Co.,

and sales engineer, York Safe & Lock Co. During these years he directed the design and construction of safes and automatic machines for making and finishing nuts and bolts, and also engaged in research work.

Mr. Baker had been with the National Carbon Company since 1924. After being at the Cleveland works for a time he was transferred as construction engineer to the Niagara Works of the company, and later was stationed at the Fostoria Works in Ohio. In 1928 he was transferred to the Edgewater Plant in Cleveland as works engineer, and in 1920 he was included in the staff of the General Engineering Department at the main office in Cleveland.

Mr. Baker became an associate-member of the A.S.M.E. in 1928. He had also been a member of the Cleveland Engineering Society. He is survived by his widow, Helen (Shackleton) Baker, and by two children.

JOHN SIDNEY BARELLI (1867–1933)

John Sidney Barelli, representative and manager of the New Orleans office for the Heine Safety Boiler Company, died on April 11, 1933. Mr. Barelli was born in New Orleans on June 2, 1867, and attended public school there. He then served an apprenticeship as a machinist with H. Dudley Coleman & Bros., New Orleans, from 1884 to 1888, and was with the Gullet Gin Company, Amite, La., during the next three years. Between 1891 and 1897 he worked at the Whitney Iron Works, Southern Electric Works and New Orleans Traction Company, all of New Orleans.

Mr. Barelli engaged in machine-shop work as a member of the firms of Barelli & Gruber and Barelli & Kittredge from 1897 to 1904. During the next two years he was chief engineer of the Schutton & Fallon Marble Works and the following year held the same position with the DeSoto Hotel. He had been connected with the Heine company since 1907.

Mr. Barelli was a member of the Louisiana Naval Militia from 1898 to 1916, rising to the rank of lieutenant-commander and chief engineer. He joined the A.S.M.E. in 1917. He was president of the New Orleans Chess, Checkers and Whist Club and of the Pine Island Fishing Club and was an Episcopalian. He is survived by his widow and by a daughter, Mrs. Leonard Scheuermann.

HENRY GREGORY BARNHURST (1872–1933)

Henry Gregory Barnhurst, middle-west representative of the Cement Sales Department of The Babcock & Wilcox Company, died of heart failure during the night of March 18, 1933, while asleep at the home of a co-worker in the company's Chicago office. His home was in Forest Hills, Long Island, N.Y.

Mr. Barnhurst was born in Philadelphia, Pa., on June 15, 1872. After being graduated from Haverford School, Haverford, Pa., he spent seven months in the drafting room of V. W. Payne & Sons Engine Co., Elmira, N.Y., and two years as an apprentice machinist with the Humboldt Iron Works, Erie, Pa. He worked in Erie for about six years, four as draftsman and general office assistant for the Union Iron Works, and two as draftsman in the Boiler Department of the Erie City Iron Works and with the Ball Engine Company, manufacturers of high-speed engines.

During the next nine years he worked in positions of increasing responsibility: draftsman for the Atlas Portland Cement Company, New York; mechanical engineer for the Lehigh Portland Cement Company, Allentown; construction engineer for the Mississippi Valley Portland Cement Company; and superintendent of the Diamond Portland Cement Company, Cleveland, Ohio.

From 1906 until recently Mr. Barnhurst had been associated with the same organization, the Fuller Engineering Company and its successor, the Fuller-Lehigh Company. He was chief engineer, with headquarters in Allentown, Pa., until 1924, when he was made advisory engineer, stationed in Fullerton, Pa. His work in the early development of pulverized fuel, as well as in the design and construction of cement plants and other activities in the cement industry, made him well fitted for his appointment as representative of The Babcock & Wilcox Company, when that company took over the Fuller-Lehigh Company in 1926.

Mr. Barnhurst became a member of the A.S.M.E. in 1916.

EDGAR GRANT BARRATT (1864–1933)

Edgar Grant Barratt, who died on February 16, 1933, was born in New York, N.Y., on February 8, 1864, the son of Oliver and Frances (Terry) Barratt. After securing a C.E. degree from the Columbia School of Mines in 1884 he went to Chicago to work for the Exhaust Ventilator Company. He was engineer in charge of erection and subsequently president of the company.

About 1900 Mr. Barratt became chief engineer for the Union Bag & Paper Co., Chicago. He returned to New York a few years later and opened his own office there. In 1905 he again became associated with the Union Bag & Paper Co., as first vice-president, with office in New York. Three years later he was made president of the company, with which he remained until 1925. Since then he had engaged in consulting work.

Mr. Barratt became a member of the A.S.M.E. in 1888. He also belonged to the University Club of Chicago, City Club and Columbia University Club in New York, and several clubs in New Jersey. He made his home in East Orange, N.J. He is survived by his widow, Katharine Bridge (Forman) Barratt, whom he married in 1889, and by a son, Roswell Forman Barratt.

ELMER D. BARRY (1870–1933)

Elmer D. Barry, whose death occurred at Mackenzie City, Demerara, British Guiana, on July 23, 1933, was born at Irasburg, Vt., on February 4, 1870. His first position after leaving high school was in the Turbine Department of the General Electric Company, working in the shops and on the road on the assembly, testing, installation, and maintenance of turbines. After four years in this position he became chief operating engineer, in 1900, of the Stone & Webster Engineering Corp. Of his six years with that company two were spent in charge of a power plant at Key West, Fla., two at Jacksonville, Fla., in charge of a traction company power plant, and two at Savannah as chief engineer of the Savannah Electric Company, in full charge of power plant and substation.

From 1916 to 1919 Mr. Barry was with the Westinghouse Electric & Manufacturing Co., engaged in the erection, testing, and maintenance of turbines and condensers in the Chicago district. During the next three years he handled similar work for the Kerr Turbine Company. In 1922 and 1923 he was chief operating engineer for the Crossett Lumber Company, Crossett, Ark., building a new power plant to change the drive of their mills from mechanical to electrical. The next few years were spent as chief operating engineer with the Arkansas Utilities Company, Helena, Ark., putting old equipment into usable condition, and subsequent power plant work took him to Texas and Marion, S.C. Since 1929 he had been superintendent of power for the Demerara Bauxite Company Limited, Mackenzie, Demerara, British Guiana, South America.

Mr. Barry became an associate member of the A.S.M.E. in 1924. He is survived by his widow, Frances C. Barry, whom he married in 1902, and by two daughters, Pauline Frances Barry and Helen J. Derry, and by one son, Robert Elmer Barry, all of Boston, Mass.

CARL BARTELS (1863–1933)

Carl Bartels, factory manager for the Mosler Safe Company, Hamilton, Ohio, died at his home in that city on February 21, 1933.

Mr. Bartels was born in Germany on April 14, 1863. He attended school there and for several years owned and operated a steam threshing outfit before coming to the United States in 1888.

During his first two years in this country Mr. Bartels was employed by the Chicago Safe & Lock Co. He became associated with the Mosler Safe Company, then located in Cincinnati, in 1890, and went with it to Hamilton the following year. He was superintendent of the Burglar Proof Department from 1900 to 1905, general superintendent of the company from then until 1917, and factory manager of the gun carriage shop during the next two years, and had been factory manager since then. He not only helped to build but also to design the safes, bank vaults, safe deposit boxes, locks, etc., manufactured by the company, and held many patents, either in his own name, or jointly with Mr. Mosler, the president of the company, on these devices and methods of building them.

Mr. Bartels became a naturalized citizen of the United States in 1894. He had been a member of the A.S.M.E. since 1921 and was active in the Masonic fraternity, Elks, and St. John's Evangelical Church. He is survived by four sons, Carl P., Leo F., and Edwin J. Bartels, all of Hamilton, and Dr. Elmer C. Bartels, of Duluth, Minn. His wife, Elizabeth (Reuther) Bartels, died in 1920.

HENRY CLAY BECKWITH (1882–1933)

Henry Clay Beckwith, president of The Byers Machine Company, Ravenna, Ohio, died on January 31, 1933. He was born on January 12, 1882, at Amenia, N.Y. He prepared for college at the Cazenovia Academy, Cazenovia, N.Y., and was graduated from the mechanical engineering course at Cornell University in 1903.

During the first eighteen months following his graduation Mr. Beckwith was employed as a draftsman by the Westinghouse, Church, Kerr Company, New York, N.Y., and for nearly two years longer

was assistant engineer for the company, in charge of the design of a service plant for the Pennsylvania Railroad terminal in New York.

From 1906 to 1908 he was in charge of design, construction, and equipment of a new plant for the Clyde Iron Works at Duluth, Minn. He was then made chief engineer and works manager and during the next four years was responsible for everything pertaining to the organization except finances, purchases, and sales. In April, 1912, his position was again changed, the company making him manager of the Pacific Coast sales office. He also became Portland manager for the Contractors Equipment Company of Seattle and chief engineer for all branches of this company. He opened offices and warehouses in Portland for both companies.

Mr. Beckwith next was engaged by The T. L. Smith Co. as Chicago district manager, but left there to become treasurer and general manager in 1915 of The Byers Machine Company. He was made president of the company in 1928.

Mr. Beckwith became a member of the A.S.M.E. in 1913.

GEORGE GORDON BELL (1884–1933)

George Gordon Bell, of Chesley, Ontario, Canada, who died suddenly of heart failure at Pittsburgh, Pa., on February 26, 1933, was born at Fort Wayne, Ind., on August 17, 1884, the son of William Douglas and Jeannie Stark Bell. He attended public and high school at Chesley and in 1905 was graduated with honors in electrical and mechanical engineering from the School of Practical Science of Toronto University. From then until 1907 he worked in the engineering office of the Canada Foundry Company in Toronto. He then returned to Toronto University, from which he secured a civil engineering degree in 1908.

During the years 1908–1910 Mr. Bell worked for the Canadian Bridge Company, Walkerville, Ont., on structural drafting and the design of mechanical equipment for drawbridges. He next spent two years in structural and hydraulic engineering for the Sawyer-Moulton Company, of Portland, Me.

In 1912 Mr. Bell went to Pittsburgh, Pa., where until 1930 he was connected with the West Penn Power Company. During the first year he worked on water power developments, then spent two years in charge of steam station design. In 1915 he took charge of power station operation and extensions, including the development of mines at the two largest power houses. From 1922 his duties related entirely to extensions to steam and proposed hydraulic plants and associated problems. Ill health forced him to retire in 1930.

Mr. Bell became a member of the A.S.M.E. in 1924. He also belonged to the Engineering Society of Western Pennsylvania, and was active on the Prime Movers Committee of the National Electric Light Association. He had made several contributions to the technical press and taken part in discussions of papers presented before the societies to which he belonged.

DANIEL HARVEY BRAYMER (1883–1933)

Daniel Harvey Braymer, whose death occurred on October 29, 1932, was born at Hebron, Washington County, N.Y., on November 29, 1883, the son of George N. and Jennie Cordelia (Smith) Braymer. He prepared for college at the Granville (N.Y.) High School and secured his A.B. degree from Cornell University in 1906, having specialized in chemical engineering. Two years later, upon the completion of courses in both mechanical and electrical engineering, he was given an M.E. degree.

During the summer of 1906 Mr. Braymer was draftsman for the Morse Chain Company, Ithaca, N.Y., and the next summer was electrical tester for the Electrical Testing Laboratories, New York, N.Y. He also held this latter position for a short time in 1908, later in that year becoming an engineer in the service and maintenance department of the New York plant of the Western Electric Company. In 1909 and 1910 he was electrical engineer handling publicity for the power plant and telephone equipment of the Western Electric Company.

Mr. Braymer entered the publishing field in 1910 and until 1915 was at Atlanta, Ga., on the staff of *Electrical Engineering* and its successor, the *Southern Electrician*, being editor the latter part of the time. He then returned to New York as associate editor of *Electrical World*. In 1917 he went over to the *Electrical Record* as editor-in-chief but in 1919 returned to the employ of the McGraw-Hill Company, becoming managing editor, and in 1921 coeditor of *Electrical World*. From 1922 to 1925 he was editorial director of *Industrial Engineer*, which succeeded the *Electrical Review* and the purpose and plan of which were conceived by Mr. Braymer.

Many of the articles for these magazines were prepared by Mr. Braymer from notes made during inspections of steam generating stations and hydroelectric plants in all parts of the United States. He was also the author of the following textbooks published by the McGraw-Hill Company: American Hydroelectric Practice (1917), Armature Winding and Motor Repair (1919), Rewinding Small Motors (1925), Repair Shop Diagrams and Connecting Tables (1927), and Rewinding and Connecting A.C. Motors (1931).

In 1925 Mr. Braymer formed the D. H. Braymer Equipment Company at Omaha, Neb., specializing in consulting and sales work in connection with ventilation, dehumidification, and air conveying and exhaust systems in industrial power plants.

Mr. Braymer became a member of the A.S.M.E. in 1928 and also belonged to the American Institute of Electrical Engineers. He is survived by his widow, Ruth (McGuire) Braymer, whom he married in 1925, and by his parents.

CHARLES EDWARD BROOKS (1886–1933)

Charles Edward Brooks, chief of motive power and car equipment for the Canadian National Railways, Montreal, Canada, died suddenly of heart disease at his home in that city on April 10, 1933. Prior to his connection with that road he had been with the Grand Trunk Pacific Railway.

Mr. Brooks was born in Constantinople, Turkey, on July 3, 1886, but was educated in Canada, attending primary schools in Montreal and vicinity and Orillia, Ontario, Trinity College School at Port Hope, Ontario, and McGill University, Montreal, from which he received a B.Sc. degree in 1908. During his early scholastic years his holidays were spent working as lumber mill hand, deck hand on a lake steamship, and boilermaker's helper. From May, 1905, to September of the following year he served as special apprentice on the Grand Trunk Railway at Point St. Charles, Quebec, and in the summer of 1907 was a special apprentice and fireman at Turcot, Quebec.

After graduation Mr. Brooks began work as a machinist with the Grand Trunk Pacific Railway and was promoted through positions of increasing responsibility to that of superintendent of motive power, with headquarters at Transcona, Manitoba, in 1915. When the Grand Trunk Pacific Railway was taken over by the Canadian National Railways in 1920 he was appointed mechanical assistant (Locomotive Department) to the vice-president in charge of operation and maintenance and was transferred to Toronto. In 1923 he returned to Montreal in the capacity of chief of motive power of the road and in July, 1932, also took charge of car equipment. He was keenly interested in the development of oil electric cars and the oil electric locomotive.

Mr. Brooks became an associate-member of the A.S.M.E. in 1917, and since 1928 had been a member of the general committee of the Railroad Division of the Society. He was also identified with committee work in the Mechanical Division of the American Railway Association, having been a member of its general committee and committee on locomotive design and construction. He served as president of the Canadian Railway Club in 1924-1925, and belonged to the University Club in Montreal and Kanawaki Golf Club, Quebec.

Mr. Brooks is survived by his widow, Florence (Rowe) Brooks, and by two children, Carol and Noel.

FRANK WILSON BUNN (1866–1933)

Frank Wilson Bunn, manager of plants for the John A. Roebling Sons Co., Trenton, N.J., died of pneumonia at Trenton on May 1, 1933. He had been with the Roebling company since 1905.

Mr. Bunn was born at Sterling, Ill., on April 8, 1866, the son of Martin A. and Sarah (McCune) Bunn. He attended the Sterling High School and worked for a year in the shop of the Eureka Manufacturing Company, Rock Falls, Ill., then spent three years, 1883–1886, at the University of Illinois, Urbana, taking the mechanical engineering course. Upon the completion of his college work he was employed for several months at Sterling in the shop of the Williams & Orton Maufacturing Co., and from November, 1886, to February, 1888, represented that company in New York and was in charge of gas engines for the Zell Engineering Company.

During the remainder of 1888 and first half of the following year Mr. Bunn was connected with the Southern Pacific Railroad Company, in the shops at Sacramento, Calif., and making fuel tests on the road. From then until the first of 1892 he spent part time with Robinson & Burr, Champaign, Ill., in shop work, and served as superintendent of the Sterling Gas & Electric Light Co.

In 1892 Mr. Bunn became Chicago representative of the Southwark Foundry & Machine Co. of Philadelphia, Pa. Three years later he was transferred to Philadelphia as sales engineer for the company, and served in that capacity and as assistant secretary until

he became associated with John A. Roebling Sons Co. as engineering assistant to the president in 1905. He was made chief mechanical engineer of the company in 1917, chief engineer two years later, and since 1929 had been manager of plants.

Mr. Bunn became a member of the A.S.M.E. in 1894. He was a Mason and Presbyterian. His wife, Caroline (Glenn) Bunn, whom he married in 1889, died in 1931.

FRANK GEORGE CARPENTER (1879–1933)

Frank George Carpenter, who died on April 11, 1933, from injuries sustained in a fall at Muncie, Ind., was born at Orion, Mich., on February 17, 1879, the son of Charles Ira and Eliza J. (Coryell) Carpenter. He was graduated from the Michigan State College of Agriculture and Applied Science, East Lansing, in 1902, and spent the next eight years with Swift & Co., Chicago, as draftsman, assistant to the master mechanic and civil engineer, and superintendent of construction of packing plants at St. Joseph, Mo., San Francisco, Calif., Hutchinson, Kan., Portland, Ore., and other cities.

In 1911 Mr. Carpenter entered upon similar work for P. Burns & Co., Calgary, Canada, for which he became chief engineer and master mechanic and later plant superintendent and manager of construction and of the engineering department. He designed and built packing plants in various parts of Western Canada. In 1920 he was general superintendent of the Midland Packing Company Stock Yards, Sioux City, Ia., and the following year superintendent of Roberts & Oakes, Chicago. He then established a packing business, the F. G. Carpenter Co., at Lansing, Mich., which he operated until 1929. He had engaged in consulting work since that time.

Mr. Carpenter became a member of the A.S.M.E. in 1918. He also belonged to the Masonic fraternity, in which he had attained the rank of a Shriner, and to the Sons of the American Revolution and the Presbyterian Church.

Surviving Mr. Carpenter are his widow, Chloe (Goodrich) Carpenter, whom he married in 1910, and two children, Charles David and Ruth Ann Carpenter.

GEORGE HENRY COBB (1856–1933)

George Henry Cobb, the son of George Washington and Mary M. (Hamm) Cobb, was born at Northampton, Mass., on July 10, 1856. He was graduated from the mechanical engineering course of Lehigh University, Bethlehem, Pa., in 1886, and during the next two years worked for Harrison Bros. & Co., Philadelphia, Pa., as assistant to the chief engineer. During the next year he was chief engineer of the Kalion Chemical Company, Philadelphia.

In 1889 Mr. Cobb went to Elmira, N.Y., to take the position of master mechanic of the New York Division of the National Transit Company. He remained there until 1900, when he transferred to Passaic, N.J., as chief engineer of the N.Y. Transit Co. Subsequently he served the company successively as master mechanic, general manager, superintendent, and vice-president. He was located at Binghamton, N.Y., from 1906 until he retired in 1920.

Mr. Cobb became a member of the A.S.M.E. in 1891, and also belonged to the Masonic fraternity, in which he was a Shriner and Knight Templar. He married Minnie A. Downing in 1887 and is survived by her and by a daughter, Beatrice E. Cobb. He died at his home in Santa Barbara, Calif., on January 21, 1933.

WILLIS EDWARD CORWIN (1900–1933)

Willis Edward Corwin, son of Willis W. and Marguerite E. (Day) Corwin, Bloomfield, N.J., died of tuberculosis at Banning, Calif., on May 31, 1933. Mr. Corwin was born in Newark, N.J., on February 20, 1900, attended the Irvington, N.J., High School, and was graduated from the mechanical engineering course at Stevens Institute of Technology in 1923. From then until ill health forced him to give up his work, he was field engineer and plant inspector for the Western Electric Company, Kearny, N.J., and New York, N.Y.

Mr. Corwin became a junior member of the A.S.M.E. in 1923. He belonged to the Beta Theta fraternity at Stevens and was a member of the Irvington Methodist Church.

HARRY O. CREAGER (1890–1930)

Harry O. Creager was born at Richmond, Ind., on December 1, 1890, the son of Johnothan W. and Hannah (Brown) Creager. He supplemented his high school education with night school courses in higher mathematics, mechanical drafting, and factory management.

From 1913 to 1925 Mr. Creager was employed by the Dayton Engineering Laboratories Company, except that during 1918 he was with the U.S. Army Air corps, stationed at Wilbur Wright Field, at Dayton, Ohio, as expert motor mechanic. Since 1925 he had been with the Delco Products Corporation. At the time of his death on March 10, 1930, Mr. Creager was in charge of the tool supply division of the company.

Mr. Creager became an associate-member of the A.S.M.E. in 1928. His widow, Marie (Himes) Creager, whom he married in 1917, survives him.

THOMAS CUNNINGHAM (1854–1932)

Thomas Cunningham, long identified with the iron industry, died suddenly of heart failure at Davis Island, Fla., on December 27, 1932. He was born on November 23, 1854, at Charlestown, Mass., and attended the public schools and a commercial college there. His parents were Thomas and Sarah (Miller) Cunningham.

In 1876 Mr. Cunningham became an apprentice at the iron and boiler works which his father had established in Charlestown in 1850. Upon his father's death about the year 1881 he succeeded to the control of the business, which he incorporated as The Cunningham Iron Works Company and removed from Charlestown to South Boston, where he built a new and extensive plant. In the capacity of treasurer and general manager of the company Mr. Cunningham prepared estimates and contracts and supervised the design and construction of boilers, standpipes, and general iron work. He retired in 1910, turning the conduct of the business over to his employees.

Mr. Cunningham was very fond of ocean travel. He had been around the world twice and made frequent crossings to Europe, and also spent considerable time on his own boats, owning a yacht, a speedboat, and a launch. He belonged to the New York Yacht Club and Bucks Harbor Yacht Club, Maine. When not traveling he spent the summers at South Brooksville, Me., and the winters in Boston, Mass. He had been in Florida for about a month before his death.

Mr. Cunningham became a member of the A.S.M.E. in 1899. He also belonged to the Algonquin and the Exchange Clubs in Charlestown, and the University Club, Boston, and was a 32d degree member of the Masonic fraternity and a member of the Episcopal Church. He was a director of the Fourth Atlantic Bank, Charlestown, for 15 years, a director of the Atlantic Works, East Boston, and for a number of years a trustee of the Rotch Infant Hospital, Brookline. He was also a member of the executive board of trustees of Tufts College, and received the degree of Master of Arts from that institution in 1905.

Surviving Mr. Cunningham is his widow, Florence (Clapp) Cunningham, whom he married in 1923. His first wife, Lilla (Sargent) Cunningham, died in 1920.

DALLAS WILLIAM DALE (1899–1932)

Dallas William Dale, whose death occurred on November 16, 1932, was born at Toledo, Ohio, on March 19, 1899, the son of Elmer E. and Emma (Strite) Dale. He attended the East High School, Minneapolis, Minn., and during the year after his graduation worked as draftsman on sawmill machinery and gas engines for the Diamond Iron Works, Minneapolis, designer and assistant master mechanic for the Waldorf Paper Products Company, St. Paul, Minn., on special paper box machinery, and designer of tools and fixtures for machining shells for the Munition Plant of the Twin City Forge & Foundry Co., Stillwater, Minn.

After two months in the army he entered the University of Minnesota, in December, 1918, and was graduated in 1924 with a B.S. degree in mechanical engineering. During his vacations he worked for the Diamond Iron Works, and in the Bridge and Building Department of the Milwaukee, St. Paul & S. Ste. Marie Ry.

For two years after he completed his college work he was employed successively as a designer for the National Supply Company and Merrell Manufacturing Company in Toledo and the Cadillac Motor Car Company, Detroit. Since 1927 he had been structural engineer for the City of Detroit.

In addition to the design work which he performed for the companies by which he was employed, Mr. Dale had assisted in the design of the Strite automatic toaster and had worked on a machine for converting flax straw into linen, one for creasing shipping containers, and another for applying paraffin to butter cartons.

Mr. Dale became a junior member of the A.S.M.E. in 1925 and also belonged to the Masonic fraternity and the Lutheran Church. He is survived by his widow, Esther (Hendrickson) Dale, whom he married in 1925.

WILLIAM POSTAN DALLETT (1860–1933)

William Postan Dallett, president and general manager of the W. P. Dallett Co., Philadelphia, Pa., dealers in pumps and other hydraulic machinery and contractors for waterworks, filtration, and sewage-disposal plants, died on March 31, 1933.

Mr. Dallett was born near West Chester, Pa., on December 3, 1860, the son of William H. and Anne Maria (Cooper) Dallett. After attending public schools near his home until he was thirteen he spent six months at the Friends Central Grammar School in Philadelphia and four years at the state normal school at West Chester. He then entered the Worcester Polytechnic Institute, from which he was graduated in 1881 with a B.S. degree in mechanical engineering. He was secretary of his class, in which he ranked third, and served in that capacity for many years. At the fiftieth reunion he represented the class as its president. He served two three-year terms as vice-president of the W.P.I. Alumni Association, and in 1920 was elected a trustee of the Institute. He was re-elected in 1922 and served until 1927. He was also one of the charter members and an early president of the Worcester Tech Club of Philadelphia.

Shortly after his graduation Mr. Dallett engaged in tool and experimental work in the drafting room and shops of Wm. Sellers & Co., Philadelphia. He was there until 1883, and again in 1889 as foreman of the erecting shop. From 1883 to 1889 he was a member of his uncle's firm, Thomas H. Dallett & Co., of Philadelphia, manufacturers of machine tools, dynamos, and motors, and from 1890 to 1893 was manager of the Philadelphia office of the Deane Steam Pump Company. In 1890 he also began to conduct business under his own name, and when the company was incorporated in 1918 he became its president and general manager. Mr. Dallett held a patent for an automatic relief valve for pumps.

Mr. Dallett became a member of the A.S.M.E. in 1888. He was an active member of the Engineers' Club of Philadelphia for more than forty years, serving as director, vice-president, and president. He was also a member of the Pennsylvania Water Works Operators Association, Philadelphia Academy of Fine Arts, the University Club, Army Ordnance Association, and the Masonic fraternity. He was active in the civic affairs of Media, Pa., where he made his home, and served as president of the Media Club.

Surviving Mr. Dallett are his widow, Rebecca R. (Thurber) Dallett, whom he married in 1886, and three daughters, Mildred, Dorothea, and Jean Dallett.

AUGUST DEMANGEON (1863–1932)

August Demangeon, whose death occurred at Portland, Oregon, on September 19, 1932, was born in San Francisco, Calif., on August 23, 1863. After leaving high school he served an apprenticeship as machinist, steam engineer, and mechanical draftsman and also studied drafting and mathematics in night school, in San Francisco. In 1886 he became machinist and steam engineer for the Tacoma (Wash.) Mill Company, where he remained for about a year.

From 1887 to 1903 Mr. Demangeon was identified with two of the largest lumber mills in the Pacific Northwest. He was mechanical superintendent during the first year for the Port Discovery (Wash.) Mill Company and then was employed in the same capacity by the Tacoma Mill Company. In 1908 and 1909 he was general superintendent for the C. A. Smith Lumber & Manufacturing Co., of Marshfield, Ore., the largest lumber manufacturing plant in that part of the country. In all three positions he had full charge of construction, operation, and repairs of buildings and machinery and designed and directed the installation and operation of many mechanical devices and machines for manufacturing lumber.

For four years beginning in 1903 and from 1911 to 1926 Mr. Demangeon was sales engineer for the Allis-Chalmers Manufacturing Company, of Milwaukee, Wis., during the latter period having his headquarters at Portland, Ore. About half of his time was spent in actual sales work and the remainder in designing lumber plants and also equipment and steam and electric power plants for them.

After leaving the Allis-Chalmers company he was connected for several years with the Chas. R. McCormick Lumber Co., being located at St. Helens, Ore., and at Port Ludlow and Port Gamble, Wash.

Mr. Demangeon became a member of the A.S.M.E. in 1925. He is survived by his widow, Ella (Wright) Demangeon, whom he married in 1904.

JOHN OVIATT DEWOLF (1870–1932)

John Oviatt DeWolf, who died of peritonitis on November 17, 1932, was born at Greenfield, Mass., on May 10, 1870, the son of Austin and Frances Orphelia (Oviatt) DeWolf. He prepared for college at the Greenfield High School and at the age of 20 was graduated with an S.B. degree from the Massachusetts Institute of Technology. During the next two years he was assistant instructor there to Professor Peabody in marine engineering.

From 1892 to 1899 Mr. DeWolf was mechanical engineer and assistant superintendent for the Boston Woven Hose & Rubber Co., Cambridgeport, Mass., and was associated with them as consulting engineer until his death.

In 1899 he took charge of the Boston office of W. B. Smith Whaley Company, later becoming a member of the firm. During that time he designed cotton mills in New Bedford and South Carolina. In 1906 the partnership was dissolved and from that time Mr. DeWolf conducted a general consulting engineering practice under the name of John O. DeWolf & Co., with offices in Boston.

During the World War, in addition to his regular practice, he was engineer and mill architect for the Simonds Saw & Steel Co., Fitchburg, Mass., which had plants in different parts of the country, and also for the Cape Ann Tool Company, Rockport, Mass. He continued to serve both these companies as consulting engineer until his death.

Mr. DeWolf became a member of the A.S.M.E. in 1901. He also belonged to the Boston Society of Civil Engineers, Engineering Societies of Boston, Inc., Massachusetts Society of Mayflower Descendants, Massachusetts Chapter, Sons of the American Revolution, Masonic fraternity, Boston Art Club, University Club of Boston, Boston Engineers Club, Technology Club of New York, Fay Club, Fitchburg, Mass., and Tuscarora Club, Lockport, N.Y. He was a member of the Episcopal Church.

Mr. DeWolf was always deeply interested in the Massachusetts Institute of Technology. He was a member of the Alumni Council, representing the Class of 1890 and of the Faculty Club, and frequently lectured before the classes in mechanical engineering.

Mr. DeWolf made several collections of antiques. He added to pieces of old pewter which he had inherited, built up a collection of firearms showing the evolution of mechanical devices, collected old colonial lamps and antique oriental lanterns and wired them for use in his home, and acquired antique brass and copper and ancient coins. He liked working in wood and metals, was an expert photographer, and held a patent for a method of making metallic printing plates.

Mrs. Anna Sprague (Frothingham) DeWolf, whom he married in 1901, survives him.

WILLIAM JOHN MARSHALL DOBSON (1847–1932)

William John Marshall Dobson, a member of the A.S.M.E. since 1881, died at the home of his sister-in-law, Mrs. J. Harding Price, in West Bridgewater, Mass., on April 28, 1932.

Mr. Dobson was born in England on April 23, 1847, and attended school there. His parents were William John Marshall Dobson and Frances Ann (Smallcorn) Dobson. After coming to the United States Mr. Dobson was for ten years (1876–1886) associated with McKesson & Robbins, wholesale druggists and manufacturing chemists, New York, N.Y., for whom he designed and erected an extensive chemical plant and served as superintendent of all chemical appliances.

During the remainder of his life he engaged in consulting work, chiefly in New York and Brooklyn, preparing plans and specifications for factory buildings and power plants and supervising construction and installation. He specialized in ice-making machinery, glue, paint, and enamel factories, chemical works, and wall paper mills. Among the many companies by whom he was employed were the American Druggists Syndicate, Inc., De La Vergne Machine Company, ice-making machinery, and G. W. Koch & Son, Inc., parquet flooring, all of New York, the Standard Wall Paper Company, Hudson Falls, N.Y., and J. H. White Manufacturing Co., brass founders, Brooklyn. He made a number of inventions, among them an auger guide, a belt tension indicator, and a drying system for wall paper and skins.

Mr. Dobson married Sarah Jane MacDonald in 1875 and is survived by her and by a son, William S. Dobson, of Red Lodge, Mont., with whom he spent considerable time during the latter years of his life. He was a member of the Episcopal Church.

HERMAN DOCK (1860–1933)

Herman Dock, whose death occurred on April 6, 1933, was born on March 29, 1860, in Philadelphia, Pa. He had four years' experience in mechanical drafting and seven years' in machine shop, and was associated for about five years with a leading British engineer. He then became a director in the Schlichter Jute Cordage Company of Philadelphia, with which he remained until 1892. He next established a Philadelphia office for mechanical and experimental engineer-

ing and continued in that work until 1907, the latter part of the time at Wyncote, Pa.

From 1907 to 1910 Mr. Dock was chief engineer for the Nichols & Langworthy Machine Co., New York, and the Safety Steam Power Company, Hope Valley, R.I. He then located at Westerly, R.I., where he was mechanical engineer for the U.S. Motor Co. until 1917. During the next two years he was connected with the American Rotary Power Company, in Boston, as chief engineer. He returned to Philadelphia in 1919 and the following year went to New York. Since 1925 he had resided at Guadalajara, Jalisco, Mexico.

Mr. Dock became a member of the A.S.M.E. in 1889. He also belonged to The Society of Naval Architects and Marine Engineers.

JOSEPH FABIUS DORFNER (1887–1933)

Joseph Fabius Dorfner was born at Cracow, Poland, on July 1, 1887, of Austrian parentage. After completing his high school work he entered the Polytechnic University at Vienna, from which he received his M.E. degree in 1912, after four years of practical experience at the Daimler Works, and for the Wiener Neu Stadt and Vienna Municipal Railway. He then came to the United States and was employed for two years in efficiency work and the design department of the Eiseman Magneto Company, Brooklyn, N.Y. From 1914 to 1919 he was engaged in similar work as shop foreman for the American Machine & Foundry Co., Brooklyn. During the next two years he was professor of machine design and mechanical drafting at the Newark (N.J.) Technical School. He then became associated with C. B. Comstock, engineer and architect, New York, in the design and supervision of power plants and mechanical bakery equipment for the Ward Baking Company at Ampere, N.J., and other plants in Cleveland and Youngstown, Ohio.

In 1923 Mr. Dorfner accepted the position of assistant engineer with the Electric Bond & Share Co., New York, with which he remained two years, designing steam electric stations in Houston, Texas, Memphis, Tenn., and Miami, Fla. For a short time after that he was appraisal engineer for the Murrie Company, New York, and then he established his own office in New York as a consulting engineer for steam and electric power plants. Among the companies which he served were the New York Edison Company and the Philadelphia Electric Company. Ill health made it necessary for him to give up active work several years prior to his death, which occurred at Nashville, Tenn., on January 8, 1933. He is survived by his widow, Mildred (Allen) Dorfner, whom he married in 1920.

Mr. Dorfner became a member of the A.S.M.E. in 1925. He also belonged to the Brooklyn Engineers' Club.

WYLLYS EDMUND DOWD, JR. (1878–1933)

Wyllys Edmund Dowd, Jr., vice-president and director of the Foster Wheeler Corporation, New York, N.Y., died on August 14, 1933, at his home in Greenwich, Conn.

Mr. Dowd was born in New York on October 13, 1878, the son of Wyllys E. and F. Isabel (Chesebrough) Dowd. He was graduated from the Sheffield Scientific School of Yale University in 1900 with the degree of Bachelor of Philosophy in the Department of Mechanical Engineering.

After his graduation Mr. Dowd joined the staff of the chief engineer of the Manhattan Elevated Railway Company, with which he was connected until the middle of 1902. He was then employed as mechanical engineer for the East Jersey Pipe Company, of Paterson, N.J., for whom he was in charge in 1903 of the erection of a complete plant where lock bar pipe was first fabricated in this country. In 1903–1905 he held various engineering positions with the Crane Company in Chicago and in Bridgeport, Conn., becoming assistant to the general superintendent.

In 1906 Mr. Dowd became associated with the Power Specialty Company of New York as manager of the Philadelphia office. He later became a director and secretary of the company. In 1927 the Power Specialty Company and other interests combined to form the Foster Wheeler Corporation, and Mr. Dowd became a director and a vice-president of the new corporation.

During the World War Mr. Dowd was captain of engineers.

Mr. Dowd became a member of the A.S.M.E. in 1911 and also belonged to the Society of Naval Architects and Marine Engineers, the Society for the Promotion of Engineering Education, and a number of clubs, including the Union Club, Down Town Association, and Racquet and Tennis Club. He was president of the Yale Engineering Association and the Round Hill Club Stables Company, secretary of the Woodstock Hotel Company of New York, and a director of the Greenwich Riding Association. He was a supporting member of the American Museum of Natural History, Metropolitan Museum of Art, New York Zoological Society, and New York Botanical Gardens.

In 1920 Mr. Dowd married Mrs. Anna Clement Knowles, daughter of a former governor of Vermont, Percival W. Clement. Mr. Dowd is survived by her and by two children, Clement and Nancy Chesebrough Dowd.

PAUL ANDREW DRATZ (1877–1932)

Paul Andrew Dratz, who died at Geneva, Ill., on October 26, 1932, following an abdominal operation, was born at Muskegon, Mich., on August 26, 1877, the son of John Adam and Johanna Christine (Haisch) Dratz. He prepared for college at the Muskegon High School. After receiving a B.S. degree in mechanical engineering from the University of Michigan in 1900 he was in charge of electrical construction work for the J. A. Erner Construction Co., Columbus, Ohio. The following year he began an association with the Whiting Foundry Equipment Company, Harvey, Ill., which continued until 1918. He was successively draftsman, inspection and erection engineer, estimator and salesman, and Chicago representative, in the latter position handling both consulting and sales work.

Since 1918 Mr. Dratz had been a partner in the firm of Clement A. Hardy & Co., Chicago, consulting and construction engineers and specialists in foundry engineering, and since 1925 he had also been connected with H. M. Byllesby & Co., Chicago, Ill. From 1921 to 1925 he was a director of the First National Bank of Harvey, Ill.

Mr. Dratz became a member of the A.S.M.E. in 1918 and also belonged to the Illinois Athletic Club, Glen Oak Country Club, Elks, and the Masonic fraternity. He is survived by his widow, Grace M. (Lyon) Dratz, whom he married in 1923.

CHARLES REGINALD ELAM (1881–1933)

Charles Reginald Elam was born at South Hill, Va., on January 31, 1881, the son of Thomas David and Linia (Ogburn) Elam. He attended the Chase City Academy for two years and the Virginia Polytechnic Institute for a year. From 1900 to 1905 he was chief draftsman and machinist in the engineer's office of the Norfolk Division of the Southern Railway Company. He then carried on machine shop work in his own name for five years. In 1910 he became vice-president of the Thomas Hardware Company, Suffolk, Va., and until 1915 was engaged in designing and supervising the construction of industrial and power plant equipment. He severed that connection to take a similar office with the Virginia Machinery & Well Co., in Richmond, directing the design and installation of water works and power plant equipment. In 1924 he became mechanical and sales engineer for the Virginia Engineering Company, Richmond. At the time of his death on February 25, 1933, he was in business for himself under the firm name of Chas. R. Elam, Engineer, specializing in power plant and water supply equipment. He was also Richmond representative of the Pardee Engineering Company, The Anthony Company, Maris Bros., Inc., Anderson Engine & Foundry Co., Schutte & Koerting Co., Goulds Pumps, Inc., Westco Pump Corporation, and Waterous Company.

Mr. Elam became an associate member of the A.S.M.E. in 1923. He is survived by his widow, Ella (Ogburn) Elam, whom he married in 1920, and by two children, Charles R. Elam, Jr., and Frances H. Elam.

ADOLPHE ST. ARMANT FAIRBANKS (1878–1932)

Adolphe St. Armant Fairbanks, whose death occurred on August 15, 1932, at Portland, Ore., where he had been located for several years, was born in Cleveland, Ohio, on March 18, 1878. His education was secured in the public schools of Brooklyn, N.Y., and at Pratt Institute, from which he was graduated in 1896 after three years of engineering studies.

For about a year after his graduation Mr. Fairbanks worked as an assistant in a chemical laboratory in New York. He then took a position with the Lawrentide Pulp Company as assistant superintendent of the construction of a paper mill at Grand Mère, Quebec, and a considerable part of his work from then on had to do with the design and construction of paper and pulp mills. Among the companies in this field for which he worked were the Central Paper Company, Muskegon, Mich., Union Bag & Paper Co. and Paper Products Company, Chicago, Ill., Sturgeon Falls Pulp Company, Ontario, Canada, West Virginia Pulp Products Company, Covington, Va., Paper Products Company, Baltimore, Md., Hawley Pulp & Paper Co., Oregon City, Ore., and Ranier Pulp & Paper Co., Shelton, Wash.

Mr. Fairbanks also specialized in the design and construction of steam and hydroelectric power plants, particularly during recent

years. In 1929 and 1930 he worked with the chief engineer of the Pacific Power & Light Co. on the design of a hydroelectric plant for Joseph, Ore., assisted in the design of a 35,000-kw addition to the steam generating plant of the Portland Electric Power Company, directed the design of a 6,000-kw stand-by plant for the City of Eugene, Ore., and served on the Eugene Water Board, supervising the construction and operation of a municipal steam plant. Earlier he had designed additional equipment for a municipal steam power plant for the village of Hinsdale, Ill., designed and built a central station for the Southern Illinois Light & Power Co. at Collinsville, Ill., and assisted in power plant design for Westinghouse, Church, Kerr & Co., New York, N.Y.

Among other public utility projects, Mr. Fairbanks was assistant engineer for a street lighting development for Oak Park, Ill., made appraisals for the Oregon-Washington Water Service Company, Portland, and designed and built a water pumping, softening, and filtration plant and an ice plant for Hinsdale, Ill.

Mr. Fairbanks was interested for a time in the Yaquina Fibre Company in Portland, made studies for a large insulating board mill there, and prepared designs for the Fir-Tex Insulating Board Company at St. Helens, Ore.

As a consulting engineer he served many other companies in the fields in which he specialized.

In 1915 and 1916 Mr. Fairbanks was engaged in the design and construction of two smokeless powder plants, one for the Union Powder Corporation at Parlin, N.J., and the other for the Western Cartridge Company at Springfield, Ill. He entered the Officer's Training Camp at Fort Sheridan in May, 1917, secured a commission as a captain in the Engineers Officers Reserve Corps, and served a year in France. Upon his return to the United States he took an examination for the regular army and was made a captain in the Corps of Engineers. He was sent to Ellington Field, Texas, to construct a storm drainage system, after which he was stationed for a year at New Orleans, working on coast defenses and other engineering assignments. He was then transferred to Camp Pike, Ark., as adjutant in the 6th Regiment of Engineers, and subsequently spent two years at Fort Lewis, Wash., and one year at the Army Engineers School, after which he was transferred to the Coast Artillery Corps. He resigned from the army in December, 1924.

Mr. Fairbanks was elected to junior membership in the A.S.M.E. in 1899. Five years later he was advanced to the grade of associate. He dropped his membership in 1911, but joined again, in 1930, as a member in full.

Mr. Fairbanks is survived by his widow, Edith M. Fairbanks, and by three children, John and Pauline Fairbanks, of Portland, and Charles M. Fairbanks, New York.

ARTHUR FALKENAU (1856–1933)

Arthur Falkenau, who died at Stamford, Conn., on June 10, 1933, was born in New York, N.Y., on April 13, 1856, the son of Moritz and Theresa (Bruckman) Falkenau. He attended the College of the City of New York from 1870 to 1872 and during the next three years served an apprenticeship as a machinist at the DeLamater Iron Works and attended Cooper Institute. He then went to Cornell University, from which he was graduated with a B.M.E. degree in 1878; he secured his M.E. degree there in 1898.

After his graduation in 1878 Mr. Falkenau spent two years in drafting work for the Brooklyn Elevated Railway, New York, and Wilson Bros. & Co., Wm. Sellers & Co., and I. P. Morris Co., Philadelphia, Pa. In 1880 he opened an office at Leadville, Colo., for consulting mining engineering. The following year he established a machine shop and foundry which he operated until 1884, when poor health forced him to lease the works and leave Leadville. In 1885 he took a position as assistant engineer with the Dickson Manufacturing Company, Scranton, Pa., soon being made chief engineer. He did not remain in Scranton long, however, but in 1887 went to Philadelphia, where he established an office and shop as mechanical engineer and manufacturer of machinery. He continued in this work until 1902, from 1896 to 1899 also being president of the Falkenau Engineering Company, electrical contractors.

In 1902 Mr. Falkenau helped to establish the Falkenau-Sinclair Machine Company, in Philadelphia, and served as its president until the company was dissolved five years later. After engaging in industrial consulting engineering for a short time, and serving as secretary and treasurer of the Reliance Steel Foundry Company at Delawanna, N.J., for several years, he became vice-president, in 1910, of the Hooper-Falkenau Engineering Company, industrial engineers, New York. He continued in this office until his retirement in 1920.

The first pneumatic postal system in the United States was built under the direction of Mr. Falkenau, and described by him in an article published in the Journal of the Engineers Club of Philadelphia in 1895. He designed and manufactured mine exploring apparatus, the forerunner of the modern gas mask, in 1880. Among other important works he designed and built an 800,000-lb chain testing machine for the Boston Navy Yard. He served in the Tank and Tractor Section of the U.S. Ordnance Department in 1918 and 1919.

Mr. Falkenau became a member of the A.S.M.E. in 1886. He was a member of the Committee on Science and Arts of The Franklin Institute; served as president of the Engineers Club of Philadelphia in 1895 and of the Metal Manufacturers Association in 1904; was a councilor of the National Metal Trades Association; and belonged to the American Society for Testing Materials, the American Geographic Society, and Tau Beta Pi fraternity. He was a former member of the Manufacturers, Aronamink Golf and City Clubs of Philadelphia, and the Machinery Club, New York.

Since his retirement from business Mr. Falkenau had resided in New York, Katonah, and Pelham, N.Y., and Stamford, Conn. He was a violinist and student of botany, particularly the wild flowers of New England and Middle Atlantic states. His wife, Emily T. (Eidlitz) Falkenau, died in 1927. He is survived by his son, Robert M. Falkenau, of New Rochelle, N.Y.

FRED WILLIAM FISCHER (1886–1932)

Fred William Fischer, general manager of the Fischer Machinery Company, Knoxville, Tenn., died on September 7, 1932, of carbon monoxide gas poisoning, being overcome while working on his automobile.

Mr. Fischer was born in Knoxville on April 7, 1886, the son of Barbara Ursula Fischer and William Victor Fischer. After completing his high-school education he served a five-year apprenticeship in the mechanical department of the United States Navy, at Washington, D.C., receiving honorable discharge in 1907. He married Miss Hazel Adell Levy in 1908, during which year he was sales and erecting engineer for the Keystone Lubricating Company, Philadelphia. In 1909 he was lubricating engineer for the Tennessee Oil Company, Knoxville. During the next two years he was in charge of maintenance work for the Evans Marble Company, Knoxville, in the capacity of assistant chief engineer and master mechanic. While in that position he took courses in steam-electrical engineering through the International Correspondence Schools and subsequently studied power plant economy at the University of Tennessee.

From 1911 to 1918 he was chief engineer and master mechanic for the Standard Knitting Mills, Knoxville, supervising the operation of power plants and mechanical equipment and the design and erection of all plant machinery. He also did considerable consulting work along combustion lines. In 1919 and 1920 he was in Jacksonville, Tenn., part of the time serving as boiler plant supervisor for the Du Pont Engineering Company and later as chief engineer of the Old Hickory Powder Plant. From there he went to Denver, Colo., on consulting work, becoming owner there of a metropolitan garage.

In 1924 Mr. Fischer became president and general manager of the Tennessee Tool Works, Knoxville, with which he remained for four years. He was next general superintendent for the W. J. Savage Co., Knoxville, and general superintendent of the Sterling Wood Products Company, Knoxville. After engaging in patent development work for some time, he became president and general manager of the Fischer Machinery Company, the position he held at the time of his death.

Mr. Fischer became a member of the A.S.M.E. in 1918. He was also a Shriner, 32d degree, a member of the St. John's Lutheran Church, and member of several other engineering societies.

He is survived by his children, Fred W. Fischer, Jr., Shirley Mae Fischer, and Alva Louis Fischer, and by his father, three brothers, and four sisters.

JOSEPH (JACK) EMERSON FOSTER (1898–1932)

Joseph (Jack) Emerson Foster was born on September 4, 1898, at Slater, Mo., the son of Jasper Peter and Harriet Foster. After completing his high school education he studied for a time at the University of Illinois. He enlisted in the Air Corps during the World War and afterward spent several years as a mechanical engineer in sales and design work for different companies, including the K. C. Oxygen Gas Co. and Hugh L. Thompson Engineering Co., Kansas City, Mo., Wharton Motors, Dallas, Tex., and National Steam Automotive Company, St. Louis, Mo. From 1928 to 1930 he was chief engineer for the American Eagle Aircraft Corporation, Kansas City, and subsequently held that position with the Cross-Foster Aircraft Company, Kansas City. He patented the wing and fuselage construction of the Cross-Foster all-metal airplane.

During the latter part of 1931 and early 1932 Mr. Foster was with

the Foster Aircraft Company, Alliance, Ohio, and from the first of March until his death on May 18, 1932, at Saginaw, Mich., as the result of an automobile accident, was engineer for the Dow Chemical Company, Midland, Mich., working on an engineering handbook on the use of Dow metal for aircraft.

Mr. Foster became an associate member of the A.S.M.E. in 1929. He also belonged to the Society of Automotive Engineers. He was affiliated with the Baptist Church. He is survived by his widow, Thelma (Shaw) Foster, whom he married in 1930, and by one child, Lee Mast Foster.

A. WARREN FRANCE (1868–1933)

A. Warren France, president of The France Packing Company, Philadelphia, Pa., died at his home in that city on March 8, 1933, after being in ill health for more than a year. Mr. France, who held a number of patents in the United States and abroad on piston rod packing, especially metallic packing, founded the company in 1898. From 1905 to 1912 he also owned and operated a 200-ton absorption ice plant and a 70-ton compression plant at Atlantic City, N.J.

Mr. France was born in Philadelphia on March 3, 1868. After leaving school he spent four years at sea as an oiler, water tender, and engineer, and from 1888 to 1891 was engaged in construction work with the Union Iron Works, San Francisco, and E. Pallis Co., Milwaukee. He next went to Fitchburg, Mass., where he was erecting engineer for the Fitchburg Steam Engine Company for three years. For an equal period, prior to the establishment of his own company, he was chief engineer of the Suburban Electric Company, Philadelphia.

Mr. France became a member of the A.S.M.E. in 1921.

J. PORTER FREEMAN (1865–1933)

J. Porter Freeman, who died at his home in Yonkers, N.Y., on May 26, 1933, was born on June 8, 1865, at Lancaster, Mass., the son of Joshua and Jane (Wellington) Freeman. He attended school at Lancaster and Worcester, Mass., and served an apprenticeship with the Bigelow Carpet Company, Clinton, Mass. He rose to the position of chief engineer of this company, with which he was connected until 1893. He then went to Yonkers as consulting and construction engineer for Alexander Smith & Sons Carpet Co., and had held that position continuously until his retirement at the beginning of 1933.

Mr. Freeman became a member of the A.S.M.E. in 1908 and was a very active member of the Masonic fraternity. He was fond of music and was organist for the Clinton Lodge of Odd Fellows for a number of years. He had served as a member of the Board of Education of the Saunders Trade School at Yonkers.

Surviving Mr. Freeman are his widow, Ethel (Hatfield) Freeman, whom he married in 1899, and a son, John Porter Freeman.

ROSS LAUDER FRYER (1881–1933)

Ross Lauder Fryer, consulting engineer, Washington, D.C., died on January 3, 1933. Mr. Fryer had been a member of the A.S.M.E. since 1918 and also belonged to the American Society of Marine Draftsman, American Society of Naval Engineers, and Washington Society of Engineers.

He was born in New York, N.Y., on April 24, 1881. He served an apprenticeship in the machine shop of the Arrow Steamship Company, Baltimore, Md., attended the Washington Business High School, and studied three years at George Washington University, Washington, D.C. At the age of twenty he secured a position as secretary to the Quartermaster General, U.S. War Department, Washington, and continued in that work until 1906. In 1905 he began the study of naval architecture and marine engineering under private tutorship and the following year became an ensign in the U.S. Naval Militia, District of Columbia, and subsequently worked up through the grades of lieutenant and assistant engineer to the rank of senior engineer officer. He remained in the Militia until 1913. From 1908 to 1915 he was also assistant marine engineer to the Quartermaster General, in responsible charge of the design, construction, and operation of a fleet of Army transports, harbor boats, and lighters. Since 1915 he had practised consulting engineering.

Mr. Fryer was one of the pioneers in applying the Diesel principle of propulsion, and in 1916 took over the representation of the Winton Engine Corporation before the United States Government, which position he held until the time of his death. Through his faith, enthusiasm, and efforts, Diesel engines have been installed and are operating most successfully in the U.S. Coast Guard, Lighthouse Bureau, Engineer Corps of the Army, and other departments of the Government.

CHARLES RALEIGH GABRIEL (1866–1933)

Charles Raleigh Gabriel, consulting and designing engineer for the E. W. Bliss Co., Brooklyn, N.Y., died on February 24, 1933. He had been a member of the A.S.M.E. since 1901.

Mr. Gabriel was born on July 18, 1866, at Seymour, Conn. He attended public school in Bridgeport, Conn., and learned the machinist's and toolmaker's trade at the Bridgeport Brass Company. He worked as a draftsman for W. F. Durfee, of Bridgeport, for about a year and then spent nine months in the employ of the Bridgeport Gun Implement Company, designing and constructing tools.

In January, 1887, Mr. Gabriel began to work for the Union Metallic Cartridge Company, at Bridgeport, constructing automatic machinery and tools. After three years there he spent part of a year in design work for the Baxter Brothers, electricians, Brooklyn, in connection with the manufacture of graphophone motors. From 1890 to 1895 he was connected with the H. L. Judd Co. of New York, designing and supervising the construction of special automatic machinery for brass work.

The Brown & Sharpe Manufacturing Co., Providence, R.I., engaged Mr. Gabriel in 1895 to design special machinery and tools. He remained with that company until the middle of 1901, when he became mechanical expert for the Gorham Manufacturing Company, silversmiths, Providence, for which he designed and supervised the construction of special machinery and tools for the silver business in all its branches and made a study for the improvement of manufacturing methods. He was connected with the Gorham company for two years, and he spent equal periods after that as chief engineer of the Dow Composing Machine Company, New York, and chief draftsman and designer of the Cincinnati Milling Machine Company, Cincinnati. He had been with the Bliss company since 1907.

HENRY DECATUR GORDON (1848–1932)

Henry Decatur Gordon, who died at his home in East Orange, N.J., on October 14, 1932, was a former consulting engineer to Jenkins Bros., manufacturers of Jenkins valves, New York, N.Y. He retired in 1923.

Mr. Gordon was born at Waltham, Mass., on August 20, 1848. His childhood was spent in Rochester, N.Y., and after completing his public school education there he served an apprenticeship as a machinist with the N.Y. Central Railroad, at the same time studying drafting. Subsequently he was employed by the Philadelphia, Washington & Baltimore R.R. and Pennsylvania Railroad, being in charge of the latter's shops at Altoona from 1889 to 1895. His association with the Jenkins firm dated from 1895.

Mr. Gordon became a member of the A.S.M.E. in 1896.

ROBERT STANISLAUS GRIFFIN (1857–1933)

Rear-Admiral Robert Stanislaus Griffin, U.S.N., who retired from service in 1921, died at the Naval Hospital in Washington, D.C., on February 21, 1933. Admiral Griffin's retirement followed a period of eight years as engineer-in-chief of the Navy and chief of the Bureau of Engineering, during which the scope of the Bureau's work was very wide and the amount enormous.

Admiral Griffin was born in Fredericksburg, Va., on September 27, 1857, and received his early education there. He was graduated from the United States Naval Academy, Annapolis, Md., in 1878, one of the five honor men of the class. His first cruises were on the *Alliance* and *Quinnebaug*, in European waters, and the *Tennessee*. He attained the rank of assistant engineer in 1880. From 1885 to 1890 he was on shore duty, with the Naval Advisory Board, Office of Naval Intelligence, and Bureau of Steam Engineering, where he served under George W. Melville, the famous engineer and Arctic explorer.

In 1890 he was returned to sea duty and for two years was on the *Philadelphia*. He next spent a few months as inspector of machinery on the *Bancroft*. From 1893 to 1897 he was again with the Bureau and subsequently on the *Vicksburg*. He was made chief engineer in 1898 and in that capacity served on the *Mayflower* during the Spanish-American War, on blockade duty north of Cuba. His promotion to the rank of lieutenant came in 1899, when the Line and the Engineers were combined. In 1901-1902 he was with the *Illinois*, then he spent a year each on the *Chicago* and *Iowa*. He terminated his service at sea as fleet engineer of the North Atlantic Fleet in 1904-1905. He was promoted to Commander in 1906, Captain in 1910, and Rear-Admiral in 1916.

Admiral Griffin was with the Bureau of Steam Engineering constantly from 1905 until his retirement. He was appointed assistant to Admiral Cone, the chief, in 1908. He was commissioned engineer-in-chief in 1913, and was reappointed in 1917. With other

Bureau chiefs he accompanied Secretary Daniels to Europe in the spring of 1919 to inspect naval materiel and personnel.

For several years after his retirement Admiral Griffin engaged in consulting work in New York with Commander W. L. Cathcart. He was also actively interested in the work of the Navy Relief Association.

In recognition of his achievements during the World War Admiral Griffin was made a Commander of the French Legion of Honor and received the Distinguished Service Medal of the United States. He was elected to honorary membership in the A.S.M.E. in 1920 and held the degrees of Doctor of Science from Columbia University and Doctor of Engineering from the Stevens Institute of Technology. He was a member of the Society of Naval Architects and Marine Engineers and an honorary member and past-president of the American Society of Naval Engineers, of whose Journal he was the first editor.

In the May, 1933, issue of that Journal, pages 207–224, will be found an appreciation of his life and work by Walter M. McFarland, a naval engineer from 1875 to 1899, who succeeded him as editor of the Journal, had been associated with him in the Navy, and was an intimate friend for more than fifty years.

Mr. McFarland has prepared for inclusion in this memorial notice the following paragraphs referring particularly to Admiral Griffin's connection with the Bureau of Engineering.

"Admiral Griffin's eight years as engineer-in-chief of the Navy and chief of the Bureau of Engineering, from 1913 to 1921, covered the period of the World War with the great increase in materiel and the responsibility of maintaining in a state of constant readiness and efficiency the machinery of the greatest Fleet ever possessed by the United States.

"The variety of the Bureau's work was great and the amount immense. A very brief summary is as follows: A great increase in the number of vessels and in the variety of the machinery equipment of the Fleet; the equipment and maintenance abroad of repair ships and repair bases; the construction of destroyers, submarines, subchasers, 'Eagles,' mine-layers, and tugs; the development of the devices for the detection of submarines; the creation of an organization for handling the power of aircraft; the production of helium in commercial quantities and the establishment of numerous stations for the production of hydrogen for lighter-than-air craft; the enlargement of the machinery equipment of the Navy Yards; the maintenance of the large fleet of Naval Overseas Transport Service; and the great extension of radio in shore stations and the Fleet.

"One of the most important features was the repair and rehabilitation of the machinery of the interned German merchant vessels, which had been wrecked by their crews with the idea that it would have to be rebuilt. Investigation led Admiral Griffin to decide to make the repairs by welding, which brought about a saving of a year's time in the work as well as many millions of dollars; and, of much greater importance, it enabled about 600,000 American soldiers to be transported to Europe a year earlier than had been believed possible.

"The repair ships and repair bases abroad were most important in the sustained efficiency of naval vessels and transports, especially of the overworked destroyers.

"During the War, all radio material work afloat and on shore was directed by the Bureau; this meant the maintenance of about 4000 ships and 210 shore stations. The Bureau designed the high-power station at Bordeaux, the most powerful station in the world, which upon completion was turned over to the French government. There was also constructed a station at Vladivostok for communicating with our troops in Siberia. At the Peace Conference after the war, his assistants were in charge of communications. In Central Europe, then in revolution, wherever a Peace Conference commission was sent, with it went his portable sets and men and officers to operate them.

"Under Admiral Griffin's administration, the horsepower of the Fleet increased from 2,000,000 to 10,600,000 for ships actually completed, and from about 400,000 to nearly 2,500,000 for ships under construction. During this time there were added to the Navy, 12 battleships, 295 destroyers, 95 submarines, 60 'Eagles,' 49 auxiliary vessels, and a large number of mine sweepers, tugs, yachts, and subchasers. The adoption of the electric drive for capital ships (battleships and battle cruisers) is an outstanding event in Griffin's régime. The machinery for six battle cruisers was designed under his direction, the most powerful marine installations in the world. Only two of them, the *Saratoga* and the *Lexington*, were completed and had their trials after his retirement. They developed over 200,000 shp each. Another momentous change was the transformation of the Fleet from coal-burning to oil-burning.

"The office of engineer-in-chief of the Navy requires for complete efficiency much more than professional ability: he must also be a great executive and administrator. In this respect, Griffin was ideal. All the officers who served under him testify to his remarkable ability in anticipating and preparing for necessary work, and the ease with which he bore responsibility. When the tremendous burden due to the World War came upon him, it found him ready and efficient. His name will ever be associated with those other great engineering executives also honorary members of our Society, Haswell, Isherwood, and Melville, who were his predecessors as engineer-in-chief of the Navy in troublous times, and like him made history."

HERBERT CARLTON HALE (1874–1933)

Herbert Carlton Hale died at his home in Washington, D.C., on April 16, 1933, after an illness of about five months. He was district sales manager for the Elliott Company, steam power specialties, Washington, D.C. He was located at the district office in Baltimore for four years before it moved to Washington in 1932.

Mr. Hale was born at Huntington, W.Va., on March 15, 1874. He prepared for college at public schools in Akron, Ohio, and the Oberlin, Ohio, Academy. He received a B.S. degree from the Case School of Applied Science in 1896 and won his M.E. three years later. During this period he secured practical experience in the shops of the Arctic Machine Manufacturing Company, Cleveland, and drafting experience with the Automatic Refrigerator Company, Cleveland, and Webster Camp & Lane Co., Akron.

In 1899 he became chief draftsman of the Webster Camp & Lane Co., and he held this position until the company consolidated with the Wellman-Seaver-Morgan Company of Cleveland in 1903. He transferred to Cleveland as a member of the Engine and Mining Department, where he remained for about a year. He then became chief engineer of the Crown Dryer Company, Cleveland, with which he was associated until 1906. During the next few years he was superintendent and general manager of the Mineral Ridge (Ohio) Manufacturing Company and engaged in contracting and designing in Cleveland.

In 1911 Mr. Hale became district manager for the Ridgway Dynamo & Engine Co., Cleveland. He was later made sales engineer and advertising manager for the company, with which he remained until it was taken over by the Elliott Company in 1926.

Mr. Hale became a member of the A.S.M.E. in 1904. He also belonged to the American Institute of Mining and Metallurgical Engineers and was on the executive board of the Alumni Association of the Case School of Applied Science, and a member of the Case Corporation. He is survived by his widow, Fannette L. Hale, two sons, Clarence H. and Gordon M. Hale, and one daughter, Elizabeth (Hale) Lally.

ROBERT CARL HANDLOSER (1875–1933)

Robert Carl Handloser, manager of the Detroit office of The Motch & Merryweather Machinery Co., died on January 3, 1933, at his home in Detroit after a short illness.

Mr. Handloser was born in Trenton, N.J., on October 26, 1875, the son of Thomas Valentine and Louisa (Herrmann) Handloser. After attending the Trenton grade schools and high school in New York, N.Y., he worked for two years as a machinist for R. H. Wolfe & Co., New York. He then entered Stevens Institute of Technology, Hoboken, N.J., from which he was graduated in 1898 with a degree in mechanical engineering.

For a few months following his graduation Mr. Handloser was engaged as a draftsman in the Switchboard Department of the Western Electric Company, New York, then was assistant to the inspector of machine tools for the Garvin Machine Company, New York, for about six months and draftsman and engineer with the Keuffel & Esser Co., Hoboken, for two months.

In May, 1899, Mr. Handloser entered upon a student course in machinery, spending some time with the Brown & Sharpe Manufacturing Co., The Pond Machine Tool Company, The Niles Tool Works Company, and others. Upon the completion of his course he was sent abroad to sell the products of these companies in Austria and Germany and until the middle of 1901 was manager of the Vienna office.

During the next year he represented the Vienna steel firm of the Bohler brothers in England and subsequently returned to the United States, where he worked for a time in his father's machine shop, designing and operating machines, and sold steel for Houghton & Richards, American agents for the Bohler firm.

From May, 1903, to June, 1905, he was in charge of the New York office of the Dilworth Porter & Co., Ltd., of Pittsburgh, Pa. He then spent some time in the plants of the Niles-Bement-Pond Company, after which he again represented those organizations in Germany and Austria for nearly two years.

Mr. Handloser joined the Motch & Merryweather organization in April, 1907, and from then until his death was in charge of the Detroit office. He was also a director of the company.

He became an associate-member of the A.S.M.E. in 1918 and also belonged to the Detroit Athletic Club and the Gowanie (Detroit) Golf Club. He is survived by his widow, Bessie Gertrude (Vorce) Handloser, whom he married in 1904, and by three sons, Robert, Jr., Thomas V., and Willard B. Handloser, all of Detroit.

HARRISON WASHBURN HAYWARD (1873–1932)

Harrison Washburn Hayward, professor of materials of engineering at the Massachusetts Institute of Technology, died at the Institute Infirmary on October 18, 1932, following a heart attack.

Professor Hayward was born at Dorchester, Mass., on January 1, 1873, the son of Edward S. and Emma (Washburn) Hayward. He attended public schools in Hyde Park, Mass., and the Berkeley School, Boston, and was graduated from the Institute in 1896 with an S.B. degree. He had been on the teaching staff of the Institute since his graduation with the exception of a year (1899-1900) as draftsman at the Universal Loom Works, Readville, Mass. He served as an assistant instructor in the chemical engineering department for two years. He then went into the mechanical engineering department, where he became assistant professor of applied mechanics in 1908. In 1912 he was made associate professor of theoretical and applied mechanics, and in 1920 he was advanced to professor of materials of engineering.

Since 1916 Professor Hayward had been in charge of the Testing Materials Laboratory at the Institute, and he had much to do with the development of this line of work. During the World War he rendered valuable service to the Government in connection with materials, and immediately following the war he accepted a commission as Major in the Reserves.

Professor Hayward was the author of a book entitled "Notes on Calculus," used by the students in the Lowell Institute School, in which he had been a teacher since it was founded in 1903, to provide evening instruction in engineering for those obliged to work during the day. He also completely revised "Materials of Construction, Their Manufacture, and Properties," written originally by A. P. Mills.

Professor Hayward was assistant director of the Institute's Division of Industrial Cooperation and Research and had served a number of industrial firms as a consultant on materials. He had been a member of the A.S.M.E. since 1915 and also belonged to the American Society of Civil Engineers, Boston Society of Civil Engineers, American Society for Testing Materials, the American Concrete Institute, the American Society for Steel Testing, and the Society for the Promotion of Engineering Education. He was a member of Theta Xi fraternity.

Professor Hayward is survived by his widow, Mabel E. (Holmes) Hayward, whom he married in 1902.

EDWIN MUSSER HERR (1860–1932)

Edwin Musser Herr, who for 18 years was president of the Westinghouse Electric and Manufacturing Company, died on December 24, 1932, at his home in New York, N.Y. Because of failing health Mr. Herr relinquished the presidency of the company in June, 1929. At the time of his death, he was vice-chairman of the board of directors of the company, and a director of the American Manufacturers' Export Association, Radio Corporation of America, and Westinghouse Air Brake Company. He had also held office as president of the Westinghouse Lamp Company and as vice-president of the Bryant Electric Company of Bridgeport, Conn., and the R. D. Nuttall Company of Pittsburgh, Pa.

Mr. Herr was born on May 3, 1860, in Lancaster, Pa., the son of Theodore Witmer and Annie (Musser) Herr. When he was twelve years old the family moved to Denver and he began working as a messenger for the Western Union Telegraph Company during summer vacations. Before completing his high school course he had learned to be an operator and after leaving school was operator and station agent for the Kansas Pacific Railway at Deer Trail, Colo. He continued his studies, while employed and after a course at the Boston Latin School he was able to enter Sheffield Scientific School of Yale University in 1881. He helped to pay his expenses by tutoring and was graduated with a Ph.B. degree in 1884.

During vacations, he worked as an apprentice in the Pennsylvania Railroad shops at Altoona, Pa., and after graduation in railroad shops of the Chicago, Milwaukee & St. Paul Ry. in Chicago until offered a job as a draftsman for the Chicago, Burlington & Quincy R.R. at Aurora, Ill. Subsequently he was made engineer of tests, superintendent of telegraphy, and at the age of twenty-nine division superintendent. In 1891 he became division master mechanic of the Chicago, Milwaukee & St. Paul Ry. and in 1892 and 1893 was general superintendent of the Grant Locomotive Works in Chicago. He then took a position as general manager of the Gibbs Electric Company, of Milwaukee, but this company failing soon afterward, he decided to visit Europe and study railroad practice there. He returned in 1895 as assistant superintendent of motive power for the Chicago & Northwestern Ry. A year later he became superintendent of motive power for the Northern Pacific Railway.

In 1898, George Westinghouse, whom Mr. Herr had met twelve years before while conducting a series of air-brake tests on the Burlington Railroad before the Master Car Builders' Association, offered him the position of assistant general manager of the Westinghouse Air Brake Company at Wilmerding, Pa. He became general manager a year later.

In 1905 Mr. Herr was elected first vice-president of the Westinghouse Electric and Manufacturing Company, to the development of which he devoted the remainder of his life. The company was forced into receivership in 1907, and he was one of the receivers and general manager in charge of reorganization. In 1911 he was elected president.

Mr. Herr was always deeply interested in educational matters and devoted much of his time to developing an educational system for the employees of his company. He was a member of the Yale Corporation for twelve years, retiring in 1932. A resolution adopted by the Corporation upon his retirement recorded his services to the University as follows:

"Before coming to the Corporation Mr. Herr had already served the University in many ways, chiefly as first president of the Yale Engineering Association and member of the Alumni Committee on Plan for University Development. He brought to these councils broad experience gained in engineering and in the management of large enterprises in transportation and industry. His advice has been sought constantly, not only in the development of the Sheffield Scientific School, but in all of the University's manifold work. He was a member of the Prudential Committee for seven years, the Committee on Educational Policy for five years, its chairman for 1922-1923. He has been a member of the Committee on Architectural Plan since 1929, of the Committee on Buildings and Grounds since 1925, and its chairman in 1928-1929 and 1931-1932. Above all he has kept a lively interest in young men and especially in those students who support themselves, for their problems were his own in his undergraduate days."

As chairman of the Building Committee of the Yale Corporation Mr. Herr laid the cornerstone of the Sterling Memorial Library. He had served on the advisory board of the *Yale Scientific Magazine*, and he was a contributor to the Fourteenth Edition of the Encyclopaedia Britannica.

Franklin and Marshall College conferred an honorary D.Sc. degree upon Mr. Herr in 1911, and four years later Yale University gave him an honorary M.A. degree. In 1920 he was decorated by the Emperor of Japan with the Order of the Rising Sun in recognition of his interest in training Japanese students at the Westinghouse works and of the company's effective aid in the electrical development of Japan. In 1924 he was chairman of an American Industrial Mission to Mexico.

He was also greatly interested in the graphic arts and had one of the finest collections of Cameron etchings in the United States.

Mr. Herr became a member of the A.S.M.E. in 1891 and served as vice-president for the term 1910–1912. He had also served as president of the American Manufacturers' Export Association and was a member of the American Institute of Electrical Engineers, The Franklin Institute, Pan-American Union, National Industrial Conference Board, American Committee of the International Chamber of Commerce, Historical Society of Pennsylvania, and the Board of the Carnegie Institute of Technology. His clubs included the Chicago Club, the Bankers', Engineers', and University clubs in New York, the Englewood (N.J.) Golf Club, the Congressional Country Club of Washington, and the Duquesne and University clubs at Pittsburgh.

He is survived by his widow, Mrs. Mary (Forsyth) Herr, whom he married in 1900, a sister, Mrs. J. W. Clise, of Altadena, Los Angeles, Calif., and two brothers, Arthur T., Denver, and Herbert T., Philadelphia, Pa.

LOUIS C. HIRSHHEIMER (1879–1933)

Louis C. Hirshheimer, president of L. C. Hirshheimer, Inc., manufacturers of gray iron castings, La Crosse, Wis., since its formation in 1926, died on June 29, 1933. Mr. Hirshheimer was born in La Crosse on March 27, 1879. He attended the University of Wisconsin for two years and Cornell University for a year.

Mr. Hirshheimer was put in charge of the foundry of the La Crosse

Plow Company in 1901, designed and built a new foundry in 1915, and for many years had charge of the power plant and also the purchase and installation of machinery for the company and served as its secretary. He also designed and erected an electric brass foundry for the General American Tank Car Corporation in 1918, and a gray iron foundry for the same company the following year.

Mr. Hirshheimer became a member of the A.S.M.E. in 1921 and was active in the Masonic fraternity.

IRA GOULD HOAGLAND (1877–1933)

Ira Gould Hoagland, secretary and treasurer of the National Automatic Sprinkler Association since he helped found it in 1914, died suddenly at his home in Brooklyn, N.Y. on August 5, 1933.

Mr. Hoagland was born in Brooklyn on January 14, 1877, and attended public schools and Pratt Institute in that city. He entered the insurance field in 1896 as a clerk and rose to the position of an inspector. He was the author of a number of articles on insurance in different industries and from 1912 to 1914 was editor of *Insurance Engineering* and its successor, *Safety Engineering*. Since that time he had devoted himself entirely to the work of the National Automatic Sprinkler Association, a liaison agency between the insurance companies and sprinkler manufacturers and between the insurance companies and the public, its purpose being the advancement of safety and the conservation of life and property. As a writer and speaker Mr. Hoagland helped to bring about wider use of sprinklers, thus lowering insurance rates.

Mr. Hoagland became an Associate of the A.S.M.E. in 1914 and also belonged to the American Water Works Association, New England Water Works Association, National Safety Council, National Fire Protection Association, International Association of Fire Engineers, New England Association of Fire Chiefs, Railway Fire Protection Association, Insurance Society of Philadelphia, Insurance Society of New York, American Trade Association Executives, the Holland Society, and the 12 St., Dutch Reformed Church, Brooklyn.

Surviving Mr. Hoagland are his widow, Louise (Evans) Hoagland, a daughter, Mrs. Dorothy Hammond Foster, and a son, Ira G. Hoagland, Jr.

JOHN M. JENCKES (1866–1933)

John M. Jenckes, who died on January 13, 1933, in Sherbrooke, Quebec, Canada, was born in that city, the son of Sylvester B. Jenckes, on February 17, 1866. He was graduated from the Academic course of the Sherbrooke High School in 1884, and during the next four years served an apprenticeship as a machinist under his father. He then went on the road selling machinery for two years. He became secretary, treasurer, and general manager of The Jenckes Machine Company, Limited, Sherbrooke, in July, 1890, and for nearly thirty years supervised the fabrication and erection of hydraulic plants and penstocks contracted for by that company. Among those served were The Ontario Power Company, Niagara Falls, Laurentide Pulp & Paper Co., Grand Mere, and the City of Sherbrooke. Mr. Jenckes was also general manager of the Engineering & Machine Works of Canada, Ltd., St. Catherines, Ontario, for a time, and during recent years had practised as a consulting engineer, with office in Sherbrooke.

Mr. Jenckes became a member of the A.S.M.E. in 1918. For ten years he was a member of the City Council of Sherbrooke, and he had served as chairman of the Protestant School Board and Protestant Hospital there. He was a member of the Congregational Church and St. George's Club in Sherbrooke.

Surviving Mr. Jenckes are his widow, Helen (Brooks) Jenckes, whom he married in 1892, a son, K. B. Jenckes, and two daughters, Mrs. J. K. Wilson and Mrs. W. L. Bones.

DAVID JOHN JENKINS (1888–1933)

David John Jenkins was born at Angus, Iowa, on November 20, 1888, the son of Samuel and Elizabeth (Woosnum) Jenkins. He secured his early education at the Cazenovia (N.Y.) Seminary and subsequently attended Syracuse University. He was graduated with an S.B. degree in mechanical engineering from the Massachusetts Institute of Technology in 1911.

Prior to the completion of his college work he had spent several years as a machinist for the Cottonwood Coal Company, Stockett, Mont., and erecting machinery for the Link Belt Company for a coal washer at Stockett. After leaving college he was engaged in similar work for the Jeffrey Company, for a washer at Midland, Ark., and then returned to the employ of the Cottonwood Coal Company, as a mining engineer at Windham, Mont. During the next few years he worked consecutively in fuel testing for the Harrisburgh Southern Coal Company, Chicago, Ill., transitman for the Oregon Short Line Railroad in Idaho and Oregon, and assistant manager of coal mines for the Great Northern Railroad, at Lehigh, Mont.

In 1916 Mr. Jenkins became fuel engineer at the United States Bureau of Mines, Pittsburgh. He enlisted as a private in the 27th Engineers in May, 1918, and served with the A.E.F. during the remainder of the World War, winning promotion and special recognition for his services. He returned to the Bureau of Mines for several years, then was connected with the Dallas Power & Light Co., Dallas, Tex. Since 1924 he had been general engineer for the Westinghouse Electric & Manufacturing Co.; he was transferred from the works at East Pittsburgh, Pa., to the Atlanta, Ga., District in January, 1925. His death occurred on June 23, 1932, at Dover, Ohio.

Mr. Jenkins became an associate-member of the A.S.M.E. in 1919. He also belonged to the National Electric Light Association and to the Atlanta Club and East Lake Country Club, both of Atlanta. He is survived by his widow, Margaret (Baker) Jenkins, of Dover, Ohio, whom he married in 1919.

EDWARD EUGENE JOHNSON (1874–1933)

Edward Eugene Johnson, superintendent of power production for The Tucson Gas Electric Light & Power Co., Tucson, Ariz., died on August 15, 1933.

Mr. Johnson was born on October 1, 1874, at Monticello, Ill., the son of Edwin and Alice Rhoda (Gifford) Johnson. After leaving high school he had about two years' experience as a shipwright apprentice, worked in a machine shop for three years, and from 1896 to 1899 fired locomotive and stationary boilers. From then until 1910 he was chief engineer for the United States Light & Traction Co., Denver, Colo., in charge of the operation of all steam equipment, working in Tucson, Ariz., and Albuquerque, New Mexico. He took a similar position with The Tucson Gas Electric Light & Power Co. in 1910 and became superintendent of power production in 1928. He was considered an authority on the operation and maintenance of Diesel oil engines.

Mr. Johnson became an associate-member of the A.S.M.E. in 1929 and also belonged to the National Association of Stationary Engineers. He is survived by his widow, Pauline Anne (Weber) Johnson, whom he married in 1904, and by three children, Pauline Eugenia, Jeanette Ada, and Alice Louise Johnson.

EDWARD CAMPBELL JONES (1861–1933)

Edward Campbell Jones, vice-president of the A.S.M.E. for the term of 1919–1921, died suddenly of heart failure at his home in Palo Alto, Calif., on July 22, 1933. Mr. Jones became a member of the Society in 1901 and was active in the work of the San Francisco Local Section for many years; he served as chairman of that Section in 1918–1919.

Mr. Jones was born in South Boston, Mass., on February 8, 1861, the son of Edward and Hannah F. (Campbell) Jones. He attended Lowell Institute and the Hawes School of Art. He served an apprenticeship under his father at the South Boston Gas Works and gained shop experience at the South Boston Gas Light Company, between the years 1876 and 1884. He was then made assistant superintendent and secretary of the latter company and put in charge of the reconstruction of the South Boston Works. Five years later he was advanced to the position of superintendent of the North End Station of the Boston Gas Light Company, and in 1890 was made assistant engineer of the company, in charge of manufacturing.

In 1891 the San Francisco Gas Light Company employed Mr. Jones as assistant engineer to construct the North Beach Gas Works. He took charge of the gas department of the San Francisco Gas & Electric Co. in 1899, and when the Edison Light & Power Co. merged with the San Francisco company in 1896, Mr. Jones became engineer of both the gas and electric departments of the San Francisco Gas & Electric Co. He was made chief engineer of the company in 1900.

In 1902 Mr. Jones accepted the position of chief engineer of the California Central Gas & Electric Co., and upon the absorption of that company by the California Gas & Electric Corp. two years later, he became chief engineer of all the plants of the Corporation. In 1906, when the San Francisco Gas & Electric Co. was purchased by the Pacific Gas & Electric Co., he was appointed chief engineer, a position which he held until 1920. Since then he had engaged in consulting gas engineering.

Mr. Jones was the inventor of many valuable improvements in gas manufacture and distribution, particularly in the field of purification and in the development of the oil-gas process. His research work for the betterment of the gas industry and his achievements in

this field won him two medals at the San Francisco Exposition in 1915 and also the gold medal of the Pacific Coast Gas Association. He had charge of the restoration of gas in San Francisco after the earthquake in 1906. He lectured in gas engineering at the University of California in 1913 and served the U.S. Bureau of Standards as advisory gas engineer for a time. He was the author of numerous articles relating to heating and gas.

Mr. Jones was a past-president of the Pacific Coast Gas Association and the American Gas Institute, an honorary member of the New England Association of Gas Engineers, and a member of the American Gas Light Association and the Masonic fraternity.

Surviving Mr. Jones are his widow, Florence C. (Harris) Jones, whom he married in 1926, and three children by a former marriage, Edward S., Leon B., and Dwight W. Jones. Their mother, Mary (Stratton) Jones, died in 1922.

FORREST SAMUEL JONES (1882–1933)

Forrest Samuel Jones, works engineer for many years for Chas. Pfizer & Co., Inc., manufacturing chemists, Brooklyn, N.Y., died at his home in that city on July 31, 1933.

Mr. Jones was born on July 24, 1882, at Alpena, Mich. He attended high school in Ann Arbor, Mich., and then spent four years in the machine shops of the Fletcher Paper Company, Alpena. After this practical experience he took a two-year special course in mechanical engineering at the University of Michigan.

From 1906 to 1911 he was employed as test engineer with the Westinghouse Machine Company, Pittsburgh, Pa., working on the development of the steam turbine and directing installations. He was next the vice-president and treasurer of the Pottle-Jones Engineering Co., Inc., New York, for a short time, then construction and operating engineer for the Federal Light & Traction Co., in connection with properties in Colorado, New Mexico, and Arkansas. In 1914 and 1915 he served as consulting engineer to Minneapolis firms and assistant engineer of power plant equipment in the appraisal of public utilities at Washington, D.C.

Mr. Jones became associated with Chas. Pfizer & Co., Inc., late in 1915, in charge of operation, construction, and maintenance, as well as design.

He became a junior member of the A.S.M.E. in 1911 and a member six years later. He belonged to the Michigan Club and the Chemists Club.

JAMES H. JOWETT (1872–1933)

James H. Jowett, executive vice-president and a director of the Ingersoll-Rand Company, New York, N.Y., died at the Flower Hospital in that city on February 16, 1933, after an extended illness. He was also vice-president and a director of the A. S. Cameron Steam Pump Works and a director of the Rathbun-Jones Engineering Corporation.

Mr. Jowett was born on September 15, 1872, in Cleveland, Ohio, and educated in the public schools and Manual Training School there. He then spent a year in marine boiler making at the Cleveland Ship Building Company and about a year and a half each as draftsman for the Variety Iron Works, Cleveland, detailing the Thaw steam shovel, Steel Motor Company, Johnston, Pa., designing street railway motors, and Anderson Dryer Company, working on dryer design.

His connection with the Ingersoll organization began in December, 1897, when he became draftsman for the Ingersoll-Sergeant Drill Company, at Easton, Pa., designing air compressors and compressed air machinery. In the fall of 1900 he engaged in general compressed air engineering and selling of air compressor and air operated machinery with the Ingersoll-Sergeant Drill Company of New York, and the following year became sales manager for the company. His duties included the design and superintendence of construction of compressed-air power plants, among which was one for use in connection with the Pennsylvania Railroad tunnels under the East River, New York. He was elected vice-president of the company in 1917 and executive vice-president in 1930.

Mr. Jowett became a member of the A.S.M.E. in 1907. He also belonged to the American Institute of Mining and Metallurgical Engineers, the Engineers' and Union League Clubs, New York, and several athletic and social clubs in the Metropolitan area. He is survived by his widow, Mrs. Genevieve (Elder) Jowett, whom he married in 1900, and by a daughter, Lucille.

AUGUST KEHM (1868–1932)

August Kehm, president of Kehm Bros. Company, steamfitting contractors of Chicago, Ill., died on December 11, 1932, in that city. His widow and a son survive him.

Mr. Kehm was born in Chicago on October 23, 1868. After leaving grammar school he served an apprenticeship with F. W. Lamb & Co., Chicago, in steam heating and fitting and machine shop operations, and continued with the company as chief estimator and designer until 1893. He then organized the firm of Kehm Bros. & Mertz, later reorganized as Kehm Bros. Company.

Mr. Kehm was a charter member and past-president of the Chicago Master Steamfitters' Association; a member of the board of governors of the American Society of Heating and Ventilating Engineers from 1908 to 1911 and treasurer of its Chicago Chapter for 24 years; and a member of the Chicago Engineers, Hamilton, Lake Shore, and South Shore Clubs. He became an associate of the A.S.M.E. in 1918.

CHESTER SCOTT KEMP (1886–1933)

Chester Scott Kemp was born on March 14, 1886, at Albany, N.Y., and was graduated from the high school there. He learned the machinists' trade and after several years' general shop experience became a co-partner and manager in 1912 of a garage and machine shop.

When the United States entered the World War Mr. Kemp went to the Watervliet Arsenal as an estimator in the Engineering Division. The following year he was made supervisor of tool requirements of the Seacoast Cannon Department and in 1919 foreman of the Planning Section of the Heavy Field Cannon Department.

After the War he worked for a time for the Amphion Piano Player Company, Syracuse, N.Y., in the capacity of production engineer. In 1924 he became vice-president and treasurer of the Jas. N. Kemp Machine Works, Inc., Albany, and in 1931 was made president of the company.

Mr. Kemp became an associate-member of the A.S.M.E. in 1920. His death occurred on June 30, 1933.

EDGAR ULF KETTLE (1868–1933)

Edgar Ulf Kettle, an expert in kiln drying of lumber, vice-president of the Grand Rapids Veneer Works, Grand Rapids, Mich., died on February 17, 1933.

Mr. Kettle was born in Birmingham, England, on March 22, 1868. After leaving school he spent four years in mercantile marine and three years in army service and worked for several years in woodworking plants in England. In 1908 he went to Kearney, Ontario, Canada, where he became manager of the Kearney Timber & Manufacturing Co., Ltd. He supervised the rebuilding of the plant, which had been destroyed by fire, and the installation of new machinery, and directed the operation of the plant and construction work in connection with getting out the timber. When this plant closed Mr. Kettle became manager of the plant of the Niagara Veneer Company, at Parry Sound, Ontario, which also burned, in 1912. Mr. Kettle then came to the United States, and had since been connected with the Grand Rapids Veneer Works. He also carried on consulting work on the design, installation, and operation of dry kilns. He became a naturalized citizen in 1921.

Mr. Kettle was the author of "Practical Kiln Drying," published in 1923. He became an associate-member of the A.S.M.E. in 1918 and a member in full in 1926. He also belonged to the National Association of Power (formerly Stationary) Engineers.

MORRIS KIND (1875–1933)

Morris Kind, president of the Hercules Cement Corporation, Philadelphia, Pa., died on March 17, 1933.

Mr. Kind was born on February 4, 1875, at Philadelphia. He was graduated from the University of Pennsylvania as a civil engineer in 1896 and during the next four years secured practical experience with the Milliken Brothers, New York. He then went to San Francisco where for many years he was general manager of the Pacific Portland Cement Company, Consolidated, directing the design, construction, and operation of cement plants for that company. He had been with the Hercules Cement Corporation since 1916.

Mr. Kind became a member of the A.S.M.E. in 1912.

HOMER LAUGHLIN (1876–1932)

Homer Laughlin, who died on December 27, 1932, after an illness of several weeks, was born at Wellsville, Ohio, on June 17, 1876. He attended the Pennsylvania Military College and Stanford University, receiving an A.B. degree in chemical engineering from the latter in 1896.

For several years following his graduation he was associated with his father in the Homer Laughlin China Company in East Liverpool, Ohio, holding the offices of vice-president and treasurer. The family

then went to Los Angeles, where father and son entered into various real estate projects, including the construction of office, store, and theater buildings. Upon the death of his father in 1913, Mr. Laughlin took over the management of these enterprises, becoming president of the Laughlin Fruit Refiners, Inc., and Homer Laughlin Engineers Corporation. These companies developed, respectively, machinery for separating and preserving lemon and orange juice and peel, and a front wheel drive automobile and the Laughlin tractor. From 1918 to 1920 Mr. Laughlin was also president of the Stability Building Corporation, during which time he erected the million-dollar theater in Los Angeles. He also laid out Laughlin Park, a Los Angeles subdivision, and organized the Grand Central Market.

Mr. Laughlin became an associate of the A.S.M.E. in 1921, was a 32nd degree Mason, and belonged to many clubs, including the California and University Clubs. He is survived by his widow, Mrs. Alta A. Laughlin, and a son, Homer Laughlin III.

AXEL GEORGE LINDBERG (1870–1933)

Axel George Lindberg, general superintendent of the U. S. Steel Foundries, Inc., Denver, Colo., died on June 29, 1933.

Mr. Lindberg was born at Lindsborg, Kan., on May 6, 1870. After leaving high school he attended Betney College and at the age of 21 began work as a foundry journeyman for the Stearns-Rogers Manufacturing Company, then located at Pueblo, Colo. Four years later he was appointed assistant foundry foreman of the plant, in 1899 general foundry and pattern shop overseer and foreman, and in 1911 general superintendent of the entire plant, consisting of machine shop, plate shop, forge shop, pattern shop, carpenter shop, gray iron foundry, and electric steel foundry.

In 1921 the Stearns-Rogers Manufacturing Company merged with four other companies of Colorado, the Denver Engineer Works, Colorado Iron Works, Vulcan Iron Works, and Queen City Foundry Company. The new organization, the General Iron Works, was established in Denver, and Mr. Lindberg was transferred there as general superintendent of the plant. The company joined with the U. S. Steel Foundries, Inc., in 1930.

Mr. Lindberg became a member of the A.S.M.E. in 1929.

VINCENT LINK (1878–1932)

Vincent Link, who died on March 19, 1932, was born at Bellevue, Ohio, on February 26, 1878. He attended high school in Bellevue and then secured a position as draftsman with the Automobile & Mfg. Co. in Geneva, Ohio. He went to Detroit four years later, in 1905, and entered the employ of the Packard Motor Car Company as chief draftsman and truck engineer. From 1910 to 1914 he was chief engineer and assistant general manager of the Universal Motor Truck Company in Detroit and from then until his retirement in 1930 did consulting work, chiefly for the Studebaker Corporation. Among other companies which he served were the Indiana Truck Corporation, Standard Motor Truck Company, Miller Transmission Company, J. C. Wilson Company, and Sewell Cushion Wheel Company.

Mr. Link became a member of the A.S.M.E. in 1918.

FRANK NORMAN MacLEOD (1866–1933)

Frank Norman MacLeod, president and owner of the Abrasive Machine Tool Company, East Providence, R.I., died at his summer home, Harwichport, Mass., on June 25, 1933. Mr. MacLeod was born at East Providence on June 4, 1866, and after attending the public schools of East Providence and Providence, served an apprenticeship at the George H. Corliss Engine Works in Providence. He then worked for about two years in the cotton mill supply business.

In 1890 Mr. MacLeod became connected with the Brown & Sharpe Manufacturing Co., Providence, as junior inspector of machine tools. Four years later he became first assistant to the chief inspector and in 1896 was promoted to the position of chief inspector. He was sent to Nashville, Tenn., in 1897, in connection with the company's exhibit at the Nashville Centennial. From 1898 to 1900 he was general representative of the company in Great Britain, with headquarters in Glasgow, and for the next fifteen years general representative and sales engineer in the United States. He resigned to form his own company in 1916 and was its president until his death.

Mr. MacLeod became a member of the A.S.M.E. in 1912 and was formerly a vice-president of the National Machine Tool Builders' Association. He was a member of the Providence Chamber of Commerce, Rhode Island Country Club, and British Empire Club.

Mr. MacLeod is survived by his widow, Helen B. (Wood) MacLeod, whom he married in 1906, and their daughter, Helen B. W. MacLeod, and by three children by a former marriage, Norman D., Colin G., and Kenneth B. MacLeod. Their mother, Grace L. (Martin) MacLeod, whom Mr. MacLeod married in 1890, died in 1905.

ALFRED FELLOWS MASURY (1882–1933)

Alfred Fellows Masury, vice-president, chief engineer, and director of Mack Trucks, Inc., New York, N.Y., died on April 4, 1933, when the U.S.S. *Akron*, on which he was a guest passenger, crashed at sea. Colonel Masury was very much interested in aerodynamics and aeronautics, particularly in the development of lighter-than-air craft. He was a passenger on the *Graf Zeppelin* on its attempted westward transatlantic flight in May, 1929, and had flown thousands of miles in various types of dirigibles.

Colonel Masury was born in Danvers, Mass., on September 2, 1882, the son of Charles Henry and Evelyn (Fellows) Masury. He prepared for college at the Danvers High School and was graduated from the mechanical engineering course of Brown University, Providence, R.I., in 1904. After going to New York to work, he took further courses at Columbia University.

He spent several summer vacations during college years in drafting and machine shop work and after graduation entered the employ of the General Electric Company, West Lynn, Mass., as a draftsman. For a time he was associated with the Corwin Manufacturing Company, Peabody, Mass., in the development of the "Gasaulec," a self-starting automobile, and subsequently was chief draftsman and assistant engineer for the Vaughn Machine Company, Peabody, manufacturers of leather working machinery, printing presses, engines, trucks, machine tools, giving him a varied experience.

In 1907 Colonel Masury went to New York, where he took the position of engineer for the Hewitt Motor Company, which was later absorbed by the International Motor Company and Mack Trucks, Inc., manufacturers of Mack trucks and buses. He became vice-president, chief engineer, and director of the company, and retained these positions until his death.

From the time that Colonel Masury joined the Hewitt organization he was concerned principally with the design of internal-combustion engines and the electrical problems involved in the manufacture of heavy-duty trucks, such as ignition, lighting, and starting equipment, magnetos, generators, welding and heat-treating by electricity.

During and after the World War he served the Army Ordnance Department in an advisory capacity, chiefly in connection with automotive equipment and transportation. This work led to his promotion to the rank of lieutenant-colonel in the Army Reserve Corps.

Colonel Masury became a junior member of the A.S.M.E. in 1905 and a member in 1913. He had been a vice-president of the Society of Automotive Engineers, chairman of its Ordnance Advisory Committee, and a member of its Military Motor Transport Advisory Committee. He also belonged to the American Society of Civil Engineers, Institute of Automobile Engineers (England), American Society of Municipal Engineers, American Institute of Electrical Engineers, American Association for the Advancement of Science, American Petroleum Institute, Institute of Metals, American Road Builders Association, Permanent International Association of Road Congresses, Society of American Military Engineers, New York State Society of Professional Engineers and Land Surveyors, New York State Board of Licensing for Professional Engineers, United States Naval Institute, and other professional organizations in this country and England.

Boating was Colonel Masury's chief relaxation. He was a commodore of the Lloyds Harbor Club and belonged to the North American Yacht Racing Association, the Cruising Club of America, and the Port Washington and Montauk Yacht Clubs. Other sports, such as golf, boxing, and racing, also interested him. He was a member of the New York Athletic Club and Psi Upsilon fraternity.

Colonel Masury is survived by his mother and by his widow, Edna (Lanphear) Masury.

EDWARD DOLIVER MAYO (1848–1933)

Edward Doliver Mayo, who died in Minneapolis, Minn., on March 30, 1933, was born on December 19, 1848, in Tremont (now Southwest Harbor), Me. His parents were Sarah Newman Doliver and Jacob Shoppy Mayo. After completing his early education he taught school for a year or so in Clearwater, Minn. He later entered Maine State College (now the University of Maine), from which he was graduated in mechanical engineering in 1875.

From 1876 to 1887 Mr. Mayo worked as draftsman and engineer at flour mills and grain elevators and on the construction of ore docks. He was secretary of the Great Western Manufacturing Company during the next year and of the Willford & Northway Manufacturing Co. in 1888–1889. From then until his retirement in 1921 he was located in Minneapolis as chief engineer at first, and

later consulting engineer for the Barnett & Record Co. He also served for a time as consulting engineer to the Barnett McQueen Company, of Ft. William, Ont. He was the inventor of the Mayo grain spout and of other equipment for grain mills and elevators.

Mr. Mayo became a member of the A.S.M.E. in 1914. He also belonged to the Minneapolis Engineer's Club, Professional Men's Club of Minneapolis, Minneapolis Institute of Arts, and the Minnesota State Horticultural Society. He was much interested in gardening and photography. He was a member of the National Society of Puritan Descendants and of the Wesley Methodist Church, in Minneapolis.

Mr. Mayo was twice married. His first wife, Sarah Elizabeth Mayo, whom he married in 1879, died in 1887. The following year he married Minnie Elizabeth Hunt, who died in 1897. He is survived by a son, William D. Mayo, of Des Moines, Iowa, and by two daughters, Mrs. John A. Casey, Bangor, Me., and Mrs. Paul W. Thayer, Minneapolis.

FRANK GOODMAN McCANN (1871–1933)

Frank Goodman McCann, chief of the Heating and Ventilating Division of the Board of Education of the City of New York, died of heart trouble on May 17, 1933, after an illness of several weeks.

Mr. McCann was born in Chicago, Ill., on September 25, 1871, the son of Henry Mackenzie and Fannie (Carkner) McCann. He was graduated from the Chicago Manual Training School (now part of the University of Chicago) in 1890 and spent the next few years in lock and hardware manufacturing, carpentering, and work in a sash and door mill. He then entered the Massachusetts Institute of Technology, from which he was graduated in 1896 with a B.S. degree in mechanical engineering. He worked on heat and ventilation design for the Boston Blower Company during the summer of 1895.

From 1896 to 1899 Mr. McCann was draftsman and construction superintendent for Evans, Almirall & Co., New York, power and heating engineers and contractors. Since then he had been with the Board of Education, for the first three years as mechanical draftsman. He took charge of heating and ventilating in 1902 as assistant to the superintendent of school buildings and prior to the formation of the Heating and Ventilating Division also handled power and light requirements. He originated the use of standard details and of standard specifications for the work of school building and equipment, wrote several articles for *Metal Worker* and other trade journals, and gave lectures on heating and ventilation to evening Building Trades classes in the College of the City of New York.

Mr. McCann became a member of the A.S.M.E. in 1915. He also belonged to the American Society of Heating and Ventilating Engineers, served on its council in 1914 and 1915, and was president of its New York State Chapter in 1912.

Surviving Mr. McCann are his widow, Louise C. (Dralle) McCann, whom he married in 1903, and a son, Forbes Ellard McCann.

JAMES ROBINSON McCLINTOCK (1883–1933)

James Robinson McClintock, of the firm of Fuller & McClintock, New York, N.Y., died on April 11, 1933, of a heart attack following an illness of several weeks with influenza and bronchitis.

Mr. Clintock was born in Malden, Mass., on February 5, 1883, the son of J. Y. and Mary (Robinson) McClintock. He was educated at the University of Rochester and the Massachusetts Institute of Technology, from which he was graduated with a B.S. degree in 1906. His entire professional career was spent with Hering & Fuller, the firm of George W. Fuller, and the firm of Fuller & McClintock, which was organized in 1916. In his earlier practice he was resident engineer on several pumping station and filtration projects at Burlington, Vt., Clarksburg, W.Va., and elsewhere. The major portion of his professional career was devoted to design work for the firm in the fields of water supply, water purification, sewerage and sewage disposal. His recent and best known work was his immediate charge of the design of the Wards Island (N.Y.) sewage treatment project. Prior to that he was in charge of the Toledo office of the firm, directing design work for Toledo, Lima, and several Florida projects. The most of his time from 1921 to 1925 was spent at Kansas City in charge of the design of the new water supply works of that city.

Mr. McClintock became a member of the A.S.M.E. in 1923 and also belonged to the American Society of Civil Engineers, American Institute of Consulting Engineers, American Water Works Association, and the American Public Health Association.

A talent for drawing and design won Mr. McClintock a first prize for the design of a school building when he was still a high school student in Rochester and gave him pleasure in sketching. He was always, as he put it, a student of the "aesthetics of life and living."

EDWARD FURBER MILLER (1866–1933)

Edward Furber Miller, head of the Department of Mechanical Engineering of the Massachusetts Institute of Technology since 1911 and a member of the faculty for 47 years, died on June 12, 1933, at his home in Newton Center, Mass.

Dr. Miller was born in Somerville, Mass., on January 18, 1866, the son of William Gibbs and Sarah (Furber) Miller. He was graduated from M.I.T. in 1886 with an S.B. degree and immediately joined the faculty as an assistant in mechanical engineering. He became an instructor in 1888, assistant professor of steam engineering in 1892, associate professor in 1899, and professor in 1905. On the retirement of Professor Gaetano Lanza in 1911, he became head of the mechanical engineering department.

Dr. Miller was very active and exercised a wise influence in the general affairs of the Institute. He was sent to Europe in 1904, by President Pritchett, to study the methods of engineering education in foreign technical schools. He visited many institutions in England, France, and Germany, and upon his return made a report to the Corporation which materially influenced later work at the Institute. He took an active part in the planning and work incident to the removal of M.I.T. from Boston to Cambridge. The plans for the laboratories, drawing rooms, and lecture rooms for the mechanical engineering department were drawn up under his supervision. The final layout of the mechanical engineering laboratories, machine tool laboratories, foundry, carpenter shop, forge shop, and mechanical engineering drawing rooms resulted from two years of intensive study under his direction.

Following the death of Richard C. MacLaurin, president of M.I.T., in 1920, Dr. Miller served as a member of the administrative committee which governed the institute prior to the inauguration of President Stratton in 1923. In 1921 he was elected chairman of the faculty.

Dr. Miller was an authority on smoke abatement, the operation and design of steam power plants, and boiler design, and his wide knowledge on these subjects brought about his service as an expert witness in many legal cases. He also served on many state boards and commissions investigating and regulating engineering matters.

He had written several books, including "Steam Boilers," jointly with Cecil H. Peabody, published in 1897, "Problems in Thermodynamics and Heat Engineering," with C. W. Berry and J. C. Riley, 1911, three editions of "Notes on Power Plant Design," the last two with James Holt, and "Notes on Heat Engineering," 1931. He also wrote articles on power plant design and heating and ventilation for engineering publications, and a pamphlet, "Training of Engineer Officers," published by the United States Shipping Board during the World War. He was the inventor of a patent on safety valves and an improvement in the Lyle gun. He was awarded the honorary degree of Doctor of Science by Rhode Island State College in 1921.

In 1917 Dr. Miller had charge of the establishment and control of a number of schools established at various places in the United States to give fundamental training to engineer officers in the operation of engine rooms of ships being constructed by the U.S. Shipping Board. Thirteen of these schools were established, one at M.I.T., and trained about 2000 men, a large number of whom were commissioned as engineer officers in the U.S. Merchant Marine. During the War Dr. Miller was engaged in confidential work for the Army under the Commanding General of the Northeastern Department.

In 1921 Dr. Miller was appointed Dean of Army Students at M.I.T. and took great pride and pleasure in facilitating the work of the numerous members of the military and naval services who were from time to time in attendance at the Institute. He was commissioned a Colonel in the Ordnance Reserve in 1922 and had been assistant district chief of ordnance in the Boston district since 1930. He was a former president of the Boston Post of the Army Ordnance Association and a charter member of the national organization. He was an active sponsor of the Reserve Officers Training Corps and helped to establish and served as first president of the unit at M.I.T.

Dr. Miller became a junior member of the A.S.M.E. in 1888 and a member three years later. He also belonged to the American Society of Civil Engineers, Boston Society of Civil Engineers, American Society of Refrigerating Engineers, and Massachusetts Charitable Mechanics' Association, and was an honorary member of the National Association of Stationary Engineers. His clubs included the Technology and University, in Boston. He is survived by his widow, Mary Willard (Reed) Miller, whom he married in 1900.

BENJAMIN MERWIN MITCHELL (1870–1932)

Benjamin Merwin Mitchell, for many years president of The Conveying Weigher Company, New York, N.Y., died suddenly of heart disease on October 8, 1932. Mr. Mitchell was also engineer of tests

for the N.Y. Lubricating Oil Co. and mechanical engineer for the Manhattan Rubber Manufacturing Company, Passaic, N.J.

Mr. Mitchell was born at South Britain, Conn., on October 24, 1870. He prepared for college at the Wesleyan Academy and received a B.S. degree from the Massachusetts Institute of Technology in 1893. During his summer vacations he worked in the shops and as locomotive fireman of the Old Colony Railroad Company. Following his graduation he entered the employ of the Manhattan Rubber Manufacturing Company. He also was connected with the Robins Conveying Belt Company, of which he was made second vice-president in 1898. In the same year he became a member of the board of directors of the N.Y. Lubricating Oil Co. and engineer for Fraser & Chalmers, Ltd., London. Five years later he was promoted to the position of assistant general manager of the latter company, designing and constructing mining plants on the Rand.

Mr. Mitchell served as captain in the Railway Pioneer Regiment of the British Army in the Boer War from 1900 to 1902. He became a junior member of the A.S.M.E. in 1894 and a member in 1905.

HENRY B. NEWHALL, JR. (1876–1933)

Henry B. Newhall, Jr., president of the Henry B. Newhall Corporation, hardware manufacturers, of Garwood, N.J., died at his home in Plainfield, N.J., on April 18, 1933. He is survived by his widow, Nellie (Elliott) Newhall, whom he married in 1903, and by a son, Richard Newhall.

Mr. Newhall was born in Plainfield, N.J., on December 13, 1876, the son of Henry B. and Ruth Frances (Evans) Newhall. He studied electrical and mechanical engineering at the Massachusetts Institute of Technology, receiving a B.S. degree in 1898. Since then he had been engaged continuously in electrical, mechanical, and civil engineering work. For many years he was vice-president of the New Jersey Foundry & Machine Co., Newhall Chain Forge & Iron Co., and Diamond Expansion Bolt Company, and secretary of the Garwood Company, with offices in New York. In 1922 he became president of the first three of these companies, and in 1928 president of the Henry B. Newhall Corporation. His work included numerous Government contracts in connection with the construction of lighthouses, lock operating machinery for dams on the Ohio River, high-pressure outlet gates for the Pathfinder and Shoshane Dams, in Wyoming, a coal-handling plant at the Newport Naval Station, and other projects. He directed the design, manufacture, and sales of the conveying apparatus, overhead track systems, etc., of the New Jersey Foundry & Machine Co., and such apparatus as expansion bolts, hand power and electric drills, of the Diamond Expansion Bolt Company.

Mr. Newhall became a member of the A.S.M.E. in 1916. He also belonged to the Union League, Engineers', Hardware, and Machinery Clubs, New York.

CLIFFORD H. OAKLEY (1869–1932)

Clifford H. Oakley, president and general manager of the Essex Rubber Company, Inc., manufacturers of mechanical specialties, rubber soles and heels, Trenton, N.J., died in Philadelphia, Pa., on August 9, 1932.

Mr. Oakley was born on October 28, 1869, at Cleveland, Ohio, the son of John Henry and Mary Ellsworth (Thompson) Oakley. He attended the Cleveland public schools and took technical courses in night school. From 1887 to 1892 he was an assistant engineer in the Maintenance of Way Department of the Erie Railroad at Cleveland. He then entered the employ of the Cleveland Rubber Company, working on the design and construction of special machinery at first and later being promoted to the position of assistant superintendent of the plant. In 1893 he was transferred to a subsidiary plant, the N.Y. Belting & Packing Co., Passaic, N.J., of which he was superintendent for two years. He returned to Cleveland in 1895 and served in the same capacity in the plant there for five years.

In 1900 Mr. Oakley took the position of factory manager of the Grieb Rubber Company, Trenton, N.J. Six years later he reorganized the company into the Ajax-Grieb Rubber Company, of which he was secretary and factory manager. In 1907 he organized the Essex Rubber Company and as its principal owner, president, and factory manager, had operated it continuously since then. For the Cleveland Rubber Company, as well as for his own, he planned and built new buildings and purchased or designed and erected the necessary machinery and other equipment.

Mr. Oakley became a member of the A.S.M.E. in 1915. He was past-president of the Trenton Chamber of Commerce, director of the New Jersey Manufacturer's Association, and commodore of the Trenton Sea Scouts. He belonged to the Trenton and Philadelphia Yacht Clubs, the Trenton and Carteret Clubs, and the Masonic fraternity. He was affiliated with the Methodist Episcopal Church.

Surviving Mr. Oakley are his second wife, Elizabeth (Moon) Oakley, whom he married in 1907, and two children, George T. Oakley, of Pennington, N.J., and Mrs. Joseph B. Hewitt, of Trenton. His first wife, Minnie (Cook) Oakley, died in 1902.

OLIVER M. OLSON (1883–1933)

Oliver M. Olson, vice-president of Samuel Olson & Co., New York, who died on April 26, 1933, was born at Sheboygan, Wis., on December 12, 1883, the son of Louis Sanders Olson. For a number of years he was employed by the Consolidated Indiana Coal Company, a subsidiary of the Chicago, Rock Island & Pacific Railway Co., advancing from clerk to assistant auditor, and by the C.R.I. & P. as accountant in the comptrollers office.

From 1912 to 1915 Mr. Olson engaged in track elevation and valuation work for the same road, then for nine years helped to supervise the machine shop and foundry of the Covel Manufacturing Company. In 1924 he went to St. Joseph, Mich., to become secretary and treasurer of Engberg's Electric and Mechanical Works, and remained there until he was made vice-president of Samuel Olson & Co.

Mr. Olson became a member of the A.S.M.E. in 1929 and also belonged to the Masonic fraternity. He is survived by his widow, Lenore (Andersen) Olson, whom he married in 1910.

FRANK EMERSON PALMER (1874–1933)

Frank Emerson Palmer, mechanical engineer for the Palmer Bros. Co., at its mills in Montville, New London, and Fitchville, Conn., died of heart failure at his home in Fitchville on April 30, 1933.

Mr. Palmer was born in Middletown, Conn., on December 23, 1874, the son of William Henry and Adelaide Randall (Wood) Palmer. In 1884 his parents moved to Norwich, Conn., where he attended the public schools and the Norwich Free Academy. He then spent two years at the Worcester Institute of Technology, Worcester, Mass., leaving that institution in 1897. The same year he entered the employ of the Palmer Bros. Co. at their Fitchville mill as carpenter's apprentice. After finishing his training he was advanced to the drafting room of the machine shop department and later made foreman of the machine shop. He had held the position of mechanical engineer since 1914. Two years later he also became general manager of the Fitchville plant and of the Bozrah Sterilizing Company, connected with the plant, and held this position until a reorganization of the Palmer Bros. Co. took place in 1929.

Mr. Palmer became an associate-member of the A.S.M.E. in 1931. In 1914 he became a trustee and shortly afterward a director of the Norwich Savings Society and also a director of the Uncas Merchants National Bank of Norwich, and continued to serve both banks until his death. He was a former member of the board of education of the town of Bozrah, Conn., and of the Connecticut National Guard. He belonged to the Phi Gamma Delta fraternity and the Episcopal Church.

Surviving Mr. Palmer is his widow, Helen Gay (Dawley) Palmer, whom he married in 1898.

FREDERICK WILLIAM PARSONS (1866–1933)

Frederick William Parsons, vice-president of the Ingersoll-Rand Company and general manager of the Painted Post, N.Y., and Athens, Pa., plants of the company, died at his home in Corning, N.Y., on April 1, 1933, following an illness of several weeks from heart trouble and complications.

Mr. Parsons was born in Corning on July 8, 1866, the second son of James A. and Emma (Land) Parsons. He attended the Corning Free Academy, graduating in the class of 1884. He served a three-year apprenticeship as a draftsman at the B. W. Payne & Sons Engine Works, Corning, was in charge of the drafting room there for two years, and then moved with the company to Elmira, N.Y., becoming superintendent of the plant.

Mr. Parsons left the employ of the Payne company in 1894 to become superintendent of the foundry of the Rand Drill Company at Ossining, N.Y. In 1898 the Rand company, upon Mr. Parsons' recommendation, took over the Weston Engine Company, at Painted Post, and established the Imperial Engine Company in its place, with Mr. Parsons as its treasurer. The first air compressors manufactured by the company were shipped in November, 1899. The next year Mr. Parsons developed an air compressor capable of obtaining pressure sufficient to produce liquid air. This machine was exhibited at the Paris Exposition that year and attracted a great deal of attention.

Records show that in 1898 the Painted Post plant employed 125

men and that the floor space was approximately 3375 sq ft. The story of expansion to the point where 1600 men were employed in a large modern factory is the story of the industrial side of the life of Mr. Parsons. He and his associates introduced many revolutionary modifications in the air compressor of those early days, permitting its application to enormous construction jobs and public works.

Products of the Painted Post plant have been used on many of the world's greatest construction undertakings during the last thirty years and more, including the Panama Canal, Boulder Dam, Ashokan Aqueduct, Italian government fortifications in the Alps, the Holland Tunnels, subway and harbor work for the City of New York, and many large buildings there, including the Empire State and Radio City. In oil fields all over the world and in gold and diamond mines in South Africa and Australia, Ingersoll Rand compressors are also used.

Mr. Parsons was responsible for a great deal of this expansion. About ten years before his death he built a large foundry at Painted Post, so that the company might cast its own parts. On his advice also, the company built many homes for its employees and aided them in purchasing them on easy terms. He worked untiringly to develop new machines and to carry the company and its employees through the depression.

In September, 1901, Mr. Parsons became superintendent of the plants at Tarrytown and Ossining, and of the Imperial Pneumatic Tool Company at Athens, making his headquarters at Tarrytown but continuing as superintendent of the Painted Post plant, which he visited weekly. He returned to Elmira to live in 1908, and since that time had been general manager of the Painted Post and Athens plants.

Four years later, in 1928, he purchased a residence in Corning and became a part of the business, social, and institutional life of his native city. He was elected a director of the First National Bank and Trust Company and was one of a small group of men who made possible the Baron Steuben Hotel in Corning. He was greatly interested in the Corning and Painted Post Community Chests and helped to plan improvements for the community. He was a director and active member of the Corning Country Club and helped to enlarge its golf course. He also belonged to the Corning Club, Elmira Country Club, and the Elmira Club. He became a member of the A.S.M.E. in 1889.

Mr. Parsons is survived by his widow, Lyda G. (Knapp) Parsons, and by three sons by a former marriage, Henry Parsons, of Great Neck, L.I., N.Y., and Frederick W., Jr., and Charles L. Parsons, both of whom are connected with the Painted Post plant of the Ingersoll-Rand Company. Their mother was Katherine (Spaulding) Parsons, who died in 1914.

JOHN BAILEY PEDDLE (1868–1933)

John Bailey Peddle died in his native city of Terre Haute, Ind., on April 6, 1933. He is survived by his widow, Alice Elliott (Oney) Peddle, whom he married in 1897, and by two daughters, Juliet Alice and Elinor Mary Peddle.

Professor Peddle was born on February 27, 1868, the son of Charles Rugan and Mary Elizabeth (Ball) Peddle. He received a B.S. degree in mechanical engineering from Rose Polytechnic Institute in 1888, and was given an M.S. in 1895 and an M.E. degree in 1900 by that institution. Following his graduation he was employed by the Thomson-Houston Electric Company, Lynn, Mass., testing dynamos and calibrating instruments for nearly a year. He then did drafting for the Dodge Coal Storage Company, Nicetown, Pa., for a few months and later assisted in the erection of storage plants for the company in New Jersey and Massachusetts. From early in 1891 until 1894 he was associated with Henry R. Worthington, Brooklyn, N.Y., in drafting work, and the Testing Department of the Worthington Hydraulic Works.

In 1894 Professor Peddle secured a position as instructor in drawing at Rose Polytechnic Institute. During the next ten years he advanced to the position of full professor and head of the machine design department. He served as acting president from 1928 to 1930, when a stroke of paralysis necessitated his giving up all work for a time. He was made professor emeritus on March 1, 1933.

Professor Peddle's work on the construction of graphical charts was published in 1910, and he wrote numerous articles in his field for publication in engineering journals. He became a member of the A.S.M.E. in 1909 and also belonged to the Society for the Promotion of Engineering Education, the American Association of University Professors, and the Theta Xi and Tau Beta Pi fraternities. He was active in the work of boys clubs and a member of the Congregational Church and much interested in pictorial photography, especially autochrome color photography.

CLIFTON DANCY PETTIS (1870–1933)

Clifton Dancy Pettis, who died in New York, N.Y., on July 9, 1933, was born in Paducah, Ky., on June 22, 1870, the son of the Rev. William Montrose Pettis and Katherine (Forrest) Pettis. At the age of 17 he entered the drafting room of the LaFayette Car Company, LaFayette, Ind., and after a year's experience there, worked as draftsman for the Terre Haute Car Company, Terre Haute, Ind., and as draftsman and superintendent for the Elliott Car Company, Gadsden, Ala., until 1892. From then until 1895 he was general manager of the American Car Company, Basic City, Va., during the next two years in charge of car drafting and specifications for the Southern Railway, Washington, D.C., and from 1897 to 1899 assistant to the master mechanic of Swift & Co., and the Swift Refrigerator Transportation Company, Chicago. He returned to railroad work as general foreman of the Car Department of the Illinois Central Railroad, at its Burnside Shops, Chicago, in 1899 and remained in that position for five years. From 1904 to 1908 he was supervisor of the Car Department of the St. Louis and San Francisco and the Chicago and Eastern Illinois Railroads, Frisco System, located at St. Louis, Mo.

In 1908 Mr. Pettis became connected with the Hewitt Manufacturing Company, with which he remained for three years. He then began an association of more then ten years with the American Brake Shoe & Foundry Co., first at Chicago, and later as works manager at Mahwah, N.J., and vice-president, with headquarters in New York.

After leaving this company in 1922 he engaged in research and development work in his field in New York for several years, leading to the establishment, in 1928, with his son, of the Continental Brake Shoe & Equipment Co., New York. Mr. Pettis was president of the company. He held many patents on repetition molds and brake shoes.

Mr. Pettis became a member of the A.S.M.E. in 1905 and was an Episcopalian. He is survived by his widow, Charlotte B. (Lyon) Pettis, whom he married in 1895, and by two children, Mrs. Elizabeth P. Croft, Washington, D.C., and William M. Pettis, Ardsley-on-Hudson, N.Y.

WILLIAM DESSAUER PHILIPS (1880–1933)

William Dessauer Philips, who died on February 4, 1933, was born in Brooklyn, N.Y., on December 17, 1880. Although his formal education did not go beyond high school, he studied steam engineering and heating and ventilating through correspondence courses of the International and American Correspondence Schools.

At the age of twenty Mr. Philips began work as New England sales representative of the Parson Manufacturing Company, New York, N.Y. Four years later, in 1904, he became head of the steam department of the Rossiter McGovern Company, in New York, with which he remained for three years. He next spent an equal period as head of the New York office and sales manager for the United States for the Bradley Pulverizer Company. For six months in 1910 he was salesman of general power equipment for the A. D. Granger Co., New York, then became sales manager for the Greeff Engineering & Manufacturing Co., Newark, N.J. His work there included air conditioning, heating, and ventilating, and during this period he originated a varnish dryer.

For about ten years, beginning in 1915, Mr. Philips was manager of engineering and sales for the A. S. Nichols Co., New York. He designed and directed the manufacture and sales of dryers for lumber, veneer, varnish, paints, as well as air conditioning equipment. He was sales engineer for Drying Systems, Inc., New York, from then until 1928, since when he had been vice-president of the Nichols Products Corporation, New York.

Mr. Philips became an associate-member of the A.S.M.E. in 1918.

CHARLES PIEZ (1866–1933)

Charles Piez, chairman of the board of the Link-Belt Company, Chicago, Ill., and Past-President of The American Society of Mechanical Engineers, died of pneumonia on October 2, 1933, at the Garfield Hospital, Washington, D.C. Mr. Piez's health had been failing for the past five years, and since February, 1932, he had been inactive in business. He had been living in Washington since May, 1933.

Mr. Piez had served the Link-Belt Company and its predecessor, the Link-Belt Engineering Company, ever since his graduation from Columbia University in 1889. His career was thus coincident with the great developments in material-handling and conveying machinery that have become such characteristic features of production and labor-saving methods for which this country is now noted. It

was another A.S.M.E. past-president, James Mapes Dodge, who saw the possibilities of developing the link belting invented by William D. Ewart, of Chicago, and joined with him, in 1888, in the establishment of the Link-Belt Engineering Company, in which Mr. Dodge carried out his ideas along strictly engineering lines and with a highly specialized engineering staff. Mr. Piez came under the influence of this environment within a month of his graduation from Columbia. Starting as a draftsman, he served successively as chief draftsman, chief engineer, general superintendent, general manager, president, and, from 1924 to February, 1932, chairman of the board. His career illustrates in a spectacular fashion how the technical graduate with capacity for executive responsibilities and a flair for the financial and non-engineering aspects of industrial life passes into a position of leadership in business following a helpful and necessary experience in the rigid discipline of engineering design and production. His estimate of the value of an education in engineering is contained in an address delivered at Akron, Ohio, in the fall of 1929, in which he said:

"If I had my life to live over again, I am certain that I would start it with a course in mechanical engineering, because that represents the best means of acquiring a broad foundation for a useful, practical life, one that not only provides opportunities for a professional career, but also affords abundant opportunity for success in a business career.

"This is true because engineering training develops the ability to dig up and properly weigh the facts before reaching a conclusion, and this ability is the basis of sound judgment in every walk of life."

As the chief executive of a large industrial enterprise, Mr. Piez had an opportunity to appraise the value of sound management principles. In a message to the members of The American Society of Mechanical Engineers, assembled in Washington in April, 1930, to celebrate the founding of the Society, Mr. Piez wrote: "It has been in large measure due to the engineer that management has been raised to the dignity of an art, and that it is now generally recognized as the most important factor in the conduct of industry."

He sensed keenly the progress that engineering has made in recent years toward a broader understanding of its economic possibilities and responsibilities. Unable, because of illness, to deliver the customary presidential address to the A.S.M.E. in December, 1930, he telegraphed his greetings and concluded with these words of high hope and far-sighted vision. "The A.S.M.E. ought to make itself a commanding factor in every field of mechanical engineering, for the profession has for its domain not only the strictly technical problems of industry, but the economic problems as well. There are glorious opportunities ahead. It is for our successors to embrace them."

Largely as a result of his ability as an organizer and manager, Mr. Piez was selected by Edward N. Hurley, then chairman of the U.S. Shipping Board, in November, 1917, to be vice-president and general manager of the Emergency Fleet Corporation. As the chief executive of this enterprise, and later succeeding Mr. Schwab as director general, Mr. Piez advised upon and directed the expenditure of three billion dollars. He had control over 600,000 persons employed in the shipyards and fabricating plants, and in industries furnishing supplies to the yards.

Mr. Piez joined the Emergency Fleet Corporation in its earlier days, and the character of the organization, its personnel, and its methods in accomplishing the great task assigned to it in the War, were largely his. Mr. Schwab, when leaving the Emergency Fleet Corporation, said of Mr. Piez: "I regard you, above all other men, as having contributed more to the work done in the Fleet Corporation than any one else."

Mr. Piez resigned on May 1, 1919, as director general, to return to his former business. He continued as president of the Link-Belt Company until 1924, when he became chairman of the board, and was active in this position until February, 1932.

Mr. Piez served as president of the Commercial Club of Chicago. His periods of service as president of the Illinois Manufacturers' Association were from 1911 to 1913 and again from 1924 to 1925. He was a director of the Drexel State Bank of Chicago; director, Illinois Bell Telephone Co.; a member of the Executive Committee of the Museum of Science and Industry founded by Julius Rosenwald. He received the honorary degree of doctor of commercial science from New York University. He was a member of the following engineering societies: American Institute of Mining and Metallurgical Engineers, Society of Naval Architects and Marine Engineers, Western Society of Engineers, and Engineers' Society of Northeastern Pennsylvania. He became a member of the A.S.M.E. in 1894, and was elected to the presidency in the fall of 1929. His term was coincident with the Fiftieth Anniversary of the Society, but failing health made it impossible for him to attend the celebrations in New York and Washington, and the Annual Meeting in 1930.

Mr. Piez was instrumental in framing much of the labor legislation of the State of Illinois, having been a member of the commission that drew the State Factory Act. He was chairman of the Workmen's Compensation Commission, and a member of the State Arbitration Board, State of Illinois.

Mr. Piez was born on September 24, 1866, at Mayence, Germany, the son of Jacob and Catherine Liebig Piez, naturalized American citizens who were traveling in Europe at the time. He was graduated from the School of Mines, Columbia University, in 1889, with the degree of engineer of mines. In 1896 he married Laura Oliva Flora, from whom he was divorced in 1919. His second wife was Mrs. Laura Sadler Cocke Piez, of Laurel, Md., whom he married in 1922. He is also survived by his sister, Miss Ernestine Piez, of New York, N.Y.

Mr. Piez was a member of the Chicago, University, Commercial, and Flossmoor Country clubs of Chicago, the Union League Club of Philadelphia, and the Chevy Chase Club of Washington. He was a Republican and a Mason.

EDWIN DANIEL PINGREE (1862–1933)

Edwin Daniel Pingree, for many years vice-president and engineer of the Manufacturers Mutual Fire Insurance Company, Providence, R.I. (one of the Associated Factory Mutual Fire Insurance Companies), died on February 1, 1933. He had been in ill health for some years.

Mr. Pingree, the son of Daniel Wentworth and Sarah Jane (Durham) Pingree, was born on December 28, 1862, at Lawrence, Mass., and was graduated from the high school there in 1878. He then worked for a time as a draftsman in the shop of J. H. Horne & Sons, paper mill machinery manufacturers, Lawrence, and later spent several years at the Ryegate (Vermont) Granite Works, becoming an expert in designing and erecting monumental granite work. He returned to Lawrence as an inspector of construction work for George G. Adams, architect.

In 1892 Mr. Pingree became an assistant to John R. Freeman, Providence, R.I., consulting hydraulic engineer and chief of the Inspection Department of the Associated Factory Mutual Fire Insurance Companies. His expertness as a draftsman quickly brought him opportunities to help Mr. Freeman in his work as a member of the Massachusetts Metropolitan Water Board, appointed to construct large additional water works for Boston, as well as in his work for the insurance companies.

Mr. Pingree entered the Massachusetts Institute of Technology in 1893 and was graduated with an S.B. degree in 1896. He was employed by Mr. Freeman during vacation periods, and returned to his office following graduation to assist him on water supply problems for Boston and New York, studies in connection with the Charles River Dam, and other investigations. He later joined the staff of the Manufacturers Mutual Fire Insurance Company, became a student of taxation, and was very helpful to the companies in litigation regarding Federal taxes in 1912 and 1913. He was elected vice-president and engineer of the Manufacturers Mutual Fire Insurance Company in 1907 and held that position until his retirement on July 1, 1932.

Mr. Pingree became a member of the A.S.M.E. in 1905. He also belonged to the Boston Society of Civil Engineers, New England Cotton Manufacturers Association, University and Turks Head Clubs, Providence, and the Congregational Church there. His wife, Louise C. (Sawyer) Pingree, whom he married in 1898, died in 1904. He is survived by a son, Daniel Pingree, of Providence.

STEPHEN HENDERSON PITKIN (1860–1933)

Stephen Henderson Pitkin, who died at his home in Akron, Ohio, on April 23, 1933, after a long illness, was first vice-president of the Wellman-Seaver-Morgan Company prior to his retirement in 1921.

Mr. Pitkin was born at Troy, Ill., on October 26, 1860, the son of Caleb J. Pitkin, a Presbyterian minister, and Elizabeth (Bancroft) Pitkin. He went to Akron in 1875 and after completing his high school education there served a three-year apprenticeship as a machinist in the works of the Webster Camp & Lane Machine Co. He then worked for the company as a draftsman for five years, as mechanical engineer for two years, and from 1887 until the company was consolidated with the Wellman-Seaver-Morgan Company in 1903 as secretary and general manager.

Mr. Pitkin has been a member of the A.S.M.E. since 1891 and of the American Institute of Mining and Metallurgical Engineers since 1887. He had served as a trustee and deacon of the First Congregational Church in Akron for many years and also belonged to the Masonic fraternity.

Surviving Mr. Pitkin are his widow, Bessie (Alexander) Pitkin, a son, Francis A. Pitkin, of Philadelphia, Pa., and three daughters,

Mrs. Elizabeth Rowell, Akron, Mrs. Grace Larkins, East Liverpool, Ohio, and Mrs. Marion St. John, by a former marriage, Canandaigua, N.Y. His first wife, Ada (Maltby) Pitkin, died in 1893.

HOLBROOK FITZ JOHN PORTER (1858–1933)

Holbrook Fitz John Porter, who after long experience as a consulting industrial engineer devoted his later years largely to the furtherance of prison reform and the establishment of a National Museum of Engineering and Industry, died of pneumonia at his home in New York, N.Y., on January 25, 1933.

Mr. Porter was a member of the Board of Directors, secretary, and one of the few honorary members of the Association of Grand Jurors of New York County and in 1926 and 1927 was editor of *The Panel*, official magazine of the Association. He is credited with being among the first to put the Association before the public as a militant civic group. He conceived the idea of removing the prisons of the City of New York from Welfare to Riker's Island and in 1924, upon designation of the August Regular Grand Jury for New York County, he prepared a comprehensive study of the conditions existing in the prisons on Welfare Island, and a plan for their removal to Riker's Island. This plan was endorsed and published by the Prison Association of New York and the Association of Grand Jurors of New York County. He was very active in obtaining the official action necessary to effect the plan, and the model penitentiary on Riker's Island, the cornerstone of which was laid in 1931, is the result of his work.

In 1920 Mr. Porter organized and became secretary of the movement to establish a National Museum of Engineering and Industry to trace and perpetuate the progress of practical scientific development. The scope of this institution, as he had envisioned it, was to be somewhat similar to that of the Deutsches Museum at Munich, Germany, and the South Kensington Museum in London, England. It was planned to have the main museum in Washington, D.C., with branches throughout the country. He had prepared a comprehensive plan for the establishment and maintenance of the museum, and this plan was shortly to have been brought before a group of educators, engineers, and museum experts for study.

Mr. Porter was born in New York on February 28, 1858, the son of General Fitz John, who fought in the Mexican and Civil Wars, and Harriet Pierson (Cook) Porter. He was educated at St. Paul's School, in Concord, N.H., and was graduated in 1878 from Lehigh University, Bethlehem, Pa., with a mechanical engineering degree. His first position was with the Delamater Iron Works, in New York, as apprentice and draftsman, from 1878 to 1882. He was then, for two years, assistant engineer of the New Jersey Steel and Iron Company, at Trenton, N.J. From 1884 to 1886 he was college engineer at Columbia University, installing heating, ventilating, and electric lighting systems in new buildings, and from 1886 to 1890 was superintendent of buildings and grounds there, being the first to hold that position. He made a thorough study of cost accounting while at the University.

During the next two years Mr. Porter worked as superintendent of the Cary & Moen Steel Wire & Springs Works, New York, and the Braddock Wire Company, Pittsburgh, Pa. He introduced the first cost system in these two companies. He then went to Chicago where he helped in the engineering work incidental to laying out the Columbian Exposition of 1893, being assistant mechanical engineer during the construction period and assistant chief in charge of Machinery Hall while the Exposition was in progress.

At the close of the Exposition Mr. Porter became associated with the Bethlehem Steel Company and was successively their western representative, with headquarters in Chicago, assistant sales manager, at Bethlehem, and eastern representative, in New York. A comprehensive paper on "Hollow Steel Forgings," presented with lantern slides before the A.S.M.E. St. Louis Meeting in 1896, was received with great interest and afterward given before engineering classes in several western universities. During the next few years Mr. Porter lectured on the subject of advanced methods in the making of steel forgings more than one hundred times, at various technical society meetings and before classes in western universities, in addition to writing articles on the subject for the technical journals. These addresses and articles were instrumental in arousing interest in nickel steels, and stimulated their increased use in industry.

As vice-president and general manager of the Westinghouse, Nernst Lamp Company from 1902 to 1906 Mr. Porter established in that firm the first shop committee with employee representation and the first factory fire drill in this country. In 1906 he opened his own office in New York for the practice of consulting engineering, specializing in introducing scientific management in industrial plants. He advanced employee representation and hygienic and safety measures, especially safeguards against loss of life by fire, and his papers and discussions before meetings of the A.S.M.E. in succeeding years reflected his strong interest in the humanitarian and educational aspects of his work.

In 1909 Mr. Porter was retained by the Russell Sage Foundation to study hygienic conditions in industrial plants in Pittsburgh in connection with the Foundation's survey of that city. His report on the condition of public swimming pools led to his appointment by the Board of Health of the City of New York to investigate pools there. He wrote the first article on this subject for *The Survey*.

Mr. Porter initiated a movement in New York which resulted in the creation by the State legislature of the New York State Factory Investigating Commission, to which he became adviser on fire hazard to life. His interest in this phase of safety work led to his invention of the vertical fire wall, a device which permitted the escape of persons through fireproof doors in a fireproof wall to an adjoining building or another part of the burning building, completely shut off from the flames. This was the first practical method of saving helpless inmates of hospitals and asylums from fires and the adoption of the device in the State of New York was required by law. His work also resulted in the revision of the New York State Labor Law and the enactment of the first fire prevention code in this country. Mr. Porter was engaged by the City of New York to install fire preventive measures in the hospital buildings of the Department of Public Charities, more than two hundred in number.

From 1915 to 1918 Mr. Porter was consultant to the Hercules Powder Company, Wilmington, Del., on employment management for increased efficiency. After the close of the World War he served as consultant on employment representation in industry to more than two hundred business concerns for the New Jersey State Chamber of Commerce.

Mr. Porter was a leading authority on the life of Captain John Ericsson. He had collected a large amount of Ericssonia and was instrumental in having appropriate tablets placed at historic sites to commemorate particularly eventful occurrences in his life. In 1918 he instituted a movement to nominate Captain Ericsson for the Hall of Fame.

In addition to his early lectures and writings on steel, Mr. Porter was the first lecturer in the course of industrial organization and management in the Alexander Hamilton Institute and was on its staff of lecturers at the time of his death. He also lectured at the Harvard University Graduate School of Business Administration, Columbia and New York Universities, Massachusetts Institute of Technology, Tufts College, the Universities of Illinois and Michigan, Lehigh University, and many other institutions of learning and engineering and business associations throughout the country. His writings included many on industrial management, as well as articles on safety and other subjects in which his interest has already been evidenced.

Mr. Porter became an associate of the A.S.M.E. in 1880 and a member four years later. In 1911 he organized and became the first secretary of the Efficiency Society, the first society devoting itself to the study of organization and management. In 1912 he was secretary of the organizing committee for the Congress of the International Association for Testing Materials. He was a member of the British Iron and Steel Institute, a member of the Board of Directors and of the executive committee of the American Society for the Prevention of Blindness, and a former president of the Lehigh University Alumni Association. He was for many years vestryman of the Church of the Ascension (Episcopalian), New York. His clubs included the Engineers', New York, and the Aztec Club of 1847 (Society of the Mexican War), Washington, D.C.

Surviving Mr. Porter are his widow, Rose (Smith) Porter, whom he married in 1888, a son, Fitz John Porter, and a granddaughter, Peggy Porter.

FRANCIS HENRY RICHARDS (1850–1933)

Francis Henry Richards, a Founder Member of the A.S.M.E., died on April 29, 1933, at a hospital in New Britain, Conn., where he had lived during the last ten years of his life. Mr. Richards, who in recent years was consulting engineer and designer for the Pratt & Whitney Co., Hartford, Conn., had more than 1,000 patents to his credit.

Mr. Richards was born at New Hartford, Conn., on October 20, 1850, the son of Henry and Maria (Whiting) Richards. He attended public and private schools at New Hartford, and for several years was employed as a mechanical engineer at the Stanley Rule and Level Works, New Britain, Conn., and by the Pratt & Whitney Co. He opened an office in 1886 at Hartford and made a study of patent laws. He removed his office to New York in 1898 and until his retirement in 1924 was consulting engineer and patent adviser to a number of large manufacturing companies.

Among the inventions of Mr. Richards was a machine for manufacturing golf balls; a machine to fold, gum, count, and bind envelopes in one process, which he invented for the United States Envelope Company; more than 250 patents for weighing machines; more than 200 for machine-shop tools and appliances; more than 100 for typographic machines; and many for coke furnaces and door springs.

Mr. Richards married Mrs. Clara (Blasdale) Dole, of Springfield, Mass., who died in 1908. His stepson, Frederick J. Dole, in charge of the New York office, survives him.

Mr. Richards was also a member of the American Association of Inventors and Manufacturers, Civil Engineers Club, Cleveland, Ohio, Odd Fellows, and Masonic fraternity.

FRANK RICHARDS (1839–1933)

Frank Richards, retired mechanical engineer and editor of engineering publications, died at his home in North Plainfield, N.J., on May 21, 1933.

Mr. Richards was born in Taunton, Somersetshire, England, on February 10, 1839, and came to the United States with his parents when he was ten years old. They resided in Whitehall, Plattsburg, and Troy, N.Y., at the latter place for 30 years. He was associated with a number of machinery firms during these years, among them the Starbuck Brothers and Tolherst Company, of Troy, and for several years after going to New York was superintendent of the Ingersoll-Sargent Company.

In later years Mr. Richards devoted himself to writing technical articles on engineering. He was assistant editor of the *American Machinist* for ten years and managing editor of the *Compressed Air Magazine* for twelve. During the past ten years he had lived with his son, Frederick H. Richards, and spent his time largely in reading, writing, and the keeping of scrapbooks containing poetry and records of all important happenings. He wrote for the *American Machinist* for several years under the pen name of Tecumseh Swift. Although in poor health during recent months, he had only two weeks before his death submitted an article to the *Compressed Air Magazine*.

Mr. Richards became a member of the A.S.M.E. in 1891.

HARRY ALLEN RISING (1880–1933)

Harry Allen Rising, mechanical engineer for the United Shoe Machinery Corporation, Beverly, Mass., died on April 19, 1933. He became a member of the A.S.M.E. in 1929.

Mr. Rising was born at Hoosick Falls, N.Y., on February 22, 1880, the son of William Henry and Isabel (Allen) Rising. His high school education was supplemented by private study in mechanical drawing, machine design, and advanced mathematics.

From 1900 to 1905 Mr. Rising was draftsman and designer for the Draper Corporation, Hopedale, Mass., working on textile machinery. The next year was spent as designer for the Chapman Valve Company, Indian Orchard, Mass., working mostly on jigs, fixtures, and small tools. At the H. B. Smith Co., Westfield, Mass., he was in charge of the design of steam radiator and boiler testing fixtures, conveyors, etc., for their factory use from 1906 to 1908. For part of the following year he designed wire forming machinery for the Scoville Manufacturing Company, Waterbury, Conn. His next position was that of designer for the Crompton and Knowles Loom Works, Philadelphia, Pa., where he remained from 1909 to 1912. After that he was again connected for four years with the Draper Corporation. He had been with the United Shoe Machinery Corporation since 1916. His work dealt only with shoe machinery until 1919. During the next three years he supervised the adoption of all machines, and since then he had been in charge of all design, standardization, experimental, and testing work for the company and served as a consultant with other companies. He had taken out a number of patents which were assigned to the United Shoe Machinery Corporation. He collaborated with Prof. Eliot F. Tozer, of Northeastern University, in the writing of a textbook on machine drawing, which was finished shortly before his death.

Mr. Rising became a member of the A.S.M.E. in 1929. He also belonged to the American Institute of Electrical Engineers, the Masonic fraternity, and the Unitarian Church. He was a private in the Massachusetts Volunteer Militia, Sixth Regiment Infantry, from 1901 to 1904.

Mr. Rising is survived by his widow, Marian (Harris) Rising, whom he married in 1906, and by a daughter, Priscilla E. Rising.

ALLAN DOUGLAS RISTEEN (1866–1932)

Dr. Allan Douglas Risteen, director of technical research and editor of safety publications for the Travelers Insurance Company of Hartford, Conn., died on December 30, 1932, at his home in that city.

Dr. Risteen, who was an author of internationally known texts upon technical subjects, with particular reference to safety engineering, was born on August 15, 1866, in Amesbury, Mass., the son of John C. and Hannah (Morrill) Risteen. He was graduated from Worcester Polytechnic Institute in 1885 with a B.S. degree in civil engineering and took advanced courses in foreign languages and physical science at Dartmouth College. Following three years of research in mathematics and physics, Dr. Risteen received a Ph.D. degree from Yale University in 1903. His thesis on "The Numerical Evaluation of the Absolute Thermodynamic Scale of Temperature" has been given to the Yale University library by Mrs. Risteen since his death. In 1929 he was awarded the honorary degree of doctor of engineering by Worcester Polytechnic Institute.

After his graduation from Worcester Polytechnic Institute he was associated with the United States Coast and Geodetic Survey. Together with Prof. Chas. S. Pierce he conducted a series of extensive pendulum observations at the Stevens Institute of Technology, Hoboken, N.J., for the Government, and later he worked for a time for the Survey in Washington, D.C. For a year beginning in August, 1887, he was mechanical editor for *Power*, and subsequently a regular contributor to that publication in the field of electricity.

In 1889 Dr. Risteen became connected with the Hartford Steam Boiler Inspection and Insurance Company as assistant editor of its house organ, *The Locomotive*, and in a short time was made its editor, a position which he retained until he joined the Travelers Insurance Company as director of technical research in 1911. During this period he also was in charge of computations for the Massachusetts State Typographical Survey at Boston; served in an editorial capacity for the Century Dictionary, of which he was an associate editor in 1892–1893, and for the Encyclopedia Americana; and wrote or directed the preparation of other scientific articles, so far as his time permitted.

Dr. Risteen wrote on astronomy, mathematics, civil engineering, and methods of promoting safety in industrial plants. His astronomical calculations have been widely used, particularly those for the orbit of 70 Orphiuca, one of the double stars, which were more minute than any previous calculations. An address delivered by him at the Worcester Polytechnic Institute in 1895 on "Molecules and the Molecular Theory of Matter," as well as special lectures in steam engineering which he gave later at Yale University, were subsequently published and are still used as textbooks. He was also a visiting lecturer in mechanical engineering at Yale. Among his other publications is a book of conversion tables for comparing metric units with other standards of measurement and a 200-page book on "Scaffolds" which he issued in the interest of safety. During the World War he worked untiringly to promote safety for workers in munitions plants.

The Yale University Library is also the recipient of three large typewritten, morocco-covered volumes, containing hitherto unknown facts about the life of Shakespeare collected by Dr. and Mrs. Risteen during the last twenty years or more of his life.

Dr. Risteen became a member of the A.S.M.E. in 1912. He was a past-president of the American Society of Safety Engineers (now the Engineering Section of the National Safety Council) and represented that society on the Sectional Committee on a Safety Code for Machinery for Compressing Air for which the A.S.S.E. and A.S.M.E. are joint sponsors. He had also served on the Joint Research Committee on Welding of Pressure Vessels, and on its Subcommittee No. 1 on Methods of Test, which were discharged in November, 1931, upon the completion of their work. He also belonged to the American Association for the Advancement of Science, Yale Engineering Society, Verein deutscher Ingenieure, the Chemist's Club of New York, and the Association des Industriels de France contre les Accidents du Travail.

In 1928 the American Welding Society, of which he was once an officer, chose Dr. Risteen to represent the organization at the world congress of engineers in Tokio, Japan, but illness prevented his acceptance. He was a vice-director of the American Bureau of Welding and a member of Sigma Xi, honorary scientific fraternity, Camp Fire Club of America, and the Twentieth Century Club of Hartford. For many years, he was a trustee of the Hartford Public Library.

In 1887 Dr. Risteen married Ella M. Whitney, of Upton, Mass., who survives him.

ROBERT RIPLEY RUST (1879–1933)

Robert Ripley Rust, for many years personal assistant to August Heckscher in both his business and charitable activities, died on July 23, 1933, at the New York Hospital-Cornell Medical Center,

New York. Mr. Rust, who had heart disease, had been taken to the hospital from his home, Five Acres, Millstone, N.J., a few days before his death.

Mr. Rust was born on February 28, 1879, at Lorraine, Minn., the son of John and Sarah (Ripley) Rust. He studied chemical engineering at the Armour Institute from 1893 to 1896 and spent the next four years working on chemical problems for Armour & Co. In 1900–1902 he was superintendent of the Somerset Chemical Company, Bound Brook, N.J., and during the next three years was connected with the New Jersey Zinc Company, at Beaumont, Tex., and as superintendent of its Waukegon, Ill., plant.

While with the New Jersey Zinc Company Mr. Rust first met Mr. Heckscher, who in 1907 gave him a position in The Central Foundry Company, one of his interests. Mr. Rust remained with the company until after the close of the World War, rising to the position of president.

Mr. Heckscher sent Mr. Rust to Germany in 1929 to study the housing situation there, particularly the German examples of slum clearance. On his return he became a member of the National Housing Committee for Congested Areas through which he and Mr. Heckscher advocated slum housing built on a large scale under a new state law with the aid of the city. Mr. Rust was also president of the Heckscher Foundation and of the Dry Ice Corporation from 1926 to 1931. He retired from the vice-presidency of the Somerville Iron Works, Somerville, N.J., in February, 1933.

Mr. Rust became a member of the A.S.M.E. in 1912 and also belonged to the Engineers' and Chemists Clubs in New York and the Raritan Valley Country Club. He is survived by his widow, Imogene (Bradford) Rust.

EDWIN RUUD (1854–1932)

Edwin Ruud, inventor of the Ruud automatic gas water heater and president of the Ruud Manufacturing Company, Pittsburgh, Pa., died on December 9, 1932, at his home in that city.

Mr. Ruud was born in Askim County, Norway, on June 9, 1854, and was educated there. After being graduated in 1876 as a mechanical engineer from the government's technical college at Horten, he worked as a draftsman and in the shops of the Katerineholms Jernvork, Frederickshold, Norway, for two years. Subsequently, for an equal period of time, he was employed as a machinist by the Bergunds Michine Werkstad, Stockholm, Sweden, principally engaged in the fitting up and erection of marine engines. He became interested in American-made machines and tools and American manufacturing methods and in 1880 came to the United States to learn more about them. During the next seven years he was employed by William Sellers, Philadelphia, Pa., the Pennsylvania Railroad Company, in its engineering office at Altoona, Pa., the Strong Locomotive Engine Company, Philadelphia, and the Scott Foundry, Reading, Pa., where he engaged in design work.

Mr. Ruud received his citizenship papers in 1887. In that year, also, he was offered a position by George Westinghouse. He was assigned to the Fuel Gas & Electric Engineering Co., Pittsburgh, which had been formed by Mr. Westinghouse and associates to develop the gas producer machine. A variety of gas burning appliances for domestic and industrial use were also developed, among them the automatic gas water heater, which was first manufactured in 1888 and 1889. With the increasing production of natural gas about this time, Mr. Westinghouse turned his attention to the problems involved in its practical utilization. The Fuel Gas & Electric Engineering Co. was reorganized as the Pittsburgh Meter Company and Mr. Ruud developed one of the first small accurate proportional meters for measuring the flow of gas from a well.

Mr. Westinghouse was also deeply interested in the development of the gas engine as a prime mover and after a time assigned Mr. Ruud to this work. The Westinghouse gas engine, for which he was largely responsible, was widely used until the steam turbine and turbo-generator took its place.

After being with the Westinghouse companies for a number of years Mr. Ruud purchased the patents for the automatic gas water heater and also an automatic water distiller which had been taken out in his name and assigned to the Westinghouse companies. Other projects had held greater interest for Mr. Westinghouse and these two inventions had never been pushed. In collaboration with James Hay, a Pittsburgh plumber, Mr. Ruud improved these appliances and began to put them on the market, the first heater, under the trade name of the Pittsburgh water heater, being built in 1895. The Ruud Manufacturing Company was formed two years later and offered two types of heaters, the automatic storage system and the instantaneous automatic water heater.

These heaters were designed for use with natural gas and it was not until 1902 that Mr. Ruud perfected improvements making them suitable for artificial gas. During the next two years a sales force was built up and branch offices and agency connections established in many large cities in the United States and Canada. In 1905 Mr. Ruud retired from his association with the Westinghouse companies and had since devoted his entire time to the Ruud organization. He had invented many devices to make the Ruud heater adaptable to various needs and built up a highly successful and widely known company.

Mr. Ruud was honored by the king of Norway's decoration, the Order of St. Olaf, and was a Commander of the Horten technical college. An honorary Doctor of Science degree was conferred upon him by the University of Pittsburgh in 1927. He had been a member of the A.S.M.E. since 1889.

Surviving Mr. Ruud are his widow, Minna (Kaufmann) Ruud, and a stepson, John Henry Sorg, of Pittsburgh, as well as two sisters in Norway. He visited his summer home at Galten, Norway, each year, enjoying the fishing there. He also had a camp in Forest County, Pennsylvania. His interest in music led him to establish a fund to aid young singers in Norway.

CHARLES MacCAUGHEY SAMES (1866–1933)

Charles MacCaughey Sames, since 1916 associate editor of The American Society of Mechanical Engineers, New York, N.Y., died suddenly of heart failure at his residence, the Hotel Robert Fulton, New York, on March 8, 1933.

Mr. Sames was born on May 12, 1866, at Rockford, Ill. He was the son of Peter and Mary E. (MacCaughey) Sames. Matriculating at the Worcester Polytechnic Institute, Mr. Sames soon transferred to the newly formed Rose Polytechnic Institute at Terre Haute, Ind., from which he was graduated, in 1886, with the degree of Bachelor of Science.

Following graduation, Mr. Sames came east and entered the Thomson-Houston Electric Company, Lynn, Mass., where he worked as a draftsman and in the testing department under Professor Elihu Thomson. In 1887 he returned to his native town to enter business with his father, Peter Sames, who was a manufacturer of agricultural implements. He continued in the business until 1900 when he entered upon his editorial career.

During his manufacturing experience, at first as superintendent and from 1894 to 1900 as business manager, Mr. Sames was in responsible charge of designing jigs, fixtures, and special tools, and the repair and alteration of the buildings, power plant, and machinery. He also designed and built various special machines and labor-saving devices, such as gang boring machines, eye benders, and a line of small direct-current generators.

Mr. Sames's first venture in literary and editorial work was the compilation and editing of a "Pocket Book of Mechanical Engineering," published in 1905, and republished in its fourth edition in 1911. This "pocket book" included valuable data gleaned from publications in foreign languages and for that reason not available to the average engineer.

In 1906 he became editor of book publications of the Engineering News Publishing Company, New York. From 1907 to 1913 he acted as associate editor of *Technical Literature*, and its successors, *The Engineering Digest* and *Industrial Engineering*, for which his experience as a designer, manufacturer, and editor of the "pocket book" provided him with an unusually valuable and comprehensive background.

Returning to the work of editing handbooks, Mr. Sames, from 1913 to 1916, assisted Professor Lionel S. Marks, of Harvard, in the preparation of Mark's "Mechanical Engineers' Handbook."

In July, 1916, Mr. Sames joined the staff of The American Society of Mechanical Engineers, as associate editor, a position which he held until his death. His talents were employed on the Society's "Journal," now known as *Mechanical Engineering*, and on the editing of engineering and technical papers for the Transactions of the Society.

Bringing to this work a broad and comprehensive knowledge of the theory and practice of the profession of mechanical engineering and an extraordinary experience in the editing of engineering literature, Mr. Sames gave the publications of the Society a deserved reputation for high quality and accuracy.

It was Mr. Sames's rare combination of editorial skill, knowledge, and experience, and familiarity with the technical literature in foreign languages as well as in English, that made him especially competent to aid in the successful publication of The Engineering Index which the A.S.M.E. acquired in 1918. His meticulous editorship of engineering papers published in the Transactions and *Mechanical Engineering* transformed hundreds of indifferently written and amateurishly prepared manuscripts into accurate and creditable contributions to the literature of engineering. And with the coming of the *A.S.M.E. News*, in 1921, his talents were employed in the final

scrutiny of its pages. On all of these publications he left the imprint of his literary craftsmanship, in the apt turning of a phrase, a remarkable clarity and conciseness of expression, a high degree of professional and editorial dignity, a meticulous accuracy, and a tastefulness in typographical arrangement.

On December 20, 1899, Mr. Sames married Lida Oliphant Falkinburgh, of Jersey City, who survives him. He is also survived by his brother, Judge Albert M. Sames, of Tucson, Ariz. He became a member of the A.S.M.E. in 1918 and also belonged to the Sons of the Revolution, New York, and to several clubs, including the Carteret and University Clubs of Jersey City, N.J. He was a Protestant and a Republican.

The following editorial appeared in the April, 1933, issue of *Mechanical Engineering:*

"For many years every page of *Mechanical Engineering* has borne the imprint of the editorial craftsmanship of Charles MacCaughey Sames, whose valued services were unexpectedly cut off in the midst of the preparation of the present issue. Elsewhere will be found a brief outline of the life of this unique figure in technical journalism. These few inadequate words are intended as a tribute to the nobility of character and the loyalty of service of one who strove to make this magazine as nearly perfect as it was in his power to do and who maintained for it traditions of worth, dignity, and accuracy.

"The combination of his qualities was unique. The dignified urbanity of his manner cloaked a warm and gracious courtesy. A sparkling sense of humor brought an informal kindliness to his relationships with his colleagues, and with the thoroughness of his culture, based on natural inclinations and an educational and professional experience that wide reading and extensive travel in other lands had augmented, made him a true gentleman, keenly sensitive to human values. An inborn love for the fine arts was expressed at times in his life by playing the piano and painting in water colors. Working for the intellectual enjoyment of the task, he developed a superior craftsmanship, but did not, as others might have done, sacrifice quantity for quality; he was a rapid and an indefatigable worker.

"Anticipating one day the time when the infirmities of old age might force his retirement from active life, he expressed to one of his colleagues the hope that some way might be found whereby he could edit an occasional paper, for he could imagine no happier way in which to end his days. His wish was answered, but not as he had anticipated, for he laid down his pencil one night and left his office in good health and with his usual vigor. Before it was time for him to return for the next day's work, without premonition, illness, or pain, he died.

"Thousands of engineers are in his debt for the papers that through him were spread upon the record, and for the skill with which he prepared their generally inept writings for publication."

EDWIN NASH SANDERSON (1862–1932)

Edwin Nash Sanderson, a member of the firm of Sanderson & Porter, New York, N.Y., and president and a director of the Federal Light & Traction Co. and of its subsidiary companies, died on November 9, 1932, at the New York Hospital.

Mr. Sanderson was born in Brooklyn, N.Y., on December 2, 1862, the son of Elnathan Lawrence and Mary Elizabeth (Nash) Sanderson. He prepared for college at the Brooklyn Collegiate and Polytechnic Institute and was graduated from Rensselaer Polytechnic Institute, Troy, N.Y., in 1886 with a civil engineering degree. He was awarded a mechanical engineering degree at Cornell University the following year and in 1925 Colgate University conferred upon him the honorary degree of Doctor of Laws.

Following his graduation from Cornell Mr. Sanderson took a position as draftsman for the Westinghouse Electric & Manufacturing Co., Pittsburgh, Pa. In 1888 he was transferred to the New York office where he worked as engineering salesman until 1894. He then spent a year at Boston, Mass., as manager of the New England District, after which he was made assistant to the general manager at Pittsburgh. He resigned in 1896 to form, with H. Hobart Porter, the firm of Sanderson & Porter, which specialized in the construction, management, and operation of electric railway, light, and power properties, and later undertook like work for industrial organizations.

Mr. Sanderson's firm began to assemble utility properties soon after its formation, and in 1910 a number of their holdings were assembled as the Federal Light & Traction Co., controlling a group of electric light, power, railroad, gas, and water properties, now more than thirty in number, in the United States and Canada. Mr. Sanderson became a director of the company in 1910 and its president in 1914.

Mr. Sanderson became a member of the A.S.M.E. in 1903. He also belonged to the American Institute of Electrical Engineers, the Rensselaer Society of Engineers, and the Cornell Society of Engineers. He was a life member of the Board of Trustees of Cornell University and served as chairman of the alumni committee for the university's semi-centennial celebration in 1919. Among his clubs were the University, City Midday, Columbia Yacht, St. Andrew's Golf, and Cornell Club of New York (of which he was president from 1916 to 1921); the Westchester Country, Town and Gown of Ithaca, Union Club of St. John, N.B., Adirondack Mountain, Orchard Lake, in the Adirondacks, and the St. Bernard Fish and Game Club of Quebec. He was a member of the Zeta Psi and Sigma Xi fraternities.

Surviving Mr. Sanderson are his widow, Mildred (Hays) Sanderson, whom he married in 1907, and three daughters by his former wife, Sarah Elizabeth Rogers, who died in 1905. The daughters are Helen (Sanderson) Chamberlain, Sybil (Sanderson) Sloane, and Katharine (Sanderson) McCaddon.

KURT G. SANDWALL (1902–1933)

Kurt G. Sandwall, who died on July 2, 1933, after a year's illness, was born on October 24, 1902, in New York, N.Y., and attended public school and Cooper Union there. Since 1920 he had been employed as a draftsman. He was with the New York Edison Company from 1920 to 1924 and again from 1928 to 1930. In the interim he worked for Thomas E. Murray, Inc., Philadelphia, Pa., and Jabez Burns & Sons, New York, except for the summer of 1926, which he spent abroad. During 1930 he was with the Wilputte Coke Oven Corporation, New York, and the following year worked for the Turner Construction Company of that city.

Mr. Sandwall became a junior member of the A.S.M.E. in 1930.

AUGUSTUS MOIR SAUNDERS (1860–1933)

Augustus Moir Saunders, for many years connected with the National Tube Company, McKeesport, Pa., died in Los Angeles, Calif., on May 23, 1933. He retired from business several years prior to his death.

Mr. Saunders was born at Tunbridge Wells, Kent, England, on March 13, 1860, a son of Francis and Sophia Caroline (Wucherer) Saunders. He attended Malvern College, Malvern, England, and served an apprenticeship in the drafting room and shops of Ormerod Grierson & Co., in London, and St. Georges Iron Works, Manchester, before coming to the United States in 1883.

Here he entered the employ of the National Tube Works Company as a draftsman. He was made assistant superintendent of the pipe mill in 1887, took charge of the mechanical department in 1896, and for 25 years was superintendent of the pipe mills, or Upper Division of the National Tube Company. He was also secretary of the Patent Department of the company for more than seven years. Either in his own name or jointly with Peter Boyd he held patents on pipe drawing and well sinking apparatus, well casing, couplers, and drill tubes.

Mr. Saunders became a member of the A.S.M.E. in 1898. He belonged to the Keystone Athletic Club, Pittsburgh, Pa., and was a charter member and for many years chairman of the Grounds Committee of the Youghiogheny Golf Club, McKeesport. He also found recreation in wood carving, water colors, and ship models.

Surviving Mr. Saunders are his widow, Harriett Gillespie (Knapp) Saunders, and four sons, Kenneth, Francis, and William Hurford Saunders, by a former marriage, and Frederick I. Saunders.

WILLIAM LAPHAM SAUNDERS (1875–1933)

William Lapham Saunders, vice-president of D. Saunders Sons, Inc., manufacturers of tools and pipe, Yonkers, N.Y., died at the St. John's Hospital there on June 22, 1933, following an automobile accident.

Mr. Saunders was born at Yonkers on October 25, 1875, and prepared for college in the schools there. He studied mechanical engineering for two years at Cornell University, in the Class of 1899, then returned to Yonkers to associate himself with the firm of D. Saunders Sons, Inc. After working several years as a machinist he was made secretary of the company in 1901. In 1910 he also took the office of vice-president, continuing as secretary until the next year, then serving six years, until 1917, as general manager, as well as vice-president.

Mr. Saunders became a member of the A.S.M.E. in 1928. He also belonged to a number of clubs in New York, Yonkers, and vicinity, including the Bankers, Cornell, and Engineers' in New York, Corinthian Yacht and Amackassin, Yonkers, and Highland Country, Garrison, N.Y. He was a member of the Masonic fraternity.

Surviving Mr. Saunders are his widow, Elizabeth (Stevens) Saunders, three sons, David, Alexander, and James, and two daughters, Eliza and Jean.

WILLIAM F. SELLERS (1856–1933)

William F. Sellers, who retired in February, 1933, as president of the Edge Moor Iron Company, Edge Moor, Del., and who was a director of William Sellers & Co., Philadelphia, Pa., died on June 14, 1933.

Mr. Sellers was born in Philadelphia on March 27, 1856, the eldest son of William Sellers, the founder of both these companies. He attended private schools and was graduated from the University of Pennsylvania in 1876 with a B.S. degree. He immediately entered the employ of the Edge Moor Iron Company as a clerk in the receiving and shipping department. Subsequently he held respectively the position of draftsman, head inspector and shipper, and engineer in charge of the construction, erection of machinery, and operation of the rolling mill. In 1886 he was elected secretary of the company, and two years later, when the Edge Moor Bridge Works was formed, taking over the bridge work of the company, Mr. Sellers was also elected its secretary. He was made treasurer of the Edge Moor Iron Company in 1897 and in 1905 also became its president. Until 1928, when his health began to fail, he directed design and production, planning and erecting buildings, and improvements in plant, machinery, and methods of manufacture. He had been a director of Wm. Sellers & Co., Inc., since 1911.

Mr. Sellers became a member of the A.S.M.E. in 1917 and also belonged to the National Association of Manufacturers and several other organizations.

JAMES A. SHEPARD (1864–1933)

James A. Shepard, who died on June 10, 1933, at his summer home, Kayutah Lake, N.Y., was born at Ohio, Herkimer County, N.Y., on January 6, 1864. At the age of 16 he joined with his father and older brother in establishing a foundry and machine shop in Montour Falls, under the name of W. H. Shepard & Sons, for the manufacture of agricultural implements and for producing special castings.

Four years later, in 1884, he became general manager of the business, which then consisted chiefly of making steam engines and boilers and building bridges. In 1895 the name of the company was changed to the Havana Bridge Works, and he continued as its general manager and chief engineer. The company was sold in 1903 and Mr. Shepard founded the General Pneumatic Tool Company, taking over the business of the machine shop of the bridge works. Mr. Shepard served as general manager and chief engineer for the first two years, then resigned the former position in order to have more time for general engineering work. He continued to act as chief engineer for the company and its successor, the Shepard Electric Crane & Hoist Co.

Several years later the Shepard Company purchased the Sprague Electric Hoist Company and the Niles Crane Company, and reorganized under the title of the Shepard Niles Crane & Hoist Corp. Mr. Shepard was actively engaged with this company as vice-president and consulting engineer up to the time of his death.

Mr. Shepard was a director of the Watkins Glen State Bank and of the Montour National Bank. For many years he was chairman of the Child Welfare Committee of Schuyler County. During the World War he was a county food administrator, and at the time of his death president of the Schuyler County Economic Council. In 1928 he was elected Assemblyman for Schuyler County in the New York Legislature, and served two terms.

Mr. Shepard became a member of the A.S.M.E. in 1914. He was active in the work of the Materials Handling Division, serving on its executive committee for some years, as chairman in 1925–1926. He also served on several subcommittees of the Division, including the Subcommittee on Formula, that on Equipment, and the Materials Handling Manufacturer's Committee. He was also a member of the Society of Terminal Engineers, and the Elmira City and Elmira Country Clubs.

Mr. Shepard is survived by his widow, Frances (Hinman) Shepard, whom he married in 1897.

HJALMAR E. SKOUGOR (1884–1932)

Hjalmar E. Skougor, consulting engineer for the Anglo-Chilean Consolidated Nitrate Corporation, New York, N.Y., died at his residence in Neshanic, N.J., on November 7, 1932. His most recent work was the design of two large nitrate plants constructed by the corporation at Maria Elena and Pedro de Valdivia, Chile.

Mr. Skougor was born at Copenhagen, Denmark, on August 12, 1884, and was graduated from the Technical Institute of that city in 1903 with an M.E. degree. His work during the next few years related to the design of steel structures, including office and loft buildings, warehouses, railroad stations, pier sheds, and theaters. He also assisted in the design of coal and ore handling machinery. For some years he was chief engineer on such construction work.

From 1913 to 1920 Mr. Skougor was associated with the companies directed by Guggenheim Bros., New York. For their Chile Exploration plant at Chuquicamata, Chile, he was designing engineer in connection with the planning and construction of a haulage system, crushing plant, leaching equipment, electrolytic plant, and the various camp facilities. He also directed the design of plants and equipment for the Braden Copper Company at Sewell, Chile.

Mr. Skougor had been responsible for other important industrial design in the course of his work as an independent consultant since 1920. He gained recognition for his original proposals regarding port, terminal, transportation, heating, and lighting projects in the metropolitan area, and presented important plans for the development of the Port of New York. He also designed complete townsite facilities for the Rosita, Mexico, property of the American Smelting & Refining Co., and designed the layout of a complete industrial city, Arvida, Canada, for the Aluminum Company of America.

Mr. Skougor became a member of the A.S.M.E. in 1921 and also belonged to the American Institute of Mining and Metallurgical Engineers.

CHARLES HEAD SMOOT (1878–1933)

Charles Head Smoot, president of the Smoot Engineering Corporation, New York, N.Y., died on January 6, 1933, at the Orange (N.J.) Memorial Hospital. He was the inventor of more than thirty devices for the regulation of steam pressure. Perhaps his best known apparatus was that for automatic steam control used in nearly all large power plants throughout the country.

Mr. Smoot was born at Ilion, N.Y., on December 6, 1878, the son of William Sidney and Mary Bunker (Head) Smoot. He attended Harvard University and the University of California and was employed for a time by small power plants on the Pacific Coast and by the Western Electric Company, Chicago, designing electrical machinery. In 1905 he engaged in designing steam turbines and kindred apparatus as American representative of Professor Rateau. From 1908 to 1922 he was engineer and director of the Rateau Steam Regenerator Company, and from 1911 to 1923 held the same position with the Rateau, Battu, Smoot Company. He became president of the Smoot Engineering Corporation in 1923.

Mr. Smoot became a member of the A.S.M.E. in 1919, and accepted the chairmanship of the newly organized Committee No. 10 on Centrifugal and Turbo-Compressors and Blowers, continuing to serve in that capacity until the committee was discharged in April, 1928. At the time of his death he was a member of the Power Test Codes Committee No. 2 on Definitions and Values. He also belonged to the Inventors' Guild, Engineers' Club, New York, Braeburn Country Club (N.J.), and the Beta Theta Pi fraternity.

Surviving Mr. Smoot are his widow, the former Katherine Elizabeth Ryan, of Freeport, Ill., whom he married in 1910, and three children, Mary Katherine, Charles Head, Jr., and William Sidney Smoot. His home was at Maplewood, N.J.

NORMAN LESLIE SNOW (1881–1933)

Normal Leslie Snow, of New Canaan, Conn., president of the Diamond Power Specialty Corporation, manufacturers of boiler accessories, New York, N.Y., died on January 27, 1933.

Mr. Snow was born on April 18, 1881, in Albany, N.Y., a son of Dr. Norman Leslie Snow and Elizabeth (Smith) Snow. He prepared for college at the Phillips Andover Academy, and received a B.S. degree from the Sheffield Scientific School, Yale University, in 1902, and a Ph.B. from the Massachusetts Institute of Technology two years later.

After leaving college Mr. Snow worked for a time for the Mosher Water Tube Boiler Company, New York, was resident engineer for about two years on the Erie Barge Canal project for the Empire Engineering Corporation, and worked in 1907, 1908, and 1909, respectively, for the Rotary Engineering Company, Sirocco Engineering Company, and American Blower Company, all of New York.

From 1909 to 1922 Mr. Snow was connected with the Terry Steam Turbine Company of Hartford, Conn., working up from the position of assistant general sales engineer to that of vice-president, general manager, and director which he held during the years 1916–1922.

Mr. Snow had been president of the Diamond Power Specialty Corporation since 1922. He was located at the Detroit office of the firm for the first four years and had been in the New York office since then. He was the inventor of many improvements on boiler apparatus.

Mr. Snow became a junior member of the A.S.M.E. in 1910 and a

member in 1926. He belonged to the Tokeneke Club and the Ox Ridge Hunt Club, of Darien, Conn. He was an excellent horseman and played on polo teams in his vicinity.

Surviving Mr. Snow are his widow, the former Helen F. Harris, whom he married in 1909, and two sons, Norman H. Snow and William H. Snow.

CHARLES AUSTIN SOARS (1895–1933)

Charles Austin Soars, president of Soars & Lovelace, Inc., Kansas City, Mo., died on July 13, 1933. Mr. Soars was born on November 5, 1895, in Philadelphia, Pa., the son of Charles A. and Margaret (Wells) Soars. He prepared for college at the Norristown (Pa.) High School, and was graduated from Bucknell University in 1917 with a B.S. degree in chemical engineering.

During the first two years after his graduation Mr. Soars was serving in the Supply Corps of the United States Navy, working for the first year at the Navy Department on the design and construction of coal storage plants and bunkering stations at seaboard points and for the second year at sea as a Supply Officer, and attaining the rank of Lieutenant.

From 1919 to 1926 Mr. Soars was employed by the Standard Oil Company of New Jersey as chemical engineer in the Development Department at Elizabeth, N.J., where he did important experimental work, including the design, construction, and experimental operation of the Ellis cracking units for refineries of the company and its subsidiaries. He was in charge of the construction of a battery of these units at the Bayway Refinery in 1922 and subsequently directed their operation.

During the years 1926–1929 Mr. Soars was connected, first, with the Louisiana Oil Refining Corporation, at Shreveport, La., as manager of refineries, and subsequently as sales and development engineer with the M. W. Kellogg Co., New York, N.Y. He went to Kansas City in 1929 as vice-president and general manager of the Refinery Engineers, Inc. He became president of Soars & Lovelace, Inc., the following year.

Mr. Soars became an associate-member of the A.S.M.E. in 1924. He also belonged to the American Institute of Chemical Engineers and the American Chemical Society, as well as to the Phi Gamma Delta fraternity.

Surviving Mr. Soars are his widow, Gladys A. (Child) Soars, whom he married in 1924, and two daughters, Margaret Helen and Gladys Jean Soars.

CARL GUSTAV SPRADO (1871–1933)

Carl Gustav Sprado, for many years associated with the Allis-Chalmers Manufacturing Company, Milwaukee, Wis., died in that city on April 11, 1933.

Mr. Sprado was born in Hamburg, Germany, on February 21, 1871, and came to the United States in 1883 with his parents, William A. and Elizabeth (Suebberstedt) Sprado, who located in Racine, Wis. Mr. Sprado attended the public schools in Racine until 1890, when he returned to Germany to complete his engineering education, graduating in mechanical engineering in 1892 from the Technikum Einbeck at Hanover.

Returning to this country after graduation he worked on the design of mining machinery for the Ottumwa Iron Works, as mechanical engineer for the Litchfield Car & Machine Co., in charge of all machinery design, on machine design for Fraser & Chalmers, for the Bucyrus Steam Shovel & Dredge Co. on dredging machinery, and for the Nordberg Manufacturing Company as designer of pumps, compressor engines, and other machinery. He entered the employ of the Allis-Chalmers Company as mechanical engineer in the engine department in 1902.

During his thirty-one years with the Allis-Chalmers organization Mr. Sprado devoted his energies almost exclusively to the design of internal-combustion engines and did important pioneer work in the development of large engines for blast-furnace-gas operation. For the last twenty years he was engineer in charge of the internal-combustion engineering division of the company. He held many patents on steam and gas engine parts, and developed one of the first successful motor cycle engines. He had contributed various articles to the technical press.

Mr. Sprado became a member of the A.S.M.E. in 1903 and had served on the Power Test Codes committees on Displacement Compressors and Blowers and Internal Combustion Engines. He was director and charter member of the Engineers' Society of Milwaukee, and belonged to the Westmoor (Milwaukee) Country Club.

Surviving Mr. Sprado are his widow, Anna (Wille) Sprado, whom he married in 1898, and two children, Erma (Sprado) Lubenow and Walter A. Sprado.

LOUIS ALVA STATES (1867–1933)

Louis Alva States, consulting engineer, Gastonia, N.C., was killed in an automobile accident there on April 25, 1933. He is survived by his widow, Annie E. (Williams) States, whom he married in 1906, and by two sons, Louis Alva States, Jr., and Thomas William States, all of Gastonia.

Mr. States was born near Wilmington, Ohio, on January 1, 1867, the son of Abram Henry and Louisa A. (Johnson) States. He attended the public schools of Livingston County, Cornell, Ill., and served a four-year apprenticeship in a power plant under his father, who was a former marine engineer. He then spent a year in the locomotive shops of the Santa Fe Railroad at Streator, Ill., and subsequently was engineer for the Streator Water Works for a short time. Consulting work from 1890 to 1900 took him to many points in the Middle West and East. During this period he made his headquarters at Chicago.

Since 1900 Mr. States had been located in Gastonia. He served as a consulting engineer to a considerable number of cotton mills in Gaston County and elsewhere in North Carolina, remodeling power plants and installing new equipment. He was instrumental in introducing the turbo-electric drive in southern cotton mills and was the inventor of the States sectional grate. At the time of his death he was consulting engineer for the American Yarn & Processing Co., Mt. Holly, N.C., Textiles, Inc., Gastonia, Merco Mills and Glenn Mills, Lincolnton, N.C.; and president of the Eureka Iron Works, Lincolnton, and the States Grate Company, Gastonia. He had contributed articles on steam engineering and combustion to *Power*.

Mr. States became a member of the A.S.M.E. in 1917 and belonged to civic and social organizations in Gastonia and to the Methodist Episcopal Church South there. Total deafness for 35 years prevented military service, but he enjoyed such sports as hunting and fishing and was a violinist.

HORATIO WARD STEBBINS (1878–1933)

Horatio Ward Stebbins, associate professor of mechanical engineering at Stanford University, California, died on February 2, 1933. He had been on the staff of the Department of Mechanical Engineering at Stanford since 1914 and in addition to his teaching duties served as chairman of the committee appointed by President Wilbur to formulate plans for the establishment of a School of Engineering. The following paragraph is taken from a memorial notice by his associate, Prof. Guido H. Marx, published in the Stanford Illustrated Review of March, 1933.

"As a teacher, his work was characterized by a thorough attention to fundamentals. Quite wisely he argued that the superstructure could take care of itself if the foundations were sound, and that no superstructure, however impressive or ornate, could long endure if erected on unsound, insecure foundations. His students always found him friendly and helpful, every ready to give his time to an unhurried clearing-up of difficulties. He had just completed a syllabus for his course in thermodynamics which was a model of selection and logical and compact presentation."

Professor Stebbins was born at San Francisco, Calif., on March 26, 1878. His father, the Rev. Dr. Horatio Stebbins, was a Unitarian minister in San Francisco who was one of those selected by Senator Stanford to aid and advise him in his plans for the foundation of Stanford University and who served on the Board of Trustees from its inception in 1886 until his death in 1902. His mother was Lucy E. (Ward) Stebbins.

Professor Stebbins attended the San Francisco public schools and was graduated from the University of California in 1899 with an A.B. degree and with Phi Beta Kappa honors. He then entered the Massachusetts Institute of Technology, completing his engineering course in 1902 and receiving the degree of B.S.

Returning to the Pacific Coast, he entered the employ of the Southern Pacific Company, Sacramento, as assistant to the engineer of tests, Mr. Howard Stillman. He engaged in general laboratory work, making physical tests and chemical analyses, and was in charge of the operation of water softeners for locomotive water supply. This work aroused in him an interest in railway transportation problems which always remained with him.

From 1905 to 1913 he was connected with the San Francisco Bridge Company as engineer and estimator. While with this company he participated in making the cost estimates for the Fort Mason Transport Docks, San Francisco, and the Pearl Harbor Dry Dock, Hawaiian Islands, designed and built three highway bridges for Lyon and Esmeralda Counties, Nevada, made a complete test of a 20-in. dredge for the company at Los Angeles, and directed other harbor improvement and hydraulic dredging work.

Professor Stebbins became a member of the A.S.M.E. in 1920. His intellectual interests covered a broad field and his reading was exten-

sive, which made his selection as a member of the Palo Alto Public Library Board a most logical and admirable one. He also belonged to the American Association for the Advancement of Science and American Association of University Professors; Engineers and Bohemian Clubs of San Francisco; and the Alpha Tau Omega and Sigma Xi fraternities.

Surviving Professor Stebbins are his widow, Elsa C. (Thompson) Stebbins, whom he married in 1906, and a daughter, Amelia Adams Stebbins.

OSCAR CARL PEDERSEN STENMAR (1893–1933)

Oscar Carl Pedersen Stenmar, who was in charge of research and development work for the Jos. Stokes Rubber Co., Trenton, N.J., died on January 17, 1933. Mr. Stenmar was born at Gosmer, Denmark, on April 1, 1893. He was educated in that country, coming to the United States in the fall of 1913. From then until 1918 he was engaged in general shop work connected with the manufacture of firebrick at the Dedier March Company, Perth Amboy, N.J. He was next engaged for short periods as draftsman on heating and power plant design for the T. A. Gillespie Loading Co., Morgan, N.J., and M. W. Kellogg Co., Jersey City, N.J., and on the design of an automatic wrapping machine for the Any Wrapping Machine Company, Newark, N.J. From 1919 to 1921 he was with the Charles Burroughs Company, Newark, during the latter part of the time as chief draftsman, working on the design of hydraulic machinery and automatic molding dies for bakelite and other compounds.

In 1921 Mr. Stenmar became supervisor of the drafting room and machine shop of the India Rubber Company, New Brunswick, N.J., working particularly on special machinery for hard rubber manufacture. About two years later this plant was transferred to Providence, R.I., as a part of the U.S. Rubber Company. Mr. Stenmar's duties there were the same as at New Brunswick. In 1930 he returned to the Burroughs Company in Newark as chief engineer. He had been with the Jos. Stokes Rubber Co. since 1931.

Mr. Stenmar became a junior member of the A.S.M.E. in 1922 and an associate-member three years later.

ROBERT LEROY STREETER (1880–1932)

Robert Leroy Streeter, consulting engineer and formerly vice-president and general superintendent of the United States Aluminum Company of Pittsburgh, Pa., died at the Veteran's Administration Hospital, Hines, Ill., on December 10, 1932.

Mr. Streeter was born at White Haven, Pa., on March 18, 1880, the son of William Faulkner and Mary Elma (Frear) Streeter. He prepared for college in the White Haven schools and entered Pennsylvania State College in 1899, graduating in 1903 with the degree of B.S. in mechanical engineering. In 1908 he received the degree of M.E. from the same institution. He served as mechanical engineering laboratory assistant for one year at the university and then entered the testing and experimental department of the Packard Motor Car Company, Detroit, Mich., where he remained for a year.

From 1905 to 1910 Mr. Streeter was employed in the construction department of the Lackawanna Steel Company, Buffalo, N.Y. This plant was then under construction and Mr. Streeter devoted his time there for four years to power research investigations and to erection work combined with testing, covering the first large installation of Koerting blast-furnace-gas engines for blowing and power purposes. This was a pioneer installation of large blast-furnace-gas engines in this country. For several years he also had charge of steam and electrical engineering courses in the Buffalo Technical Night High School, and when the attendance became so large that a separate day school was built he was also placed in charge of its machine design and experimental engineering courses.

In 1910 Mr. Streeter joined the teaching staff of Rensselaer Polytechnic Institute, Troy, N.Y. He was assistant professor of mechanical engineering for four years and professor of steam and gas engine design for three years. During this time he was engaged in consulting work and wrote many articles for the technical magazines. His book entitled "Internal Combustion Engines" has had three printings and has been widely adopted as a college textbook.

Mr. Streeter was commissioned a captain in the Ordnance Department of the U.S. Army in April, 1917, and stationed at Rock Island Arsenal. He was at first put in charge of testing and experimental work on automotive equipment and artillery materiel. Subsequently he served as armament officer for nineteen states, chief engineer, works manager, and general manager. The arsenal had about 14,000 civilian employees. His rank was advanced to that of major and he was discharged in 1919 as a lieutenant-colonel.

After his discharge Mr. Streeter joined the Aluminum Company of America and formed the mechanical engineering department. He directed this work for one year. He was then elected vice-president and general superintendent of the United States Aluminum Company, a subsidiary. During the following eleven years he was general superintendent of all manufacturing plants, making sheet, tubing, rod, wire, transmission cable, foil, powder, bottle caps and seals, extruded and rolled sections, heavy plates, and many other specialties. During this time he had general supervision of the mechanical engineering department. He was responsible for the expenditure of between fifteen and twenty million dollars used in the building of new mills and modernizing old ones, and directed the development of many new manufacturing processes which reduced cost and improved quality. The work also included the design and manufacture of many kinds of machines, rolling mills, stamp mills, printing presses, automatic machinery, and special machines peculiar to the fabrication of aluminum and its alloys.

Mr. Streeter joined the A.S.M.E. as an associate in 1907 and had been a member since 1913. He was also a member of the main Research Committee of the Society from 1926 to 1930 and its chairman in 1929-1930. At the time of his death he was a representative of the A.S.M.E. on the Division of Engineering of the National Research Council and an associate member of the Executive Committee of the Iron and Steel Division of the Society.

Mr. Streeter was co-author with Dr. P. V. Faragher of a paper entitled "Aluminum and Its Light Alloys," presented at the Milwaukee, 1925, meeting of the A.S.M.E. and revised and published by the authors in pamphlet form in 1927. He presented a paper on "The Rolling and Extrusion of Aluminum and Its Alloys" at the December, 1932, meeting of the Society.

Surviving him are his widow, Louise (Rapalje) Streeter, whom he married in 1907, and four daughters, Mrs. Paul Franklin Eves, Oakmont, Pa., Jean Rapalje Streeter, Barbara Frear Streeter, and Anne Richards Streeter. He also leaves one son, Robert Leroy Streeter, Jr., of Oakmont, Pa.—Curtis C. Myers.[1]

ARTHUR CHARLES TAGGE (1870–1932)

Arthur Charles Tagge, a member of the Board of Directors of the Canada Cement Company Limited, Montreal, Quebec, Canada, died at his home in Monroe, Mich., on December 6, 1932. He retired as president of the company in 1930.

Mr. Tagge was born on January 31, 1870, in Ann Arbor, Mich. He received a B.S. degree from the Engineering Department of the University of Michigan at the age of twenty-seven, several years of teaching in Monroe having delayed the completion of his course.

After securing some drafting and shop experience at the Frontier Iron Works in Detroit and with Bissell, Dodge and Erner in Toledo, he took a position as draftsman with the Link-Belt Machinery Company, Chicago, where he worked until January, 1900. For the next fifteen months he was employed by the Osborn Engineering Company, Cleveland, in the design and construction of a cement plant for the Peninsula Portland Cement Company, Woodstock, Mich. He was mechanical engineer and mill superintendent for this plant in 1901 and 1902.

Mr. Tagge then became associated with the group which formed the International Portland Cement Company, of Ottawa, Ontario, and remained with that company for three years in the capacity of engineer and operating advisor. He was next connected, in a similar capacity, with the Western Canada Cement & Coal Co. A few years later, when the Canada Cement Company Limited was formed as a consolidation of the principal cement plants of the Dominion, Mr. Tagge became its general superintendent. The consolidated company operated a chain of 14 plants, then the most extensive cement mill operation of any single corporation, and Mr. Tagge distinguished himself in the work of standardizing and securing greater economy of operation.

He subsequently became general manager and later vice-president and director of the company. After six years as vice-president he was elected to the presidency in 1927 and in that capacity directed the very outstanding program of rehabilitation which the company carried out, entirely rebuilding a number of its principal manufacturing units and modernizing others.

Since his retirement in 1930 Mr. Tagge had made his home at Monroe, where he enjoyed his flowers and art treasures. His death was the result of a heart ailment with which he had suffered for some time.

During his active years in the cement industry Mr. Tagge held many important committee appointments in the Portland Cement Association, the most outstanding of which was his long chairmanship of the committee on accident prevention and insurance. A year after his retirement he was unanimously elected an honorary member

[1] Buffalo, N.Y. Mem. A.S.M.E.

of the Portland Cement Association, being one of only three men who have ever been accorded this distinguished recognition. Mr. Tagge became a member of the A.S.M.E. in 1901.

JOHN EVAN TAYLOR (1889–1933)

John Evan Taylor, since 1927 connected with the Central Railroad Company of New Jersey, died on May 31, 1933. He was at first estimator in the Motive Power Department at Jersey City, N.J., and at the time of his death locomotive draftsman in the office of the mechanical engineer at Elizabethport, N.J.

Mr. Taylor was born at Picton, Ontario, Canada, on August 25, 1889. He was graduated from the College of Applied Science of Syracuse University with an M.E. degree in 1912 and during the next two years was employed at the Depew Shops of the New York Central Railroad, Depew, N.Y. For several months in 1914 he was with the chief mechanical engineer of the road, in charge of locomotive road tests and locomotive design. He was then transferred to the DeWitt Engine House at East Syracuse as assistant foreman.

From 1915 to 1918 Mr. Taylor was connected with the Locomotive Superheater Company, New York, as engineering inspector in charge of the manufacture, installation, and inspection of locomotive superheaters at various plants of the American Locomotive Company, Lima Locomotive Works, and Canadian Locomotive Works.

Mr. Taylor enlisted in the United States Army in April, 1918, and served for more than a year in France with the Transportation Corps. After his discharge in 1919 he returned to the employ of the Locomotive Superheater Company, working in the New York office on superheater design. He left there in 1922 to take a position in the Service Department of the Simmons-Boardman Publishing Company, New York, where he remained for several years before entering the employ of the Central Railroad Company of New Jersey.

Mr. Taylor became a junior member of the A.S.M.E. in 1914 and an associate-member in 1921.

LEON IRVING THOMAS (1884–1933)

Leon Irving Thomas, vice-president and secretary of the Gage Publishing Company and editor of *Electrical Manufacturing*, New York, died on July 30, 1933, at his summer home at Pocasset, Mass.

Mr. Thomas was born on October 2, 1884, at Brockton, Mass., the son of Myron Franklin and Mary M. (Tallman) Thomas. He attended the Brockton High School and took the electrical engineering course at Worcester Polytechnic Institute, receiving his B.S. degree in 1907.

During the first two years after his graduation Mr. Thomas was personal laboratory assistant to Dr. Lee De Forest, wireless telephone and telegraph expert and inventor, and in 1909 was appointed factory superintendent of the De Forest Radio Telephone Company in Newark, N.J. The following year, after working for a few months on wireless design and construction with Mr. Fritz Lowenstine, consulting engineer, he took a position in the traffic department of the American Telephone & Telegraph Co. He was associated with this company for two years, the most of which time he spent as assistant traffic chief in the Chicago office.

Mr. Thomas entered the publishing field in 1912, when he joined the A. W. Shaw Company, Chicago, as associate editor of *Factory*. In 1915 he became managing editor of that magazine and in 1925 was appointed editor of *Industrial Power*. Meanwhile he had also been an advisory editor of *System* and was an editor and co-author of the "Library of Factory Management" and of several business administration textbooks for La Salle Extension University. In 1929 he left Chicago and joined the Gage Publishing Company as managing and engineering editor. He had been vice-president and secretary and editor of *Electrical Manufacturing* for the past three years.

Mr. Thomas had been an associate-member of the A.S.M.E. since 1916. He also belonged to the Society of Industrial Engineers, the New York Electrical Society, Engineers' Club, New York, and the Sigma Alpha Epsilon fraternity. He is survived by his widow, Grace (Howe) Thomas, and by two daughters, Barbara and Priscilla Thomas. His home was in Montclair, N.J.

SIR HENRY WORTH THORNTON (1871–1933)

Sir Henry Worth Thornton, who resigned as president of the Canadian National Railways in 1932, died at the Doctor's Hospital, New York, N.Y., on March 14, 1933. He was internationally known as a railroad executive, having been associated with leading roads in the United States, Canada, and England, and a prominent figure in transportation work during the World War.

Sir Henry was born at Logansport, Ind., on November 6, 1871, the son of Henry Clay Thornton and Millamenta C. (Worth) Thornton. He was sent to St. Paul's School, Concord, N.H., to prepare for college, and was graduated from the University of Pennsylvania as a civil engineer in 1894, having not only made a brilliant scholastic record but also taking a leading part in athletics, playing football and breaking records in various track meet events. He was a tall, heavily built man, and throughout his life continued to enjoy sports, playing tennis, golf, and handball, and riding and hunting.

He coached the Vanderbilt University football squad during the autumn after his graduation and then found work as a draftsman in the office of the chief engineer of the Southwest System of Pennsylvania Railroad lines west of Pittsburgh. The next year he was made assistant engineer of construction of the Pennsylvania's Cleveland and Marietta line and subsequently was supervisor of the yards at Columbus, assistant engineer of the Cincinnati Division, engineer of maintenance of way, Erie & Ashtabula Division, and superintendent of the Cleveland, Akron & Columbus R.R. Early in 1911 he was sent to New York, and later in the year was made general superintendent of the Long Island Railroad.

When Mr. Thornton had been with the Long Island Railroad for about three years he was invited to England to take over the management of the Great Eastern Railway, in an endeavor to build up its business. Although the appointment of some one outside England was regarded with disapproval in many quarters, his ability to handle the task was soon recognized and he was later made a member of the executive committee of railroad managers which, under the direction of the government, controlled and worked the English railways during the World War. He was made an honorary lieutenant colonel in the Royal Engineers when he entered upon war work, and in 1915 was promoted to the rank of colonel and placed in charge of cross-Channel transportation. His next assignment was that of deputy director of inland water transportation in charge of navigation in northern France, Egypt, and Mesopotamia. In 1917 he was promoted to assistant director general of movements of railways, with headquarters in Paris, and received the rank of brigadier general. During the last year of the war he was inspector general of transportation with the rank of major general.

Mr. Thornton had become a British citizen during the war and in 1919 he was knighted by the King. France made him a companion of the Legion of Honor, Belgium conferred upon him the cross of an officer of the Order of Leopold, and his native country gave him the Distinguished Service Medal. He was given a D.Sc. degree by the University of Pennsylvania in 1923.

In the years immediately following the war, in addition to railway reorganization, Sir Henry served on a committee to investigate operations and financial conditions of the Metropolitan Water Board of the City of London.

His return to this continent came in 1922 when he was asked to become president and chairman of the board of the Canadian National Railways. He spent several months traveling over the various lines, moved the headquarters from Toronto to Montreal, and at the end of the first year showed a net operating income of more than three million dollars, as against a deficit of more than eleven million on the previous year's income account.

The improvement continued year by year until, in 1928, net operating revenues reached 60 millions—sufficient to pay the interest on all the securities of the system held by the public. Sir Henry's greatest service to the Canadian National Railways, perhaps, was his welding of the several separate constituent companies into a unified and harmonious system. There is no more difficult managerial problem than that of making two or three organizations into one, with the conflicts of precedence which inevitably follow. This task Sir Henry accomplished in a remarkably short space of time and in such a way that the morale of the staff not only did not suffer, but was greatly improved, and the C.N.R. as a result of his policies soon drew attention by the enviable esprit de corps shown by all ranks of its employees.

Since his resignation in 1932 Sir Henry had spent the most of his time in Florida and Cuba, endeavoring to restore his failing health.

In 1901 Mr. Thornton married Virginia Dike Blair, of New Castle, Pa., from whom he was divorced in 1926. He is survived by their two children, Anna Blair and James Worth Thornton, and by his second wife, Martha (Watriss) Thornton.

Sir Henry became a member of the A.S.M.E. in 1917.

JOSEPH JAMES TYNAN (1871–1933)

Joseph James Tynan, who died in San Francisco, Calif., on June 6, 1933, was born in County Tyrone, Ireland, on October 8, 1871, a son of Robert and Elizabeth Tynan. He had the equivalent of a grammar and high school education in that country and after coming to the United States in 1890 attended the Spring Garden Institute,

Philadelphia, Pa., and served an apprenticeship at the Philadelphia Engineering Works. He was then employed as a journeyman at the Wm. Cramp & Sons Ship & Engine Building Co., Philadelphia, subsequently rising to the position of general superintendent. He became a naturalized citizen in 1896.

Mr. Tynan had been with the Bethlehem Shipbuilding Corporation, Ltd., San Francisco, since 1906, becoming vice-president in 1909. He was the builder of the Columbia River-Longview Bridge, and among his other achievements was the building of ten submarines in five months' time for the British Admiralty in Montreal, Canada, during the World War. He was awarded a gold medal by Charles M. Schwab, then director of the Emergency Fleet Corporation, for merchant vessels delivered during the War.

Mr. Tynan became a member of the A.S.M.E. in 1922 and belonged to a number of clubs in San Francisco. He was a member of the Board of Police Commissioners of San Francisco in 1930 and had been a member of the Board of Harbor Commissioners of the State of California since 1931.

Surviving Mr. Tynan are his widow, Ruth (Williams) Tynan, whom he married in 1922, and their daughter, Ruth P. Tynan, and two children by a former marriage, Joseph J. Tynan, Jr., and Mrs. Margot (J. Curtis) Taylor. His first wife, whom he married in 1902, was Margaret McGinty.

WILLIAM CAWTHORNE UNWIN (1838–1933)

In 1890, when the Cataract Construction Company began laying plans for the large-scale development of power at Niagara Falls, the problem of the engineering features involved was referred to an international commission consisting of Sir William Thomson (Lord Kelvin), president, Dr. Coleman Sellers, Prof. E. Mascart, of Paris, France, Col. Theodore Turrettini, of Geneva, Switzerland, and Prof. William Cawthorne Unwin, secretary.

The modern age to which the younger engineers now practising belong finds it hard to understand the uncertainties that faced this commission. For the generation of electricity in large amounts by hydraulic turbines, the transmission of the current generated over long distances, its distribution and use in mechanical and chemical industries, and hundreds of technical problems such as the relative advantages of alternating and direct current were being attacked in pioneer fashion. Nor was it certain that the development should be an electrical one. Electricity was new; it had never been generated centrally in large quantities for distribution over long distances to points where it could be used. While lighting by electricity had been broadly introduced, industrial power was still largely secured by mechanical means. The Evershed project, upon which the Niagara scheme was based, was entirely hydraulic and contemplated a community of industries deriving their power from individual hydraulic turbines fed from a common canal—the use of power at the site. Mechanical (telodynamic) and pneumatic transmission systems were in a position to contend for consideration with hydraulic, steam, and gas methods of transmission and distribution. Moreover, the machinery and auxiliary apparatus, as well as the construction of the hydraulic details, were of a scale heretofore untried, so that the stakes, engineering as well as financial, were high and hazardous. Professor Unwin's book, "On the Development and Transmission of Power From Central Stations," comprising the Howard lectures delivered before the Society of Arts in 1893, contains his reflections on the methods in use and proposed at that time, and is indicative of the state of the art when the Niagara development was under consideration.

Professor Unwin, as secretary of the international commission conducting, on behalf of the Cataract Construction Company, the contest for the best solution of the Niagara problem, received the great variety of plans and proposals that were offered and, in 1893, wrote the report in which they were analyzed and appraised. It was this service that brought him into most intimate relationship with engineering on this continent and won for him, in 1898, honorary membership in The American Society of Mechanical Engineers.

Writing 25 years later of the work of the Commission, Professor Unwin said: "The projects were of extremely varied character. Hydraulic turbines of impulse and reaction types of from 2000 to 10,000 horsepower; distribution by electricity, by compressed air, and by wire ropes. Most of the electrical schemes proposed direct-current production and distribution with varying current and constant voltage. Messrs. Siemens proposed constant direct current at varying voltage; Professor Forbes only proposed alternating current. From the first, electrical distribution of the power was in favor, but was not definitely decided upon until May, 1893." Of the significance of the project itself he wrote: "Civilization may be measured by the degree in which human labor is replaced by power derived from natural sources of energy. . . . At Niagara some 7,000,000 horsepower which might have been available for industrial purposes was being wasted. The conditions were favorable for utilization in other respects, for the fall was a high one and the Great Lakes from which the water flows act as reservoirs equalizing the supply. The installation has been carried out with complete success, and it is the first large-scale undertaking of the kind. It has been the parent of many others in different parts of the world, or at any rate showed the way to others to achieve a similar success. But the risk of misadventure in so novel and complicated an enterprise in the days when it was carried out was great, and that no serious mistake of judgment or calculation occurred witnesses to the remarkable care with which the preliminary investigations and discussions were conducted."

The Niagara decisions were momentous ones and have had their effect on the industrial and engineering development not only of the district immediately surrounding Niagara Falls and the city of Buffalo, but of the rest of the country and the world at large. For success at Niagara led the way to future developments, and gave a decisive answer to the major policies affecting the generation, transmission, and distribution of hydro power for years to come. It made feasible the development of sources of hydro power that are remotely located with respect to centers of population and ultimate use. It raised social, economic, and political problems in controversies over which this country in particular is still enmeshed. While many engineers and financiers were engaged in the historic events of those days, Professor Unwin played an important part that was in keeping with his character, training, and methods of work. For Professor Unwin was primarily a teacher and an investigator who analyzed and rationalized engineering science and practice and put the results of his studies and experiences in such a form as to make them available for all men, in all lands, and for all time.

William Cawthorne Unwin was born on December 12, 1838, at Coggeshall, Essex, the son of Dr. W. J. Unwin, principal of Homerton College. Following his education at the City of London School, he was, by good fortune, associated in Manchester with Sir William Fairbairn, whose contributions to engineering literature and research have kept his memory alive. Fairbairn found in the young Unwin a man of similar aptitude and zeal for research and rational statement of empirical practices. Together they conducted a series of tests on the properties of steam, the strength of plate girders, and the collapse of tubes. By 1861 Unwin was manager of an engineering works manufacturing the vortex hydraulic turbine, where he acquired that personal contact with men and engineering practice without which the research worker and scholar are likely to remain impractical theorists.

Professor Unwin's career as a teacher began in 1868 with an appointment as professor at the Royal School of Naval Architecture and Marine Engineering in South Kensington. Four years later he was made professor of hydraulic engineering at the Royal Indian Engineering College, Coopers Hill, and in 1884 became the first professor of engineering at the Central Technical College of the City and Guilds of London. In the fulfilment of this office he rendered notable services to the school and to engineering education, and continued with his researches on the steam engine and the mechanics of materials.

No student of engineering can have escaped the influence of Professor Unwin's work. If his most notable textbooks, "The Elements of Machine Design," "Wrought Iron Bridges and Roofs," and "Testing Materials of Construction," are today not actually studied, for indeed they have been replaced by more modern texts, no later books on any of these subjects, or upon hydraulics, can be found without reference to his work, or without a formula bearing his name. For it was his especial ability to study the underlying theory, the practice, and the experimental researches in connection with these subjects and to evolve fundamental formulas upon which engineers and designers could place reliance. While no great engineering structure stands today called by his name because his genius brought it into being as concrete evidence of a man's ability to perform spectacular engineering feats, it would not be an exaggeration to say that his work influenced those who have erected such monuments and provided at least some of the tools wherewith the designer worked.

One other factor of Professor Unwin's interests deserves comment in this brief review because of the importance that engineers in this country are today attaching to it. This is his work for the advancement of the professional status of the engineer and the esteem in which the profession is held. He was president, in 1911, of the Institution of Civil Engineers, and, in 1915, of the Institution of Mechanical Engineers. To both of these institutions he devoted much time and labor in the development of examinations for membership. His addresses before these institutions will repay careful reading, particularly those parts that refer to engineering education, research, the induction of young engineers into the profession, and the value of engineering societies to their members. On this final topic a paragraph from his 1915 address is worthy of quotation. He said: "The Institution as a corporate body of men of similar avocations is a

source of strength to all its members. It gives publicity to the work of engineers, which is essential in creating the estimation in which they are held by people outside the profession. It is able to secure facilities and gifts, and it elicits and records information of great value which would otherwise remain private property. One has heard of persons who think that because they cannot attend the meetings in London, and have no time to study the Proceedings, they can get no benefit from the Institution. The case is rather an extreme one. Bacon said that every man is a debtor to his profession, and it follows that he has a duty to support whatever benefits it. Most of us subscribe to objects from which we expect no direct recompense, and few engineers could say that they receive no indirect benefit from the work done here. Knowledge diffuses itself in various channels, and improvements in industrial methods bring advantage to many who do not know where they originate."

Professor Unwin died on March 17, 1933. In an appreciative obituary in *Engineering* it is said: "In him has passed away a great teacher, a sound engineer, and a fine character."—G. A. S.

ROBERTO JUAN URIE (1886–1933)

Roberto Juan Urie, technical director of illuminating, electrical, and mechanical engineering for the city of Tucumán, Argentina, South America, died on April 30, 1933. He was born at Ayacucho, Province of Buenos Aires, Argentina, on February 6, 1886. He secured his elementary education at English and North American schools in the city of Buenos Aires, and in 1903 completed the national college course of secondary education in that city. In 1910 he completed the first year in civil engineering at the Buenos Aires University. From 1911 to 1915 he studied electrical and mechanical engineering at the Royal Technical College, Glasgow, Scotland, acquiring practical experience at the same time in various departments of the North British Locomotive Company in Glasgow. During the next year he was assistant in the technical office of the Argentina State Railways Works at Tafi Viejo. He was then appointed "Director Técnico de Alumbrado e Instalaciones Eléctricas y Mécanicas," Tucumán, Argentina.

Mr. Urie became an associate-member of the A.S.M.E. in 1930. He also belonged to the American Institute of Electrical Engineers and the Asociacion Argentina de Electro-Técnicos.

ANTON E. WEINGARTNER (1870–1933)

Anton E. Weingartner, designing engineer with the American Engineering Company, Philadelphia, Pa., died on February 5, 1933.

He was born in Budapest, Hungary, on September 12, 1870, and attended the Arad, Hungary, high school and the technical institute in Funfkirohen. After his graduation as a construction engineer in 1889 he was draftsman and assistant to the master mechanic of the Edmond Hendl Brass Foundry and General Machine Shop in Arad for a year, and spent the following year as junior foreman with the National Harvester Company, in Budapest, manufacturing portable engines, locomobiles, and agricultural machinery.

Mr. Weingartner came to the United States in 1891 and for 26 years was associated with the Bethlehem Iron Company and Bethlehem Steel Company, Bethlehem, Pa. During the first ten years his work included drafting for plant construction, designing heavy machine tools and other plant equipment, and reconstructing blast furnaces. He spent three years in designing and building hydraulic press equipment and power machinery and two in rolling mill engineering, in the staybolt mill and crucible steel plant, after which, in the summer of 1905, he visited a number of countries abroad, investigating steel and rolling mill practice in general and the manufacture of steel products. After his return he assisted in the reconstruction of the Saucon Plant of the Bethlehem Steel Company, and then for nearly ten years was engaged in plant construction and ordnance equipment work. In 1916 he entered upon sales engineering work for the company, in the heavy machinery department, becoming sales agent in charge of the machinery department the following year.

Mr. Weingartner terminated his connection with the Bethlehem Steel Company in May, 1918, and for a short time carried on a machinery business of his own in Bethlehem. Since then he had been with the American Engineering Company.

Mr. Weingartner was credited with the design of the electro-hydraulic steering gears used on the navy airplane carriers, the *Lexington* and the *Saratoga*. He also designed and patented a device intended to allow men imprisoned in a sunken submarine to make their escape without assistance and was the inventor of a water motor for washing machines and patented numerous inventions in connection with stoker and marine equipment.

Mr. Weingartner became a member of the A.S.M.E. in 1923.

WILLIAM WALLACE WEST (1877–1933)

William Wallace West, who died at his home in Elyria, Ohio, on May 20, 1933, was born at Johnstown, Pa., on July 13, 1877, the son of Emery and Amanda (Berry) West. He attended the Johnstown public schools and the Rowe Business College there, and in later years took correspondence, Y.M.C.A., and library courses in technical subjects.

His first employment, at the age of 17, was with the Cambria Steel Company, Johnstown, and he continued with that company until 1910. He then spent four years with the Colorado Fuel & Iron Co., Pueblo, three with the Crucible Steel Company, Midland, Pa., and two with the Wilmington Steel Company, Wilmington, Del. Since 1919 he had been with The Elyria Iron & Steel Co., Elyria, Ohio, as general superintendent for the past 12 years.

Mr. West became an associate member of the A.S.M.E. in 1922. He served in Company H of the 5th Pennsylvania Volunteer Infantry in the Spanish-American War and belonged to the Veterans of Foreign Wars and the Spanish-American War Veterans. He also was a member of the Sons of the American Revolution and of the Elks and the Masonic fraternity. He was director of the Elyria Chamber of Commerce and attended the Methodist Church.

Surviving Mr. West are his widow, Bertha M. (Patterson) West, whom he married in 1899, and two daughters.

CLARENCE E. WHITNEY (1869–1933)

Clarence E. Whitney, a leading manufacturer of Hartford, Conn., died suddenly of a heart attack at his home in that city on January 22, 1933. He was president of the Whitney Manufacturing Company, the Hartford Faience Company, and the Liberty Street Realty Company, and secretary-treasurer of the Hanson-Whitney Machine Company. He started the Whitney Manufacturing Company in 1896, taking over the small Woodruff Manufacturing Company at Colt's West Armory, and helped to found all of these companies. He was also a director of the S.K.F. Ball Bearing Company, for some time located in Hartford.

Mr. Whitney was born in Hartford on November 26, 1869, the son of Amos Whitney, one of the founders of the Pratt & Whitney Co. He attended the Hartford public schools and at the age of eighteen entered the Massachusetts Institute of Technology, where he studied mechanical engineering for three years. He worked in the Pratt & Whitney shops for a year between the completion of his high school work and entrance to M.I.T. and again during the summer of 1890. In the fall of that year, instead of returning to college, he took advantage of an opportunity offered him by his father to spend a year in Europe. He went to Germany and from then until July, 1891, worked in the designing department of Ludwig Loewe & Co., in Berlin, while studying industrial methods there and at other plants in Germany. He then returned to Hartford and until 1896 was part of the Pratt & Whitney organization, working in many departments and in the office and on the road for the company.

The Whitney Manufacturing Company at first conducted tests on the use of roller bearings for bicycles, for the Hyatt Roller Bearing Company. This work led Mr. Whitney into the manufacture of bicycle chains, and later the manufacture of automobile chains was undertaken. It was not long before larger quarters were necessary and the company built its own plant. Under Mr. Whitney's direction new types of chains were developed as required by the automotive industry, in which the company has taken a leading place.

In 1894 at the request of Eugene Atwood, Mr. Whitney helped to form the Atwood Faience Company, which made slow progress at first and from which Mr. Atwood withdrew after a few years. It was reorganized as the Hartford Faience Company and has continued the manufacture of this special type of earthenware.

The Hanson-Whitney Machine Company was organized in 1919 for the manufacture of taps, gages, and hobs.

Mr. Whitney was a supporter of the principle of the open shop and devoted a considerable share of his time and money to the cause. He constantly sought to increase the efficiency of his plants and was recognized and respected as an energetic and courageous leader, devoted to the maintenance of Hartford's prestige in industry.

Mr. Whitney was very active in the National Metal Trades Association and was affiliated with the National Association of Manufacturers and the Manufacturers' Association of Hartford County for many years. He was instrumental in forming the Employers' Association of Hartford and was its first president. During the World War he was a leader in Liberty Loan and Red Cross campaigns and shortly afterward conducted a drive for funds for new buildings for the Newington (Conn.) Home for Crippled Children.

Mr. Whitney became a member of the A.S.M.E. in 1911. He was a member of the Question Club, a group of 75 leading manufacturers in the United States and Canada, the Engineers' Club, New York,

the Masonic and Theta Delta Chi fraternities, Sons of the American Revolution, and a number of clubs in Hartford and vicinity.

Surviving Mr. Whitney are his widow, Nellie (Hurlburt) Whitney, whom he married in 1900, and four children, Mrs. Robert P. Stevens, Plainfield, N.J., Mrs. John L. Hawkinson and Lawrence Amos Whitney, Hartford, and Winthrop Hurlburt Whitney, Los Angeles, Calif.

FLOYD ELLIS WILDER (1878–1933)

Floyd Ellis Wilder, chairman of the Executive Committee of the Printing Industries Division of the A.S.M.E., died on August 12, 1933, at his home in New York, N.Y., after an illness of several weeks. Mr. Wilder, who became a member of the Society in 1930, had been active in the work of the Division for several years, assisting in preparing the programs for its annual conferences and furthering its activities in other ways. He had been chairman of the Executive Committee during the past year.

Mr. Wilder was born at Wacousta, Mich., on September 27, 1878, the son of Jason and Louisa (Tyler) Wilder, the latter a descendant of President Tyler. He was graduated from the Wacousta High School and studied mathematics and mechanical drawing under private tuition for about a year. He then obtained a position as general assistant to the plant engineer of the Lake Lumber Company, Lake, Mich., working in different mills for a year prior to the Spanish-American War. He was in service during the War and subsequently in the Philippines and then was commissioned a captain in the 14th Infantry of the New York National Guard.

In 1900 Mr. Wilder went to New York where he learned the printing trade in the plant of the Robert L. Stilson Company. In 1907 he entered the employ of the P. F. Collier & Son Company, New York, as assistant superintendent. In 1909 he was made superintendent, and a year later director of production and assistant to the president, which position he held until 1925. During that period he designed and supervised the entire manufacture of many pieces of machinery. Among these was a complete conveying system for handling roll paper on the ceiling, using a monorail track. He helped to design several improvements for the Hoe web press and designed and manufactured the equipment necessary for instantaneous splicing of paper rolls used on web presses. Other equipment designed by Mr. Wilder include an automatic feed for book smashing machines, an attachment for Smythe casemaking machines for making flexible cases for books, and an automatic metal furnace for use in backing up electrotypes.

After leaving the Collier company Mr. Wilder was associated briefly with the Conde Nast publications and in 1925 became production manager in charge of printing for the International Magazine Company, Inc., New York, publishers of the Hearst magazines.

Mr. Wilder was one of the organizers and a former vice-president of the New York Club of Printing House Craftsmen, now the National Association of Printing House Craftsmen, and was secretary of the New York Employing Printers Association, Inc.

Surviving Mr. Wilder are his widow, Martha (Middleton) Wilder, and a son by a previous marriage, Robert Wilder.

AAGE E. WINCKLER (1881–1933)

Aage E. Winckler, president and chief engineer of the Winckler Engineering Company, Milwaukee, Wis., died suddenly at his home in that city on January 25, 1933.

Mr. Winckler was born in Copenhagen, Denmark, on November 7, 1881, a son of Oscar and Laura Winckler, and received his early education there. After a year's apprenticeship in a Copenhagen factory he entered the technical high school in Ilmenau, Germany. For his university training he went to Zurich, Switzerland, and later to Karlsruhe, Germany, where he secured a mechanical engineering degree in 1904.

For a short period after his graduation Mr. Winckler studied factory organization and automotive manufacturing at the Daimler Motoren Gesellshaft (manufacturers of the Mercedes car), Stuttgart, Germany, and Rheinische Automobilwerke, Aachen, Germany. In 1905 he returned to Copenhagen and became mechanical engineer for the Dansk-Automobil-Selskab, organizing automobile mail and passenger routes and supervising the construction of buses for use on them at several automobile plants in France. From 1906 to 1909 he was engaged in developing a taxicab service for Copenhagen. Under his supervision the cabs were designed and built at Uhren Cotthaus & Co., Cologne, Germany, and the service was introduced and managed by the Aktieselskabet Automobil Korsel Kompagniet, of which he was president and general manager.

Mr. Winckler came to the United States in 1909 and during the next few years handled development work on a rotary airplane motor and racing cars, as chief engineer of the automobile department of the J. I. Case Threshing Machine Co., Racine, Wis. The Lorenz Motors Company, Racine, which was organized in 1912 and of which he was president, carried on special development work on motors and automobiles for the Case company during the following year, and Mr. Winckler again served the company as chief engineer from 1913 to 1916.

In 1916 Mr. Winckler visited Europe and after returning to this country spent half a year with the Pittsburgh Model Engine Company, after which he returned to the Lorenz Motors Company to take charge of development and research work on aeronautical motors for the United States Navy.

In 1920 the Lorenz Motors Company was reorganized under the name of The Winckler Engineering Company, with Mr. Winckler as president. The company was located in Milwaukee, Wis., and specialized in the development of small, light-weight, high-speed compression-ignition automotive motors and direct, synchronized, airless-fuel-injection systems for them. He held patents on his work in the United States and a number of foreign countries.

Mr. Winckler became a member of the A.S.M.E. in 1930 and also belonged to the Society of Automotive Engineers.

He is survived by his widow, Else (van Norde) Winckler, whom he married in 1905, and by two children, Per van Norde and Margot van Norde Winckler.

ROBERT WOOD (1874–1932)

Robert Wood, consulting engineer, specializing in problems relating to superheat, locomotives, and internal-combustion engines, died suddenly of heart failure on November 24, 1932, at Newlands, Garwick, Isle of Man, England, where he had lived since his retirement in 1921. He is survived by his widow, Pearl F. Wood.

Mr. Wood was born on September 8, 1874, in Edinburgh, Scotland. After a five-year apprenticeship he was employed by the firm of Hick Hargreaves & Co., Ltd., of Bolton, Lancashire, England, to install and supervise plants for the company. He also was engaged to supervise the installation of bleaching machinery for the firm of Jackson Bros. Wharf Foundry, Bolton, which likewise dealt in paper manufacturing and rolling machines.

At the age of thirty, Mr. Wood opened an office in Bolton for consulting engineering, his specialty being the lubrication of steam-engine cylinders, and the firm of Robert Wood & Co., Ltd., was established to manufacture his patent lubricators. After a visit to the United States in 1908 he decided to return to this country to live. In 1911 he gave up his Bolton office and went to Montreal, Canada, to complete some consulting work for the Robert Mitchell Co., Ltd. Two years later he allied himself with the Nathan Manufacturing Company, New York, as works manager. He specialized on the locomotive from a lubrication standpoint and also designed the company's mechanical equipment, such as balance valves, coal sprinklers, vacuum breakers, flange oilers, pumps, injectors, etc.

In 1919 he took a similar position with the Fletcher Works, Inc., of Philadelphia, Pa., where he remained for about two years before going to Garwick. Since his retirement he had been experimenting with a device designed to utilize superheated steam in internal-combustion engines. This superheater, which can be fitted to any automobile engine and is also suitable for stationary and marine engines of the internal-combustion type, is said to effect a considerable saving in running costs, increase engine efficiency, and eliminate the necessity for decarbonizing. Mr. Wood had secured patents on the superheater in Great Britain and the United States.

Mr. Wood became a member of the A.S.M.E. in 1915 and also belonged to the Masonic fraternity.

HERBERT WALDO YORK (1864–1932)[1]

Herbert Waldo York, who died on November 21, 1932, was born at West Roxbury, Mass., on February 18, 1864. He was the son of John and Julia A. (Hinckley) York. He attended the public schools and was graduated from high school at Washington, D.C., where his father, who came to the United States from England in 1850, was employed in the office of the Commissioner of Customs, United States Treasury Department.

From November, 1882, to November, 1885, Mr. York served as assistant to Charles E. Emery, consulting engineer for the United States Revenue Marine (now United States Coast Guard Service). As such, his duties involved work on plans and specifications for new machinery and alterations and repairs to old equipment.

It was urged upon him at this time that he should acquire the sea

[1] Extracted from memoir prepared by John Barnard, Lt-Com. (Retired), U.S.N.R., New York, N.Y., for the Memoirs of the American Society of Civil Engineers.

service prerequisite to an examination for a sea-going Marine Engineer, and it was arranged that he should go out in the Engineer's Department of that famous old United States Revenue Cutter, *Bear*, then being sent from the Atlantic Coast to the Pacific for the beginning of her internationally known services in the Behring Sea and the Arctic. Upon this ship's arrival in San Francisco, Calif., in the spring of 1886, Mr. York's sea service was still inadequate for promotion, and he continued on the ship for her first summer's patrol in the Arctic. Upon the *Bear's* return to San Francisco, Mr. York successfully passed the examinations for, and was commissioned as, a Second Assistant Engineer in the United States Revenue Marine.

From this service he resigned in December, 1886, to return East, where he entered the employ of the New York Steam Company, of New York, N.Y., as assistant engineer in charge of building service connections, repairs to street mains, tests, etc., for one year, until November, 1887. He then engaged in private consulting work until July, 1889, when he became assistant superintendent of the Phoenix Construction Company, on the construction of electrical subways for The Consolidated Telegraph & Electric Subway Co. Mr. York was in charge of all construction south of Twenty-Third Street, New York, and, to January, 1890, had built about 110 miles of ducts. He then again engaged in private work until May, 1890, when he was employed for about six months as superintendent on the construction of new shops for The Yale & Towne Manufacturing Co., Stamford, Conn.

In November, 1890, Mr. York joined The United Electric Light & Power Co., from which were run the first alternating-current circuits in the city of New York. This company bought the old Brush Electric Illuminating Company and with it The United States Illuminating Company was merged. He joined the company as assistant engineer and in May, 1892, was made chief engineer in charge of operation and maintenance of all stations and of the design, construction, and operation of a new station on East Twenty-Eighth Street and the East River.

In these days alternating-current generating plants have replaced the earlier and simpler direct-current stations, and much is now commonplace which then presented problems. Although today these problems are largely forgotten, they were serious in the older days and had to be solved if success was to result. Accordingly, it is interesting to hear from the first electrical engineer of the Twenty-Eighth Street Station, of some of the devices originating with Herbert York for synchronizing those reciprocating engines. The first of these devices consisted of little whistles blowing at a given point of the stroke so that as the engineer brought up to speed an engine about to be cut in, the operator at the switchboard would hear its whistle gradually come into unison with the others. This method Mr. York later replaced by a control on the valve gear which automatically accomplished synchronization. On this, he obtained a patent which was acquired by the Westinghouse Company.

On April 1, 1903, Mr. York became chief engineer of the United Lead Company, holding that position until May 1, 1905, when he accepted the position of engineering manager for The J. G. White Engineering Corp., which had undertaken the function of consulting engineers for the American Smelting & Refining Co., coupled with an understanding that Mr. York personally should handle its work. The attempt, however, proved too taxing, and on June 30, 1906, he transferred to the American Smelting & Refining Co. as consulting engineer, in which capacity he continued until his retirement on April 1, 1930.

Mr. York was elected a Junior of the American Society of Civil Engineers in 1888 and a Member in 1896. Shortly after his election as a Junior he submitted a paper on "The Twenty-Eighth Street Central Station of the United Electric Light & Power Company," which was accepted for publication.[2] As a result, in 1896, he was honored by the society's Collingwood Prize for Juniors. He became a member of the A.S.M.E. in 1894, and also belonged to The Franklin Institute and the American Institute of Mining and Metallurgical Engineers. His clubs included the Engineers', New York Yacht, and Bankers, New York.

In 1900 he joined the First Battalion of the Naval Militia of New York State as Lieutenant (Junior Engineer), serving as Assistant Engineer of that Battalion, and transferring in 1904 to the duties of Gunnery Officer. In these positions he continued until 1911, when he was promoted to the rank of Lieutenant Commander, and served as Ordnance Officer on the Staff of the Commanding Officer of the New York Naval Militia. In 1917 he wished to enter active service in the World War and was keenly disappointed when the Navy Department deemed him too important a factor in munitions production. After the war he continued his active efforts in the State Naval Service until he was retired at his own request in 1927, and was transferred to the State Reserve List with the rank of Commander. He also held at his death the rank of Lieutenant-Commander on the Honorary Retired List of the United States Naval Reserve.

Mr. York was married on June 18, 1930, to Mrs. Jane Dolen-Lenz, who survives him.

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[2] Transactions, A.S.C.E., vol. xxxv (1896), p. 429.

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Bozeman — Montana State College

Nebraska

Lincoln.....University of Nebraska
Omaha.....Public Library

Nevada

Reno.....University of Nevada Library

New Hampshire

Durham.....University of New Hampshire

New Jersey

Bayonne.....Free Public Library
Camden.....Free Public Library
Elizabeth.....Free Public Library
Hoboken.....Stevens Institute of Technology
Jersey City.....Free Public Library
Newark.....Newark College of Engineering
Free Public Library
New Brunswick...Rutgers University
Paterson.....Free Public Library
Princeton.....Princeton University
Trenton.....Free Public Library

New York

Albany.....New York State Library
Brooklyn.....Polytechnic Institute
Pratt Institute
Brooklyn Public Library
Buffalo.....The Grosvenor Library
Engineering Society of Buffalo
Buffalo Public Library
Ithaca.....Cornell University
Jamaica, L. I.....Queens Borough Public Library
New York.....Engineering Societies Library
Public Library
College of the City of New York
Cooper Union
Columbia University
New York Museum of Science and Industry
New York University Library
Potsdam.....Clarkson College of Technology
Rochester.....Rochester Engineering Society
Schenectady.....Union College
Syracuse.....Syracuse University
Public Library
Troy.....Rensselaer Polytechnic Institute
Utica.....Public Library
Yonkers.....Public Library

North Carolina

Chapel Hill.....University of North Carolina (Engineering Library)
Raleigh.....North Carolina State College

North Dakota

Fargo.....North Dakota State Agricultural College
Grand Forks.....University of North Dakota

Ohio

Ada.....Ohio Northern University
Akron.....Public Library
University of Akron
Canton.....Public Library
Cincinnati.....University of Cincinnati
Public Library
Engineers Club of Cincinnati
Cleveland.....Public Library
Case School of Applied Science
Cleveland Engineering Society
Columbus.....State of Ohio Library
Public Library
Ohio State University
Dayton.....Engineers Club of Dayton
Toledo.....Public Library
University of Toledo
Youngstown.....Public Library

Oklahoma

Norman.....Oklahoma University
Oklahoma City...Public Library
Stillwater.....Oklahoma Agricultural and Mechanical College
Tulsa.....Public Library

Oregon

Corvallis.....Oregon State Agricultural College
Portland.....Portland Library Association

Pennsylvania

Allentown.....Free Library
Bethlehem.....Lehigh University
Easton.....Public Library
Lafayette College
Erie.....Public Library
Lewisburg.....Bucknell University
Philadelphia.....Engineers Club
Drexel Institute
University of Pennsylvania
Franklin Institute
Pittsburgh.....University of Pittsburgh
Engineers' Society of Western Pennsylvania
Carnegie Institute of Technology
Carnegie Library (Schenley Park)
Carnegie Free Library of Allegheny
Reading.....Public Library
Scranton.....Public Library
State College.....Pennsylvania State College
Swarthmore.....Swarthmore College
Villanova.....Villanova College
Wilkes-Barre.....Public Library

Rhode Island

Kingston.....Rhode Island State College
Providence.....Brown University
Providence Engineering Society
Public Library

South Carolina

Clemson College..Library, Clemson College

Tennessee

Kingsport.....Public Library
Knoxville.....University of Tennessee
Memphis.....Goodwin Institute
Nashville.....Vanderbilt University

Texas

Austin.....University of Texas
College Station...Agricultural & Mechanical College of Texas
Dallas.....Public Library
El Paso.....Public Library
Forth Worth.....Carnegie Public Library
Houston.....Rice Institute
Public Library
Lubbock.....Texas Technological College (School of Engineering)
San Antonio.....Carnegie Library

Utah

Salt Lake City...University of Utah
Public Library

Vermont

Burlington.....University of Vermont

Virginia

Blacksburg.....Virginia Polytechnic Institute
Charlottesville....University of Virginia
Norfolk.....Public Library
Richmond.....Virginia State Library

Washington

Pullman.....State College of Washington
Seattle.....Public Library
Engineers Club
University of Washington
Spokane.....Public Library
Tacoma.....Public Library

West Virginia

Morgantown.....West Virginia University

Wisconsin

Madison.....Library, University of Wisconsin
Milwaukee.....Public Library
Board of Industrial Education, Vocational School Library
Marquette University

Wyoming

Laramie.....Wyoming University

Depositories for A.S.M.E. Transactions Outside the United States

Argentine

Buenos Aires.....Biblioteca de la Sociedad Cientifica

Australia

Adelaide.....Public Library of Adelaide
Melbourne.....Public Library of Victoria
Perth.....University of Western Australia Library
Sydney.....Public Library, N. S. W., Sydney

Brazil

Rio de Janeiro....Bibliotheca da Escola Polytechnica
Bibliotheca Nacional
Sao Paulo.....Bibliotheca da Escola Polytechnica

Canada

Montreal.....McGill University
Engineering Institute of Canada
Toronto.....University of Toronto, Library

Chile

Santiago.....Universidad de Chile, Facultad de Ciencias Fisicas y Matematicas (Engrg. School)

Cuba

Havana.....Cuban Society of Engineers

Czechoslovakia

Prague.....Masarykova Akademie Prace
Society of Czechoslovak Engineers

Danzig Free City.....Bibliothek der Technischen Hochschule

Denmark

Copenhagen.....The Royal Technical College

England

Birmingham.....Birmingham Public Libraries
Bristol.....University of Bristol
Cambridge.....University of Cambridge
Leeds.....University of Leeds
Liverpool.....Public Library of Liverpool
Liverpool Engineering Society
London.....City & Guild Engineering College
Institution of Automobile Engineers
Institution of Mechanical Engineers
Institution of Civil Engineers
Institution of Electrical Engineers
The Junior Institution of Engineers
The Royal Aeronautical Society
Manchester.....Manchester Public Libraries (Reference Library)
Oxford.....University of Oxford
Newcastle-upon-Tyne.....The North East Coast Institution of Engineers and Shipbuilders
Sheffield.....Sheffield Public Libraries

Wales

Cardiff.....Cardiff Public Library

France

Lyons.....University of Lyons
Paris.....École Nationale des Arts et Metiers
École Nationale Supérieure de L'Aeronautique
École Centrale des Arts et Manufactures de Paris
Société des Ingénieurs Civils de France

Germany

Berlin.....Verein deutscher Ingenieure
Bibliothek der Technischen Hochschule
Breslau.....Bibliothek der Technischen Hochschule
Cologne (Köln)...Universitäts- und Stadtbibliothek
Dresden.....Bibliothek der Technischen Hochschule
Düsseldorf.....Bücherei des Vereines deutscher Eisenhüttenleute
Frankfort.....Technische Zentralbibliothek

Germany (continued)

Hamburg.....Bibliothek der Technischen Staatslehranstalten
Hanover.....Bibliothek der Technischen Hochschule
Karlsruhe.....Bibliothek der Technischen Hochschule
Leipsic.....Stadtbibliothek
Munich.....Bibliothek der Technischen Hochschule
Bibliothek des Deutschen Museums
Stuttgart.....Bibliothek der Technischen Hochschule

Hawaii

Honolulu.....University of Hawaii Library

Holland

Amsterdam.....Koninklijke Akademie von Wetenschappen
Delft.....Bibliotheek der Technische Hoogeschool
The Hague.....Koninklijk Instituut van Ingenieurs
Rotterdam.....Nationaal Technisch Scheepvaartkundig Institut

India

Bangalore.....Mysore Engineers Association
Calcutta.....Bengal Engineering College
Poona.....Poona College of Engineering
Rangoon.....University of Rangoon

Ireland

Belfast.....Queen's University of Belfast

Italy

Milan.....Biblioteca della R. Scuola d'Ingegneria
Comitato Autonomo per l'Esame della Invenzioni
Naples.....Biblioteca della R. Scuola d'Ingegneria
Rome.....Biblioteca della R. Scuola d'Ingegneria
Consiglio Nazionale delle Ricerche presso il Ministero della Educazione Nazionale
Turin.....Biblioteca della R. Scuola d'Ingegneria

Japan

Kobe.....Kobe Technical College
Tokyo.....Imperial University Library
The Society of Mechanical Engineers
Yokohama.....Library of Yokohama

Mexico

Mexico City.....Asociacion de Ingenieros y Arquitectos de Mexico
Library of the Escuela de Ingenieros Mecanicos y Electricistas

Norway

Oslo.....Den Polytekniske Forening

Poland

Warsaw.....Bibljoteka Publicazna

Porto Rico

Mayaguez.....University of Porto Rico

Portugal

Lisbon.....Institute Superior Technico

Roumania

Bucharest.....Scoala Polytechnica din Bucharest

Scotland

Glasgow.....Royal Technical College
Mitchell Library

South Africa

Cape Town.....University of Cape Town
Johannesburg.....South African Institute of Engineers

Sweden

Stockholm.....Kungl. Tekniska Hogskolan
Svenska Teknologföreninger
Gothenburg.....Chalmers Tekniska Institut

Switzerland

Zurich.....Eidgenossische Technische Hochschule

Turkey

Istanbul.....Robert College

U.S.S.R.

Kharkov.....Supreme Economic Council of Ukraine
Leningrad.....Leningrad Polytechnic Institute
Moscow.....Supreme Council of National Economy
Tomsk.....Tomsk Polytechnic Institute

Indexes to A.S.M.E. Papers and Publications

THE following pages contain indexes to papers and publications of the A.S.M.E. during 1933. A considerable number of papers and reports presented at meetings of the Society during the year were issued in mimeographed or photo-offset form, but at the time this index was prepared had not been included in any of the A.S.M.E. publications. Complete sets of these are on file for reference purposes at the office of the Society and the Engineering Societies Library, New York, under the title of "Miscellaneous Papers Presented at A.S.M.E. Meetings in 1933." Photostatic copies of any of the papers may be secured from the Library at regular rates. A list of these papers and reports is furnished here, grouped under headings corresponding to the Sections of Transactions.

This section of the Record and Index also includes a list of regular and special publications, and indexes to *Mechanical Engineering* and the Transactions.

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Standardization of Freight Equipment Cars by the Mechanical Division of the American Railway Association	F. H. Hardin	60e
Development of Passenger Cars	Peter Parke	60f
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Locomotive Running Gear and Counterbalancing	A. B. Trumbull	60j
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Mechanical Catalog, 1933–1934 edition
The Engineering Index for 1932

SPECIAL PUBLICATIONS

Boiler Codes

Power Boiler Code, Sections I, II, VI, and Appendix, A.S.M.E. Boiler Construction Code, September, 1933
Revisions and Addenda to Power Boiler Code, Locomotive Boiler Code, Low-Pressure Heating Boiler Code, and Unfired Pressure Vessel Code, August, 1933
Suggested Rules for the Care of Power Boilers, Section VII, A.S.M.E. Boiler Construction Code, March, 1933

Power Test Codes

Instruments and Apparatus
Part 3, Temperature Measurement—Chapter 8 on Optical Pyrometers, August, 1933
Part 4, Head Measuring Apparatus, March, 1933

Research

Report of Tests on Electrical Equipment for Drilling Rotary-Drilled Oil Wells, April, 1933
Determination of Carbonate, Phosphate and Hydroxide in Boiler Waters, September, 1933
Fluid Meters, Part 3, Their Selection and Installation, December, 1933

Standards

Wrench Head Bolts and Nuts and Wrench Openings (revision of 1927 edition), March, 1933

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